

Comparison of Quantitative In-Cylinder Equivalence Ratio Measurements with CFD Predictions for a Light Duty Low Temperature Combustion Diesel Engine

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ABSTRACT

In a recent experimental study the in-cylinder spatial distribution of mixture equivalence ratio was quantified under non-combusting conditions by planar laser induced fluorescence (PLIF) of a fuel tracer (toluene). The measurements were made in a single cylinder, direct injection, light duty diesel engine at conditions matched to an early injection low temperature combustion mode. A fuel amount corresponding to a low load (3.0 bar indicated mean effective pressure) operating condition was introduced with a single injection at -23.6° ATDC. The data were acquired during the mixture preparation period from near the start of injection (-22.5° ATDC) until the crank angle where the start of high-temperature heat release normally occurs (-5° ATDC). In the present study the measured in-cylinder images are compared with a fully resolved three dimensional CFD model, namely KIVA3V-RANS simulations. The impacts of computational grid resolution and of the flow initialization method are discussed as they pertain to the mixture preparation process. The simulation results indicate that a fine grid resolution is required to capture the nominal spray penetration in the experiments, however coarse and fine grids are shown to yield similar overall fuel distributions at the start of combustion, which is ultimately important for predicting combustion characteristics and engine-out emissions. It was found that the simulations do an excellent job at reproducing the experimental observation that the majority of the incomplete combustion products (UHC and CO) stem from overly lean mixtures at the start of combustion, which reside in the upstream portions of the fuel spray, near the injector. Additionally, plume-to-plume variability seen in the experiments is explained by possible irregularities in the fuel injector resulting in slightly different fueling rates per injector hole.

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INTRODUCTION

Due to their high compression ratio and overall lean operation, direct injection compression ignition (DICI) engines are a proven short-to mid-term solution to reducing CO₂ emissions and crude oil usage. However, conventional diesel combustion results in high oxides of nitrogen (NO_x) and particulate matter (PM) emissions. Stringent regulations for NO_x and PM emissions have required many engine manufacturers to use exhaust after-treatment, which adversely affects fuel economy and increases overall cost. To reduce the cost, complexity, and fuel economy penalty of using after-treatment systems, many new compression ignition

combustion strategies are being investigated to meet emissions mandates in-cylinder.

Homogeneous Charge Compression Ignition (HCCI) combustion, in which the fuel and air are completely premixed, has been shown to have advantages in terms of thermal efficiency, resulting from lower heat transfer losses and shorter combustion durations [1]. In addition, the lean, low temperature combustion process yields near zero NO_x and soot emissions. Although HCCI combustion appears thermodynamically attractive, the HCCI concept presents a controllability challenge, as it gives up two combustion control aspects. First, the timing of ignition is not controlled, by either fuel injection, as in a DICI engine, or by the spark, as in a SI engine, but rather by the chemical kinetics of the

fuel-air mixture. Second, the rate of heat release is not controlled, by the rate of fuel injection, as in a DIC engine, or by finite turbulent flame propagation, as in a SI engine, but is also dictated by the combustion kinetics. As a result, the near constant volume combustion event can lead to a very rapid rate of heat release, and hence a very rapid rate of pressure rise. These advantages and challenges of HCCI combustion have motivated the investigation of Partially Premixed Compression Ignition (PPCI).

To avoid high NO_x and soot formation rates, the fuel must have sufficient time to mix before ignition takes place. In the case of PPCI the fuel is injected early in the cycle to promote mixing before ignition, but maintain a close coupling between fuel injection and combustion to provide engine control. Diesel PPCI was characterized by researchers at Lund University in a heavy duty engine [2]. They found that the traditional NO_x-soot trade-off that exists with conventional DIC diesel combustion can be defeated with diesel PPCI, but at the cost of a decline in combustion efficiency. They presented EGR sweeps at varying loads and it was shown that at EGR rates of 60% and higher, the NO_x emissions are near zero and the soot emissions begin to decline sharply. Unfortunately, this high EGR level resulted in combustion efficiencies below 90%, which was indicated by a dramatic rise in CO and UHC emissions.

In an effort to understand why highly dilute PPCI combustion can lead to poor combustion efficiency, studies of the kinetics of fuel oxidation [3] have shown that to complete the oxidation process of UHC and CO under highly dilute, low temperature conditions, the local fuel-air equivalence ratio (ϕ) must be maintained within strict limits. Overly rich mixtures ($\phi > 1.4$) cannot complete oxidation due to a deficiency in O₂, while overly lean mixtures ($\phi < 0.45$) do not completely oxidize due to the low combustion temperatures. At low load conditions, UHC and CO emissions stemming from overly lean mixtures have been shown to be particularly problematic [4].

There are several optical diagnostic techniques that have been used to measure the local equivalence ratio in the cylinder of an operating engine, including Raman scattering, Rayleigh scattering, and laser-induced fluorescence (LIF). Of these techniques, laser-induced fluorescence is generally preferred due to its high signal level and the ability to discriminate the signal from elastically scattered and reflected laser light. Hence, laser-induced fluorescence techniques have often been used to qualitatively visualize the fuel distribution in the cylinder [5]. Musculus et al. [4] acquired quantitative equivalence ratio measurements under non-reacting conditions in a large-bore, heavy duty diesel engine using toluene as a fuel tracer. However, quantitative measurements of the fuel distribution in a light duty diesel engine, in which fuel jet interactions with the piston surfaces and swirling in-cylinder flows are much more significant, are bound to be very different. The qualitative differences

between fuel sprays seen in large bore, heavy-duty engines and those in light duty engines is shown in Figure 1.

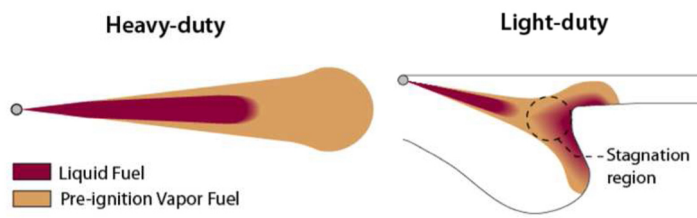


Figure 1. Qualitative differences between free jets that are experienced in large bore heavy duty engines and constrained jets from smaller, light duty engines.

Using the diagnostic techniques of Musculus et al. [4], Sahoo et al. [6] determined the spatial distribution of the fuel/air equivalence ratio within two horizontal planes in a light duty diesel engine: one in the squish region and the other deep in the piston bowl. The measurements were made under non-combusting conditions by planar laser-induced fluorescence (PLIF) of a fuel tracer. Despite the opposing squish flow, the fuel jets penetrate through the squish region to approximately the cylinder liner. Although rapid mixing is observed in each jet, the equivalence ratio remains near stoichiometric in the head of the jet at the start of high-temperature heat release (HTHR). In contrast, the fuel mixture is generally lean in the clearance volume above the bowl, though near-stoichiometric pockets are observed near the bowl rim. Significant pockets of mixture in the clearance volume above the bowl and between the jets are sufficiently lean that complete oxidation is unlikely.

The goal of the present study was to compare engine CFD simulations, using the KIVA3V-RANS model, to the experimental results of Sahoo et al. [6] at varying crank angles. The impact of the computational grid resolution and of the flow field initialization method is discussed. It was observed that the simulations can reproduce the plume-to-plume variability seen in the experiments by assuming small irregularities in the fuel injector that could lead to slightly different fueling rates per injector hole. It was found that the simulations do an excellent job at reproducing the experimental observation that the majority of incomplete combustion products (UHC and CO) stem from overly lean mixtures at the start of combustion.

EXPERIMENTAL MEASUREMENTS OF LOCAL EQUIVALENCE RATIO

The single-cylinder optical engine used in the experiments of Sahoo et al. [6] was developed from a General Motors 4-cylinder 1.9 L light duty diesel engine. The engine and injector geometry is summarized in Table 1 and a schematic of the engine is shown in Figure 2. The piston top was constructed from fused silica, but retains the geometric

characteristics of a production-like piston geometry featuring a reentrant bowl design and piston crown valve cutouts.

Table 1. Engine and injector specifications for the experiments of Sahoo et al. [6].

Engine Geometry

Bore / Stroke [mm]	82.0 / 90.4
Displacement [cc]	477
Compression Ratio	16.7
Squish Height [mm]	0.88

Injector Geometry

Bosch Common Rail CRIP2.2

Sac Volume [mm ³]	0.23
Number of Holes	7
Hole Diameter [mm]	0.14
Included Angle [deg]	149
Tip Protrusion [mm]	2.1
Hole Protrusion [mm]	0.3

The Bosch CRIP2.2 injector is a production, high pressure, solenoid-driven, common-rail fuel injector which was centrally mounted in the cylinder head. For the experiments compared to here, a 7-hole, mini-sac injector tip was employed, having an included angle of 149° (15.5° down-angle from the firedeck; i.e., the bottom surface of the head). The 7 fuel holes are equally spaced and have a nominal diameter of 0.14 mm. The injector tip protrusion (i.e., distance the tip of the injector protrudes into the combustion chamber past the firedeck) of 2.1 mm was determined with knowledge of the injector body geometry, head geometry, and washers used in the installation. Then, using the engineering drawing of the injector tip, the protrusion of the injector holes could be determined (i.e., location where injector holes discharge fuel), which was 0.3 mm.

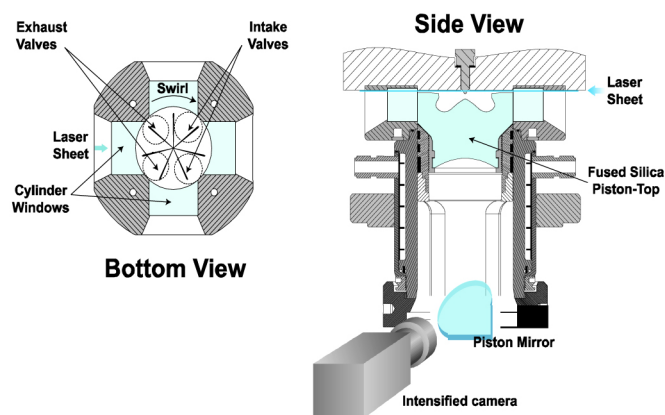


Figure 2. Schematic of the light duty optical diesel engine depicting the horizontal laser sheet location and camera setup. The left-hand image shows the cylinder head from the bottom, as seen by the camera. This is the orientation in which the experimental images will be presented [6].

To avoid oxygen quenching of the toluene fluorescence, the engine was operated with a pure nitrogen intake stream and thus under non-combusting conditions. The near-TDC temperature and density of a baseline combustor operating condition were closely matched by maintaining the same intake charge mass flow rate and lowering the intake temperature to account for the lower specific heat of the pure N₂ charge. The operating conditions for the baseline combustor case and the non-reacting case are shown in Table 2.

Table 2. Engine operating conditions for the baseline low-temperature combustor case and the non-reacting condition studied in this work.

	Baseline	Current
Engine Speed [rpm]	1500	1500
Intake Charge [Mole Fractions]	O ₂ : 0.10 N ₂ : 0.81 CO ₂ : 0.09	N ₂ : 1.0
Intake Temperature [K]	372	300
Intake Pressure [bar]	1.5	1.5
Injected Fuel [mg]	8.8	8.8
Fuel Rail Pressure [bar]	860	860
Global Equivalence Ratio	0.3	N.A.
Start of Injection [° ATDC]*	-23.6	-23.6
Injection Duration [° CA]	4.5	4.5
Swirl Ratio	2.2	2.2
Motored TDC Density [kg/m ³]	20.9	21.1
Motored TDC Temp. [K]	909	908

*Actual start of fuel injection, which was optically confirmed in the engine.

Commercial fuels contain numerous compounds that strongly fluoresce upon excitation with ultra-violet (UV) light. In addition, the fluorescence signal level depends not only on the concentration of each fluorescing component in the fuel, but also on temperature, pressure, and local mixture composition. Therefore, a quantitative interpretation of the signal as a fuel/air equivalence ratio is difficult with commercial fuels. To address this problem a single fluorescing component (tracer) is mixed with a non-fluorescing base fuel. The fluorescing tracer used in the experiments was toluene (C₇H₈), which was added to the base fuel at a rate of 0.5% (by weight). The non-fluorescing base fuel was a mixture of 75% (by volume) HPLC-grade n-heptane (C₇H₁₆) and 25% HPLC-grade iso-octane (C₈H₁₈). One challenge with a fuel tracer diagnostic is that there is an inherent assumption that the local fuel concentration is proportional to the tracer concentration. In other words, it is assumed that the base fuel and the fuel tracer evaporate and mix simultaneously. Thus, it is important that the physical properties of the base fuel and the tracer are closely matched. In this case, the boiling points are very similar, iso-octane (99° C), n-heptane (98° C), and toluene (110° C), so the

toluene should track the base fuel reasonably well. At the baseline combustng operating condition summarized in Table 2, this fuel mixture matches the heat release characteristics of a conventional US diesel fuel with a cetane number of 47 [6]. Previous studies [7] have shown that the engine-out emissions of UHC and CO at this condition are also well matched between this PRF blend and diesel fuel.

Table 3. Locations of the horizontal laser sheet as a function of crank angle.

Laser Sheet Position	
Crank Angle [° ATDC]	[mm below firedeck] Squish Plane/Bowl Plane
-22.5	2.5/N.A.
-20	2.1/N.A.
-17.5	1.7/13.0
-15	1.4/12.4
-12.5	1.1/11.8
-10	0.8/11.3
-7.5	0.6/11.0
-5	0.5/10.7

CFD simulation results were compared to the experimental images at two different regions of the combustion chamber: the squish plane and the bowl plane. The vertical locations of the laser sheet for these two distinct regions are listed in Table 3. In addition, a depiction of the squish plane and the bowl plane at -15° ATDC is shown in Figure 3. At each crankshaft position, a set of 100 images, comprised of images from 100 separate cycles, were acquired and ensemble averaged. All images were collected with the camera looking up through the piston, toward the bottom surface of the cylinder head, as shown in Figure 2.

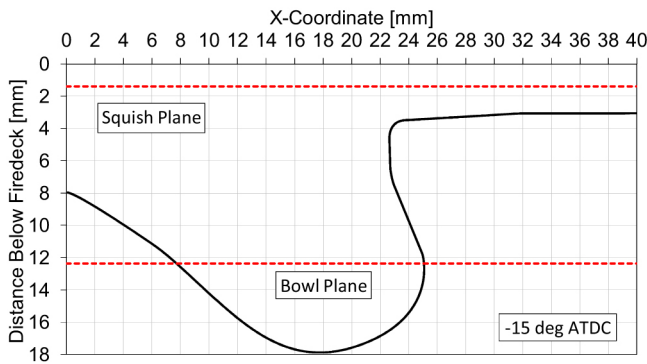


Figure 3. Illustration of horizontal laser sheet locations at -15° ATDC.

Toluene fluorescence was excited by the 266-nm (UV) output of a 10-Hz pulsed Nd:YAG laser, with a pulse energy of 40 mJ. The laser beam was focused in the cylinder using a cylindrical lens, resulting in a laser sheet thickness of approximately 500 μm across the cylinder bore. For the sake of brevity, the detailed image processing steps for calculating the equivalence ratio from the observed toluene fluorescence

are not discussed here. For a detailed description of the experimental methodology, refer to the experimental publication by Sahoo et al. [6]. However, an abbreviated description of the experimental procedure and major image processing steps - and the unavoidable challenges associated with this laser based diagnostic - is given below.

Sahoo et al. [6] processed the acquired data images by comparing them to calibration images, which were carefully prepared with a homogeneous fuel/air mixture with a known quantity of fuel. Additionally, the temperature sensitivity of the toluene fluorescence was accounted for in the data images. Spurious background signal from the windows and injector tip was eliminated by subtracting an average background image that was obtained with no fuel present in the chamber. A WG295 long-wave-pass filter rejected elastically scattered 266-nm laser light, so its contribution to the background is minimal as well. Attenuation of the laser sheet was negligible and hence not accounted for in the fluorescence calculations. Laser light attenuation in the data images could be significant at early crank angles when there is liquid present and regions of very high equivalence ratio. However, due to the low toluene concentration used in the base fuel, it is likely to be a small effect. As will be shown in a later section, the spray plumes with the highest observed equivalence ratio are on the right-hand side of the experimental images, after the laser sheet has traversed the entire cylinder bore, supporting the notion that laser attenuation is minimal.

Due to the reentrant piston bowl geometry it was not possible to obtain any meaningful data in the squish volume region directly above the reentrant portion of the bowl. Thus, measurements in this region were not reported. This region is indicated in the experimental images, as well as the CFD simulations, by a white ring. Also, the laser sheet was partially blocked by the injector tip at crank angles later than -20° ATDC. Within this shadow measurements are likewise not reported. Ultimately the processed, ensemble averaged data images at each crank angle provide the local fuel mole fraction, χ_{fuel} . The measured fuel mole fraction can be used to calculate the local fuel/air equivalence ratio for the baseline combustng condition using Equation 1. The value of 113.4 corresponds to a charge composition consisting of 10% intake oxygen (O_2).

$$\phi = \frac{113.4 \chi_{fuel}}{1 - \chi_{fuel}} \quad (1)$$

Even though careful steps were taken to account for reflections and elastically scattered laser light, both of which may cause artificial fluorescence to be detected, ultimately there are two sources of interference which could not be corrected: reflections of laser induced fluorescence from images with a wide range of equivalence ratios and fluorescence from liquid located outside of the laser sheet that is excited by scattered laser light. The first source of

interference will be most severe at the earliest crank angles up until shortly after the end of injection, at which point it will begin to dissipate. Reflections of fluorescence from regions of high equivalence ratio can cause a significant overestimation of the equivalence ratio in regions of lower equivalence ratio. This interference can result in an apparent equivalence ratio of ~ 0.3 in a region where no fuel is expected to be present. The second source of interference will also result in an overestimation of the local equivalence ratio. In this case, scattered laser light excites fluorescence from liquid droplets that can be captured by the camera. This source of interference will also be greatest at early crank angles and begin to dissipate after the end of injection. The CFD simulations indicate that nearly all the liquid fuel has evaporated by -15° ATDC, which is 4.1° after the end of injection. Formal quantification of the accuracy of the experiments is difficult due to the aforementioned interferences, but is currently a work in progress. However, the unavoidable interferences noted above should have little to no impact on the experimental results at later crank angles (i.e., later than -15° ATDC).

SIMULATION METHODOLOGY & GRID CHARACTERISTICS

MULTI-DIMENSIONAL ENGINE CFD

Multi-dimensional engine CFD codes are an important part of the engine development process today. Powerful simulation tools allow engineers and researchers to develop combustion strategies in a timely and cost effective manner. As stated in the introduction, the main focus of this research was simulation validation for the fuel spray and subsequent mixture preparation processes in a light duty diesel engine operating under partially premixed conditions. In this work the KIVA-ERC code, which is based on the KIVA3V Release 2 code, was used [8, 9, 10]. The KIVA3V code uses the most popular approach for modeling engine sprays, the Lagrangian-Drop Eulerian-Fluid (LDEF) method, in which liquid fuel is treated as Lagrangian parcels and the ambient gas is discretized into cells. The Lagrangian liquid fuel parcels traverse the stationary Eulerian mesh, exchanging momentum with the gas phase.

In the optical engine experiments a mixture of 75% n-heptane/25% iso-octane by volume was used for the base fuel. Considering these are both pure substances, the physical properties of the fuel (for spray break-up, evaporation, and mixing) were taken to be exactly that of the previously stated mixture, instead of using a fuel surrogate, which is typically done with commercial fuels. The 0.5% by weight toluene fuel tracer was not considered in the fuel composition. In the CFD simulations the intake charge was pure N_2 and the equivalence ratio was calculated using Equation 1 where the mole fraction of fuel, χ_{fuel} , is taken to be the sum of iso-octane and n-heptane in a given CFD cell.

Several sub-model improvements have been made to the KIVA3V code, including fuel droplet break-up, droplet evaporation, droplet collision modeling, and turbulence modeling. Beale and Reitz implemented a hybrid spray break-up model, namely the KH-RT model [11]. The KH-RT model consists of two distinct steps: primary and secondary break-up. The Kelvin-Helmholtz (KH) instability model was used to predict the primary break-up of the initially injected fuel parcels, which are introduced into the combustion chamber using the blob method [12]. The secondary break-up (i.e., aerodynamic break-up) was modeled as a competition between Kelvin-Helmholtz and Rayleigh-Taylor (RT) instabilities. In an attempt to reduce the grid size dependence of the Lagrangian spray model, the GASJET model of Abani et al. [13] was used to model the relative velocity between the fuel droplets and the gas phase in the near nozzle region. The droplet collision model is from the original KIVA3V code and is based on O'Rourke's model. However, a radius of influence method was implemented to reduce the mesh dependency of the collision process [13]. The interaction of the liquid fuel parcels with the wall was modeled with the standard KIVA3V wall film model developed by O'Rourke and Amsden [14]. Ra and Reitz made improvements to the droplet vaporization sub model, which now allows for discrete multi-component fuel sprays to be considered [15]. Turbulence is modeled using the RNG k- ϵ turbulence model as coded in KIVA3V. The original RNG model was developed for incompressible flows. However, internal combustion engine flows are compressible and subject to large density changes. Considering this, Han and Reitz modified and closed the epsilon transport equation through an isotropic rapid distortion analysis. This modified RNG k- ϵ turbulence model was implemented into the KIVA3V code by Han and Reitz [16]. Previous computational research at this operating condition studied the effects of using the k- ϵ versus the RNG k- ϵ turbulence model [17]. The differences in overall vapor penetration were found to be small, with the RNG model showing slightly less fuel/air mixing during the fuel injection process.

IN-CYLINDER COMPUTATIONAL GRIDS

As noted earlier, the optical engine employed a fused silica piston that retained the geometric characteristics of a production-like piston featuring a reentrant bowl and piston-top valve cut-outs. In order to conduct axisymmetric (sector mesh) simulations, a compromise must be made to determine the squish height (i.e., distance from the cylinder head to the piston top at top dead center) because the details of the valve cut-outs cannot be considered. Figure 4 illustrates how the average squish height was determined for the sector mesh simulations. The region of the piston-top that is flat and not tapered is encompassed by the red circles in the left image. The image on the right shows points on the piston-top inside the encompassed region illustrated on the left, which is

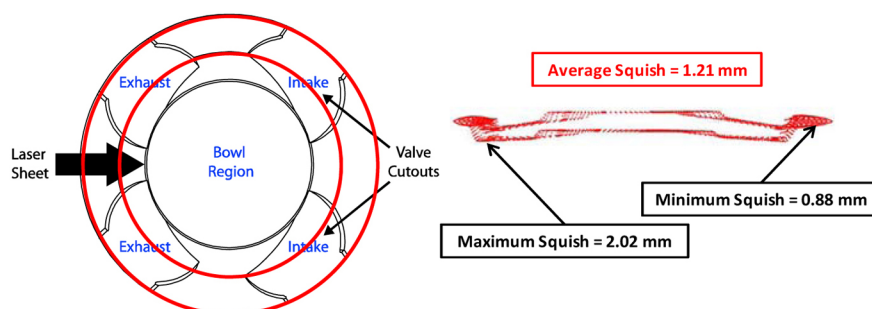


Figure 4. Determination of squish height for sector mesh simulations. Left image shows the details of the experimental piston-top with the non-tapered flat portion encompassed by the red circles. Right image shows points on the piston-top inside the encompassed region illustrated on the left, which is shown in a side view in order to highlight the variation in the depth of the piston-top valve cut-outs.

shown in a side view in order to highlight the variation in the depth of the piston-top valve cut-outs. As can be seen, the minimum squish height is 0.88 mm, which is the quoted value in the engine geometry table, [Table 1](#). The largest squish height is 2.02 mm and occurs in the deepest portion of the valve cut-outs. The effective squish height was determined by averaging all the squish heights in this flat piston-top region. The resulting squish height is 1.21 mm and was used for all three sector meshes shown in [Figure 5](#).

The equations that govern scientific and engineering problems that are of practical interest rarely have an analytical solution. Full 3D computational fuel dynamics (CFD) is no exception. CFD solutions solve the Navier-Stokes equations approximately using numerical approximations of derivatives in time and space. Whenever a set of equations is solved with numerical approximations, an artificial numerical viscosity is introduced, which adds to the actual molecular viscosity and over predicts the diffusion term. This artificial viscosity increases as the grid size increases. To understand the grid sensitivity of multi-dimensional engine CFD with liquid sprays three different grid resolutions were used in this study, which are shown in [Figure 5](#). These grids are sector meshes that only consider one of the seven fuel injector holes. This modeling approach assumes the combustion chamber is axisymmetric and these grids have periodic boundaries to the left and right of the fuel spray. Considering that combustion strategy development using CFD is typically done using sector meshes, it was of interest to conduct the spray and mixing validation on these sector meshes as opposed to a full 3D simulation that considers all seven injector holes.

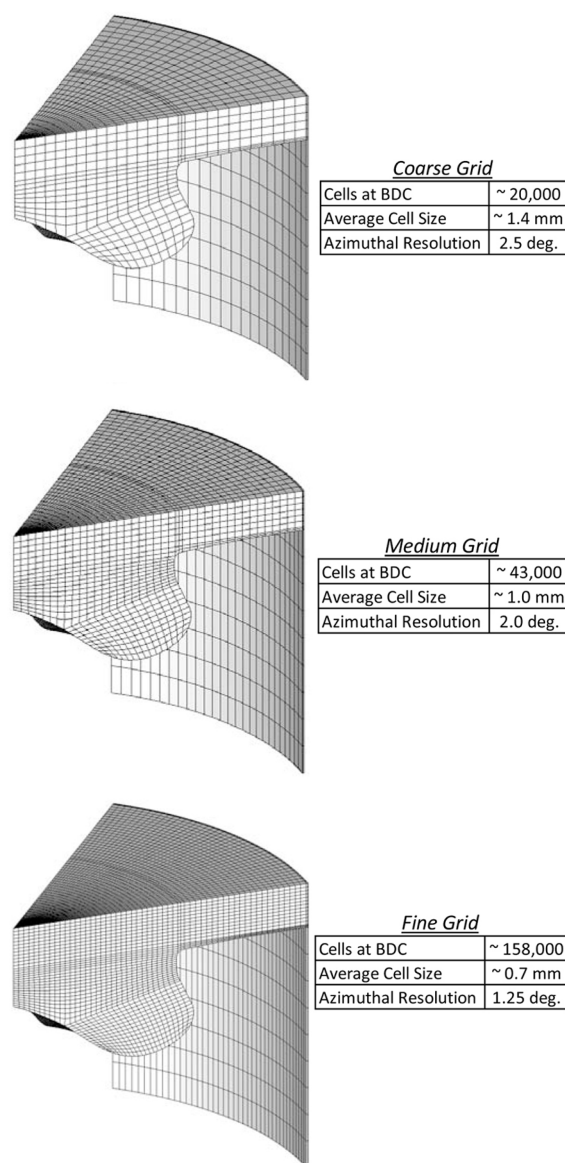


Figure 5. Computational grids used for sector mesh simulations of one fuel injector hole (i.e. 51.4° sector).

As shown in [Table 1](#), the optical engine used in the experiments had a geometric compression of 16.7. However, to accurately match a motored trace with the CFD simulations an “effective” compression ratio of 15.7 needed to be used. The added volume needed to reach this compression ratio was put at the bottom of the top land ring crevice. The reason for this needed reduction in compression ratio is due to the fact that the CFD simulations do not consider any mass loss from the combustion chamber and that the extended piston assembly used in the optical engine may compress slightly under high pressure. As shown in [Figure 6](#), the simulated motored trace is entirely grid independent and matches the experimental trace very well. The only discrepancy is slight over prediction of the cylinder pressure near top dead center (TDC), where mass loss (i.e., blow-by) and extended piston compression could be important in the experimental trace. Considering that the crank angle range of interest for this study is -25° ATDC to -5° ATDC, the CFD simulations reproduce the in-cylinder pressure conditions very accurately, which is extremely important for spray penetration and mixing.

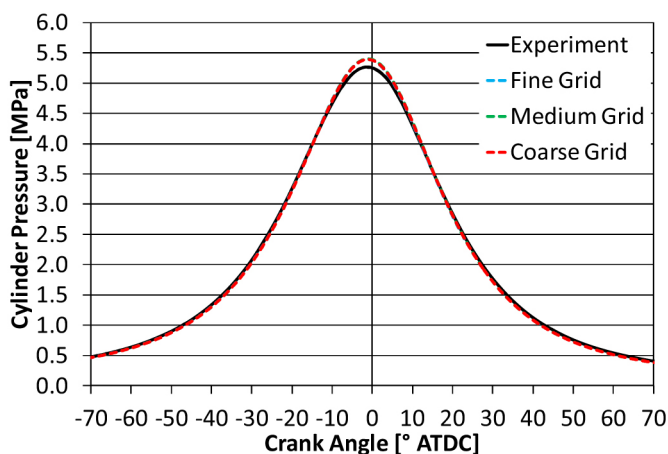


Figure 6. Experimental motored trace vs. CFD simulations.

In order to study the impact of the flow initialization process on the resulting fuel spray and mixture preparation, full 3D (360°B intake simulations were conducted [18]. The full 3D computational grid is shown in [Figure 7](#) and includes details such as valve recesses in the head and valve cut-outs in the piston. Once the intake simulations were complete, the resulting flow field surrounding each of the seven injector holes in a 51.4° sector was mapped over to the medium sector mesh to study the impact of considering the intake process.

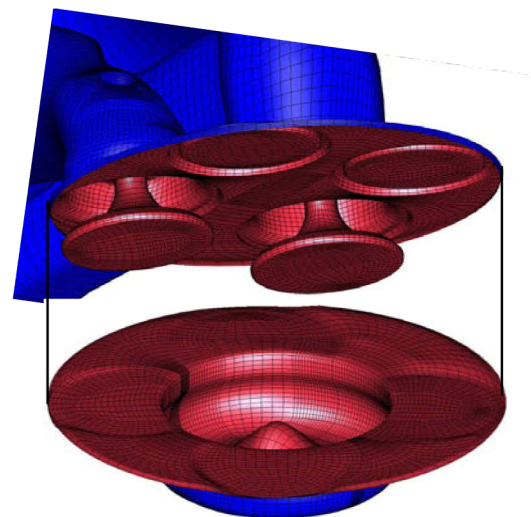


Figure 7. Geometric details of the full 3D intake simulation computational grid, which includes features such as valve recesses in the cylinder head and valve cut-outs in the piston [18].

RESULTS

IMPACT OF GRID RESOLUTION

As mentioned previously, CFD simulations are an approximate solution to the governing equations and are inherently grid dependent due to numerical viscosity. In addition, LDEF spray simulations have been shown to be extremely grid dependent on relatively coarse grids [13]. Considering these notions, the first study conducted in this work was to understand the impact of grid resolution. The following section steps through comparisons of the simulation results to the experimental images just after the start of injection through -5° ATDC, which is the start of HTHR.

In these figures, the top row of images show the ensemble averaged experimental results from 100 separate cycles in which the laser sheet entered from the left, just as shown in the bottom view of [Figure 2](#). The full experimental image shows the entire field of view with all seven spray plumes. The average experimental image is the ensemble average of all seven spray plumes and is plotted in the position of plume #1, which was arbitrarily chosen. Note the field of view extends out 36 mm from the cylinder axis, which is 5 mm short of the cylinder liner. The equivalence ratio (ϕ) scale that is displayed at the top of the experimental images is the same scale for the simulation results at each respective crank angle. The first row of simulation results is a horizontal cut plane that is in the same location as the experimental laser sheet, which varies with crank angle, as shown in [Table 3](#). The black line bisecting these simulation images corresponds to the location of a vertical cut plane, which is shown in the second row of simulation images. This first viewing angle