

Experiments and Modeling of Adaptive Injection Strategies (AIS) in Low Emissions Diesel Engines

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ABSTRACT

Homogeneous Charge Compression Ignition (HCCI) has been shown as a promising technique for simultaneous NO_x and soot reduction while maintaining diesel-like efficiency. Although HCCI has been shown to yield very low emissions levels, spray-wall impingement and high pressure rise rates can be problematic due to the early injection timings necessary for certain HCCI operations. To address spray-wall impingement, an Adaptive Injection Strategy (AIS) was employed. This strategy uses multiple pulses at both low and high injection pressures to prepare an optimal in-cylinder mixture.

A unique Variable Pressure Pulse (VPP) was developed to investigate the AIS concept experimentally. The VPP has the capability of delivering multiple injections at both low and high injection pressures (~100 bar and ~1000 bar respectively) through a single injector in the same engine cycle. Comparisons were made between model predictions and engine experiments using the VPP system. The models were able to adequately capture the emissions and performances trends observed in the experiments.

This study uses engine experiments and a multi-dimensional CFD code coupled with detailed chemistry and a Multi-Objective Genetic Algorithm (MOGA), the KIVA-CHEMKIN-MOGA code, to explore the AIS concept. The computational optimization was performed at a high speed – light load operating condition (2000 rev/min and 5.5 bar IMEP) considering six objectives (NO_x, soot, CO, HC, ISFC, and peak PRR) and seven

parameters (IVC timing, EGR rate, fuel split, early injection timing, late injection timing, early injection pressure, and late injection pressure). The results show low pressure early injections are key to minimizing spray-wall impingement. Furthermore, a Pareto solution was found with near zero NO_x and soot, a peak pressure rise rate of 6.3 bar/deg, and a net ISFC of 190 g/kW-hr.

INTRODUCTION

Rising fuel costs and a focus on reduction of greenhouse gases has driven a need for increased efficiency from the internal combustion engine. This need for increased efficiency has placed the diesel engine in the spotlight. Furthermore, current and future legislation place stringent standards on the exhaust emissions. These demands require development of advanced engine technology, which can simultaneously yield high efficiency and low emissions.

Many strategies have been investigated to meet the future requirements of clean, efficient diesel engines. Most of the current strategies can be lumped into the category of low temperature premixed combustion. Lower combustion temperatures result in NO_x reduction due to the high activation energy of NO formation reactions [1]. In addition, utilizing long ignition delay allows adequate time for mixing prior to the start of combustion; thus, rich regions are reduced and soot formation is inhibited. Many researchers have shown Homogeneous Charge Compression Ignition (HCCI) and Premixed Charge Compression Ignition (PCCI) concepts

to be promising techniques for simultaneous NO_x and soot reduction [2-4]. Sun [5] has shown the PCCI, HCCI, and traditional diesel operating regimes on the equivalence ratio-temperature diagram, see Figure 1. This diagram was originally presented by Kamimoto [6] and has been extensively discussed by Pickett et al. [7] and Akihamo et al. [8]

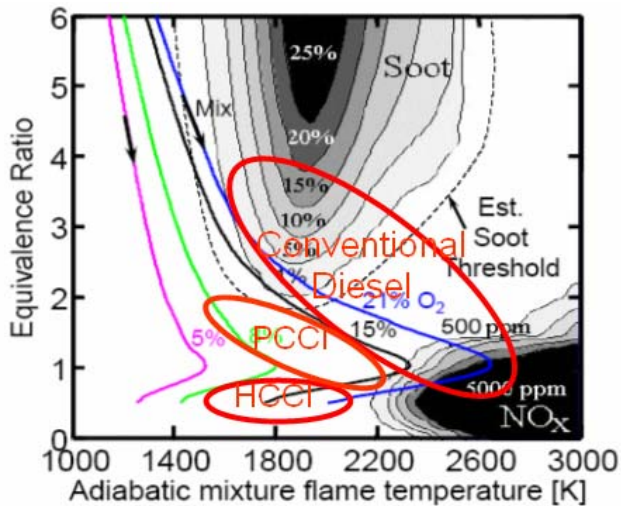


Figure 1 PCCI, HCCI, and conventional diesel operating regimes shown in T- Φ space [5]

Although, HCCI and PCCI combustion have been shown to yield very low NO_x and soot levels, several problems must be solved before these strategies can see widespread use. The first problem is spray-wall impingement. Because a premixed charge is required, fuel must be injected early in the cycle to allow adequate time for mixing; however, early cycle charge densities are often low which can result in very long liquid lengths (on the order of the cylinder bore) and spray-wall impingement. The second problem which must be addressed is combustion phasing control. Because the ignition delay must be extended to allow adequate time for mixing, injection timing can no longer be used to control combustion phasing. Therefore, chemistry time-scales must also be extended (i.e., slow down reaction rates via lower gas temperature or reduced oxygen concentration) so that ignition does not occur prematurely. The third problem which must be addressed is the rapid heat release associated with premixed combustion strategies. Because the combustion rate is no longer limited by the mixing process as in conventional diesel combustion, the heat release can be rapid and result in excessive engine noise and structural damage.

Spray wall impingement has been addressed by many researchers [9-13]. Kawaguchi [14] proposed an injector capable of changing the cone angle depending on the piston position in order to maintain optimal spray targeting over a wide range of injection timings. Kohketsu et al. [15] developed an injection system that altered the pressure of the spray midstream by using a solenoid to switch between low and high pressure common rails allowing for the first part of the injection to

be at low pressure and the second portion at high pressure. Ra and Reitz [16] performed a numerical evaluation of the variable geometry spray (VGS) concept and found wall film levels at the time of ignition were reduced from 25% for a fixed geometry nozzle to below 5% for the VGS case. They also evaluated a more practical two-pulse spray concept which had two different fixed spray angles (i.e., a narrow angle for the first pulse and a wider angle for the second pulse) and found similar wall film levels as the VGS case. Sun and Reitz [17] showed the advantages of using low-pressure injections with a fixed spray angle as a means for reducing spray wall impingement for early injected fuel and proposed an adaptive injection strategy (AIS) concept which allows the use of low-pressure injections for fuel injected early in the cycle and high pressure injection of fuel injected late in the cycle. This investigation further explores the AIS concept as a means for achieving a premixed charge using direct injection.

Combustion phasing control of premixed compression ignition engines has received much attention in recent years. Sjöberg and Dec [18] have shown repeatable control of an HCCI engine by tuning the intake temperatures and EGR rate. Hasegawa and Yanagihara [19] explained the Uniform Bulky Combustion System (UNIBUS) which involves a dual injection strategy. An early injection (~50 °BTDC) creates a premixed charge. Engine parameters (intake gas temperature, IVC timing, fuel quantity, and injection timing) are set such that high temperature heat release does not occur prior to the start of late injection. The second injection occurs after TDC and is used to control the start of combustion of the premixed fuel. They found that after the late injection, combustion occurred at many points simultaneously throughout the chamber; thus suggesting HCCI combustion was achieved. The UNIBUS concept was able to achieve low NO_x and near-zero soot emissions.

For premixed combustion strategies to gain widespread acceptance the rate of heat release must be controlled. Sun and Reitz [5, 20] proposed a Two-Stage Combustion (TSC) concept as a means for extending the operating range of HCCI combustion. The first stage is HCCI-like combustion and the second stage is diffusion combustion under low oxygen concentration conditions. Note that in this paper, the terminology 'cool flame reaction' describes the low temperature reaction associated with forming a pool of hydrogen peroxide which will later be consumed in the high temperature reaction, 'First stage combustion' refers to high temperature heat release of an early injected or premixed fuel, and 'second stage combustion' refers to combustion of a late injected fuel. Because in the TSC strategy only a fraction of the total fuel is burnt in pure HCCI combustion, it may be possible to use the TSC concept for noise reduction. Kokjohn and Reitz [21] showed very low peak pressure rise rates could be achieved through optimized TSC. In their work, a homogenous charge was assumed at IVC and

combustion characteristics were explored by varying the parameters of the fuel injection event late in the cycle.

SCOPE AND OBJECTIVES

The main objective of this study is to explore Two-Stage Combustion and Adaptive Injection Strategies to address the several problems associated with premixed combustion strategies (combustion phasing control, combustion rate control, and spray-wall impingement). Operating conditions which will simultaneously result in low noise, high efficiency, and clean combustion will be identified and investigated numerically and experimentally. Additionally, tradeoffs between noise, emissions, and fuel consumption are identified. TSC/AIS operation is explored first through the use high fidelity computational fluid dynamics (CFD) with the aid of a multi-objective genetic algorithm (MOGA). Low noise, high efficiency, clean combustion points are selected from the optimization design space and discussed.

ENGINE SPECIFICATIONS

The engine used this study is a single cylinder version of the GM 1.9 L engine with specifications given in Table 1. To accommodate concurrent research projects at the Engine Research Center (ERC), validation experiments are performed on a slightly modified version of the engine of Table 1. The modifications are discussed in the model validation section.

Table 1 Engine specifications used for computational investigation.

Base engine type.....	GM1.9 L [†]
Bore x stroke.....	8.2 x 9.04 cm
Connecting Rod Length.....	16.1 cm
Squish height.....	0.0617 cm
Displacement.....	0.4774 L
Compression ratio.....	16.5:1
Swirl ratio.....	2.2
Bowl type.....	reentrant
Intake valve opening.....	344° ATDC Firing
Intake valve closing.....	-132° ATDC Firing
Exhaust valve opening.....	112° ATDC Firing
Exhaust valve closing.....	388° ATDC Firing

Table 2 Injector specifications used for computational investigation.

Injector type.....	High-pressure solid-cone
Manufacturer.....	Bosch
Injection pressure.....	860 bar
Included angle.....	155°
Number of holes.....	7
Nozzle hole diameter.....	141 μm

[†] A single cylinder version of the GM 1.9 L engine is used

EXPERIMENTAL SETUP

VARIABLE PRESSURE PULSE SYSTEM – To investigate the AIS concept experimentally, a Variable Pressure Pulse (VPP) system was developed in the Diesel Emissions Reduction Consortium (DERC) lab at the University of Wisconsin-Madison's ERC utilizing the Adaptive Injections Strategy (AIS) of Sun and Reitz [17]. The system is similar to Kohketsu's [15]; however the VPP is used to create two separate pressure pulses and not one pulse with variable pressure. The VPP system, shown in Figure 2, is capable of delivering two independent injections at various pressures. The first injection was designed to be at low pressure (10-35 MPa), which has been computationally shown to reduce wall wetting and to create a semi-homogenous charge within the cylinder [5]. The second injection is at high pressure (80-120 MPa) to promote air entrainment and droplet breakup.

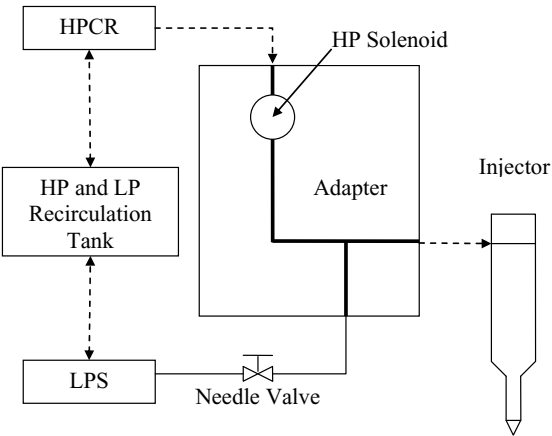


Figure 2 VPP system schematic.

The VPP system consists of a high pressure common rail (HPCR), a low pressure system (LPS), a high pressure solenoid, a recirculation tank, a needle valve, and an adapter. The adapter houses the solenoid, pressure transducer, and connects the HPCR, LPS, and injector. The solenoid is initially turned off for the low pressure injection, allowing the LPS to control the pressure in the adapter. After the first low pressure injection, the solenoid opens allowing fuel from the HPCR to flow into the adapter. The needle valve prevents massive amounts of fuel from escaping from the adapter and acts as a critical flow orifice only allowing a small amount of the fuel to pass into the LPS. The resulting effect is a rapid increase of pressure in the adapter. The second injection is then fired, and the solenoid is turned to the off position. The pressurized fuel bleeds through the needle valve into the LPS. Figure 3 shows the injector and solenoid signal timing. The solenoid is opened at the end of the first injection and held until the second injection is fully complete.

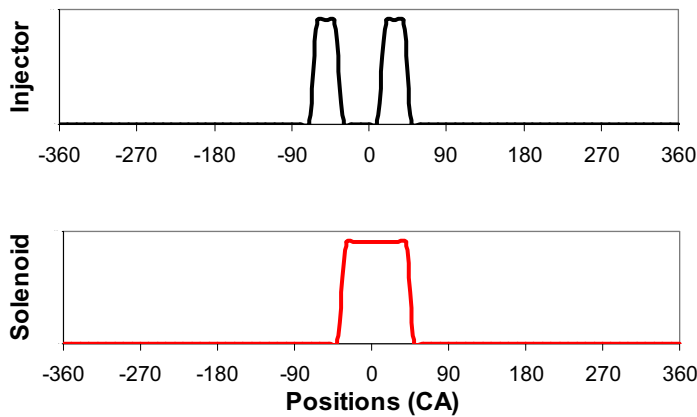


Figure 3 Injector and solenoid signals.

Rate of Injection – To visualize the injection events and the pressure rise rate in the adapter, the injector was mounted in a Bosch style rate of injection (ROI) bench [22]. The ROI is measured by recording the pressure wave produced by the injector when it fires into a length of compressible fluid. Furthermore, a Kistler piezoresistive high pressure sensor (Type 4067) was mounted in the adapter to monitor the injection pressures. Table 3 and Figure 4 show the injector commands and the corresponding ROIs. It can be seen that the duration of the first, low pressure injection is much longer than the second for the same amount of fuel, as expected. Furthermore, each injection event is repeatable and easily controlled. The rail pressure for shape 1 demonstrates the increase in pressure of the adapter. All the rail pressure traces were similar.

Table 3 Injector settings for ROI sweep.

Shape	Press. (bar)	SOI 1 (°CA)	ET 1 (μ-s)	SOI 2 (°CA)	ET 2 (μ-s)	RPM	Rate (mg/inj.)
1	600	65	1500	0	450	1788	11.8
2	600	65	1700	0	450	1788	13.2
3	600	65	1900	0	450	1788	14.0
4	600	65	1900	0	550	1788	18.5
5	600	65	1900	0	650	1788	23.8

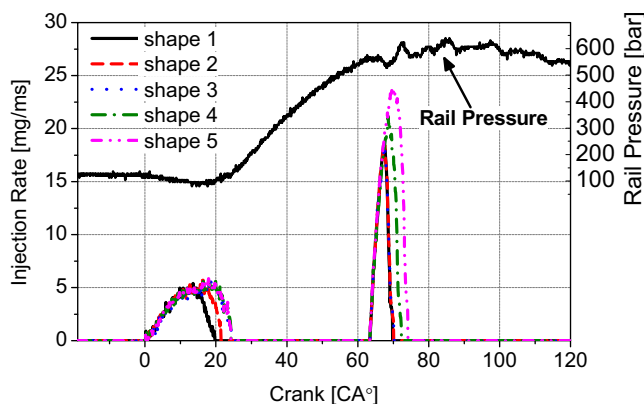


Figure 4 ROI and rail pressure for various injection parameters.

VPP System Capabilities – The VPP system can easily control the injection timing and duration of both injections. However, Figure 4 demonstrates that there is a limitation corresponding to the duration between the injections. It has been found experimentally that it takes approximately 30 CA at 1788 rev/min to pressurize the adapter. The second injection pressure set point does not affect the pressurizing duration. At higher second injection pressures, more fuel is pushed into the adapter, increasing the pressure rise rate in the system. An alternative adapter (shown in Figure 5) with smaller total volume increases the pressure rise rate, allowing for higher second injection pressures and lowering the pressurization duration.

The VPP system can be used to investigate various fueling strategies such as AIS/TSC. The TSC case would consist of low temperature combustion (LTC) due to the first injection, followed by diffusion limited combustion with low oxygen concentration from the second injection. To achieve a single heat release case with multiple injections, either the EGR could be increased or the fuel amount in the first injection could be decreased to prevent LTC.

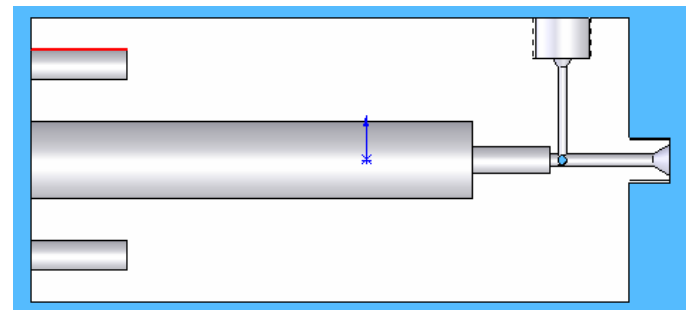
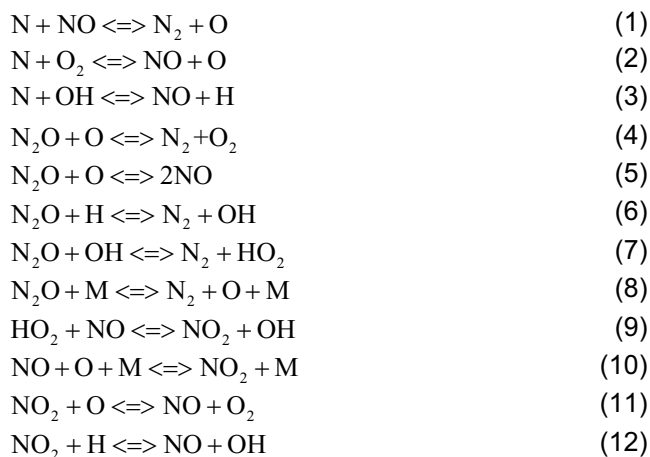


Figure 5 Cross-section view of adapter that houses the solenoid and pressure transducer.

COMPUTATIONAL MODEL

Computations are performed using the KIVA-3v release 2 code [23] with improvements to many physical and chemistry models developed at the ERC [24-28]. The KIVA-3v code is coupled with the CHEMKIN II solver for detailed chemistry calculations. A 39 species and 131 reaction mechanism for n-heptane [29] is used to simulate diesel fuel chemistry. Soot is predicted using a phenomenological soot model [30] based on the approach of Hiroyasu [31]. The soot model used in the present study uses acetylene as an inception species, which allows the soot model to be coupled to the chemistry solver through the addition of 13 reactions involving acetylene. NO_x emissions are predicted using a reduced NO mechanism [5] consisting of 4 additional species and 12 reactions. The reduced NO mechanism is based on the Gas Research Institute (GRI) NO mechanism [32]. The reduced NO mechanism described by Sun [5] is shown below:



Because detailed chemistry calculations are often time consuming, the Adaptive Multi-grid Chemistry (AMC) model of Shi et al. [33] is employed. The AMC model reduces the number of cells that need to be computed by the chemistry solver by grouping cells with similar gas properties for chemistry calculations and remapping the results to the original mesh. Previous investigations by Shi et al. [34] showed computational times were reduced by more than a factor of three while maintaining an accurate prediction of the emissions results.

The spray model employed in this study uses the Lagrangian-Drop and Eulerian-Fluid (LDEF) approach. Because a detailed chemistry model is used, it is desirable to use a relatively coarse computational mesh; however, severe grid size dependency has been observed in the LDEF spray models. This problem is most severe in the near nozzle region where the droplets are very close together and occupy only a small portion of the Eulerian mesh. Abraham [35] showed that accurate modeling of the near nozzle region required grid resolution on the order of the orifice diameter. However, it is not feasible from a computational time standpoint to solve engine problems on such a fine mesh. Furthermore, a fundamental assumption of the LDEF approach is that the volume fraction of droplets in each cell is small, that is., the void fraction is near one. Thus, if the mesh size is overly refined (i.e., mesh size equal to the droplet size) this assumption may be violated. In order to reduce the grid size dependency of the LDEF spray model and allow accurate spray simulation on a relatively coarse grid, the Gasjet model of Abani et al. [25] is employed to model the relative velocity between the droplets and gas phase in the near nozzle region. Their approach assumes that the relative velocity of between a droplet and the gas phase is equal to that between the droplet and a turbulent gas-jet with the same mass and momentum of that of the injected fuel. This approach imposes an axial component for the gas phase velocity as a function of position from the nozzle, which is used in the droplet acceleration equation given by:

$$\frac{dU}{dt} = \frac{3}{8} C_D \frac{\rho_g}{\rho_l r_d} |U - V_{gas}| (U - V_{gas}) \quad (13)$$

where U is the droplet velocity vector, C_D is the droplet drag coefficient which is a function of Reynolds number, ρ_g and ρ_l are the gas and liquid phase densities respectively, r_d is the droplet radius, and V_{gas} is the gas phase velocity vector given as $V_{gas} = (V_x, V_y, V_z)$ where V_y and V_z are the perpendicular to the spray axis and obtained from the Eulerian gas phase solution and V_x , the axial component of the gas phase velocity, is found from gas-jet theory [36] as:

$$V_x = U_{inj} \min \left[1, \frac{3d_{noz} \sqrt{\frac{\rho_l}{\rho_g}}}{K_{entr} x} \left(1 + \frac{12r^2}{K_{entr}^2 x^2} \right)^{-2} \right] \quad (14)$$

where U_{inj} is the injection velocity, d_{noz} is the nozzle diameter, K_{entr} is a model constant taken to be 0.7 as suggested by Abani et al. [24], x is the position downstream of the nozzle on the spray axis, and r is the radial position.

Droplet breakup is modeled using the hybrid Kelvin Helmholtz (KH) – Rayleigh Taylor (RT) model described by Beale and Reitz [26]. The droplet collision model is based on O'Rourke's model; however, a radius of influence method is used to determine the possible collision partners to further reduce mesh dependency [37]. In addition, the collision model was expanded by Munnannur [38] to include a more comprehensive range of collision outcomes. The current implementation of the droplet collision model considers bounce, coalescence, and fragmenting and non-fragmenting separations. Droplet interactions with the wall are considered through a wall film submodel [39, 40], which includes the effects associated with splash, film spreading, and motion due to inertia.

Parameter optimization is performed using a multi-objective genetic algorithm (MOGA). A genetic algorithm (GA) is a search technique inspired by the Darwinian idea of "survival of the fittest" [41]. The GA technique can be summarized as follows:

1. An initial population is generated by randomly assigned values for each optimization parameter to a specified number of optimization citizens.
2. The "fitness" of each citizen is evaluated by performing experiments or simulations.
3. Citizens with high "fitness" are allowed to reproduce resulting in a new generation of citizens. At this point, changes in a portion of the population may be introduced via mutations.
4. Successive generations are performed until the goals of the optimization are attained.

In the present work, a multi-objective approach is favored over the use of a merit function based single-objective optimization scheme in order to generate a set of solutions (i.e., Pareto optimal solutions) from which

the engine designer may choose an in-cylinder strategy that best matches the available aftertreatment system. A solution is defined as Pareto optimal if there are no other solutions which better minimize all objectives. This thought is shown graphically by Shi et al. [42] (Figure 6) for a two objective minimization problem. It is observed that no solutions outperform A-D in the task of minimizing both objectives simultaneously. In other words, points A-D are not dominated by any of the other solutions, E-H; thus, these non-dominated solutions are termed Pareto optimal. The set of non-dominated solutions makes up a Pareto front, as shown in Figure 6.

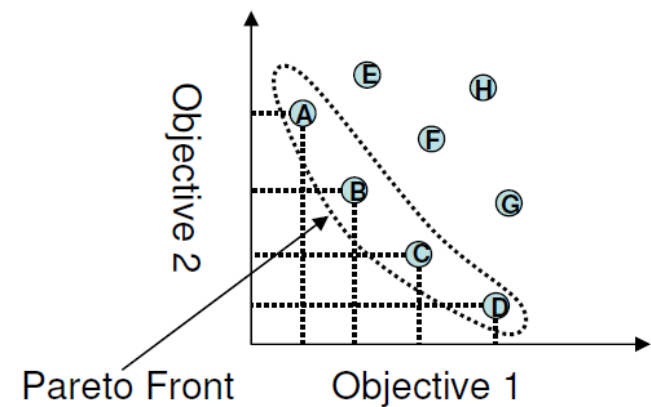


Figure 6 Illustration of the concept of Pareto optimality. [42]

Shi [42] has shown that the Nondominated Sorting Genetic Algorithm (NSGAI) proposed by Deb et al. [43] performs well in engine optimizations; therefore, NSGAI is selected as the optimization tool for this study

The computational grid used in this study is a 51.4° sector mesh which represents a single nozzle hole of the 7-hole Bosch injector used in the engine. Abani et al. [24] and Shi et al. [34] have shown the improved spray models used in this study yield similar results on both fine and coarse grids. Therefore, in order to further reduce computational expense, the coarse mesh of Figure 7 is used. The selected cases are then validated on a fine grid, see the appendix.

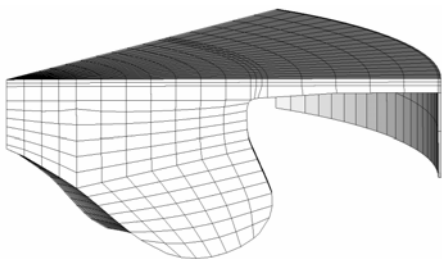


Figure 7 Computational grid with crevice volume, shown at TDC. The grid is a 51.4 degree sector consisting of 11,000 cells at BDC.

MODEL VALIDATION

This section consists of two parts. In the first part, model predictions are compared to the single injection PCCI experiments of Opat et al. [4]. In the second part, model predictions are compared to engine experiments using the VPP injection system and a multiple injection strategy.

SINGLE INJECTION PCCI COMBUSTION MODEL VALIDATION – Engine experiments performed by Opat et al. [4] are used for computational model validation. The experiments were conducted using a single injection PCCI approach at the same operating condition investigated in the present study (5.5 bar nominal IMEP and 2000 rev/min). To achieve PCCI combustion, Opat utilized an early injection to attain adequate mixing prior to the start of combustion and a high EGR rate (~67%) to suppress in-cylinder temperatures.

Table 4 Operating conditions for model validation.

Engine speed	2000 rpm
IMEP	5.5 bar
Fuel flow rate	0.895 kg/hr
Inlet oxygen concentration	9.5 %
Average equivalence ratio	0.90
IVC temperature	350 K
IVC pressure.....	1.91 bar

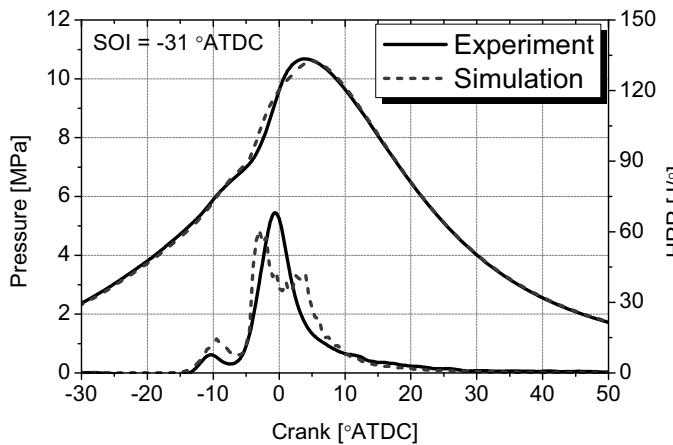


Figure 8 Comparisons of computed and measured cylinder pressure and heat release rate. The start of injection timing was 31 degrees before TDC.

Model validation is performed over a start of injection (SOI) sweep from 34.2 to 16.2 deg before TDC. Figure 8 shows the cylinder pressure and heat release rate comparison from a representative case with an SOI of 31 degrees before TDC. The locations of the cool flame and main heat release are predicted very accurately; however, the simulation predicts two peaks on the high temperature heat release, which is not observed in the experiments. The two-peak behavior is explained by