1 Multi-Level Computational Exploration of Advanced Combustion

2 Engine Operating Strategies

3 David Vuilleumier ^{a *}, Ivan Taritas ^b, Benjamin Wolk ^a, Darko Kozarac ^b, Samveg
4 Saxena ^c, Robert W. Dibble ^a

5 ^a *Department of Mechanical Engineering, University of California, 6141 Etcheverry Hall, Berkeley, CA*
6 *94720-1740, USA*

7 ^b *Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, Ivana Lucica 5, 10000*
8 *Zagreb, Croatia*

9 ^c *Energy Technologies Area, Lawrence Berkeley National Laboratory, 1 Cyclotron Road, MS 90R1121,*
10 *Berkeley, CA 94720, USA*

11 *Corresponding author. Tel: +001-916-396-6699; fax: none.

12 E-mail address: david.vuilleumier@berkeley.edu

13 Abstract

14 Advanced combustion engine (ACE) research is typically carried out on single-cylinder research engines.
15 These engines are designed to tightly control fueling and conditions at intake valve closure (IVC) and to
16 precisely measure in-cylinder conditions and emissions. However, to be able to measure and control engine
17 operation so precisely, these research engines typically do not feature intake and exhaust tracts that
18 resemble those in production engines, specifically in regards to turbomachinery, heat exchangers, and
19 exhaust gas recirculation (EGR) systems. For this reason, these research engines are effective for
20 understanding in-cylinder combustion parameters such as heat release rate, burn duration, combustion
21 efficiency, pollutant formation, and exhaust valve opening (EVO) conditions.

22 This paper applies high fidelity simulations to determine the feasibility of achieving a chosen single
23 cylinder engine operating point on a production type homogeneous charge compression ignition (HCCI)
24 engine, using a partial fuel stratification (PFS) strategy. To accomplish this, a Converge 3 dimensional (3D)
25 – computational fluid dynamics (CFD) model of the experimental combustion chamber and intake and
26 exhaust runners was created to simulate the experimental engine. This model was used to simulate an
27 operating point achieved experimentally, as well as to determine the sensitivity of the operating point to

variations in intake pressure, intake temperature, injection timing, injected mass, and EGR fraction. The results from these simulations were fed into a 1-dimensional engine simulation created in AVL Boost, featuring production-type intake and exhaust systems, including turbomachinery and heat exchangers necessary to create the required IVC conditions. This full engine simulation was used to assess the cycle efficiency of the engine at the experimental operating condition, and to assess whether changes to this operating point in intake temperature, intake pressure, direct injection timing, or fueling are beneficial to the cycle efficiency and engine-out emissions. In addition, the sensitivity of promising engine operating points to injection timing and injection mass are determined to evaluate the potential stability of these operating points.

Keywords: Homogeneous Charge Compression Ignition; Partial Fuel Stratification; Gasoline Compression Ignition; Low Temperature Combustion; Computational Fluid Dynamics; cycle-simulation

1. Introduction

The development of an engine operating strategy requires careful balancing of many simultaneously interacting in-cylinder fundamental phenomena that determine overall efficiency, emissions, stability, and noise-vibration-harshness characteristics. In the case of low temperature combustion (LTC) engines, the focus of the present study, important in-cylinder phenomena include chemical kinetics that determine heat release rates, spatially- and temporally-varying in-cylinder charge conditions, heat transfer, fuel vaporization, and emissions formation, as has been demonstrated by Saxena in [1]. These fundamental phenomena can be affected by several controllable parameters, including fuel type, port and direct injection strategy (including port fuel injection - PFI/direct injection - DI - injection fractions, DI timing, DI duration), intake temperature and pressure, equivalence ratio, EGR dilution, and engine speed. This large number of controllable parameters makes it difficult to identify optimal engine operating conditions

across the entire load-speed operating range. Furthermore, in-cylinder operation must be managed to ensure that critical limits are not exceeded, including ringing limits, misfire limits, and stability and cyclic variability limits, while maintaining acceptable overall engine operating efficiency and emissions characteristics [1]. The parameter space of operating variables and interacting fundamental phenomena grows even larger when considering the entire engine system, which includes the aforementioned in-cylinder phenomena, but also the intake and exhaust manifolds, turbocharger, intercooler, and EGR system.

This study aims to predict performance of a gasoline fuelled homogeneous charge compression ignition (HCCI) engine which employs a partial fuel stratification (PFS) operating strategy. Recent studies have demonstrated PFS as a promising operating strategy that can extend the operating regime of HCCI engines without exceeding the aforementioned operating limits. Dec et al. [2] found that at boosted conditions, PFS operating strategy is effective in increasing thermal efficiency; they increased peak thermal efficiency from 47.8% for premixed fueling to 48.4% with PFS strategy. In [3] the authors found that PFS enables staged combustion with reduced pressure rise rate, and therefore offers a potential to increase the high load range of HCCI engine operation. IMEP_g increased from 537 to 597 kPa, with NO_x emissions under US 2010 standards. In [4], Dec et al. found PFS strategy to shift ringing/stability limits and thus allows the maximum IMEP_g increase from 11.7 to 13 bar. Yang et al. [5] showed that fuel composition significantly affected the ability to carry out a PFS strategy, and the same authors also demonstrated [6] the influence of operating conditions, namely engine speed, intake temperature, and fuel composition on the efficacy of a PFS strategy. In [7], Yang

et al. found that high load extension in a gasoline HCCI engine that uses PFS strategy is limited by the trade-off between CO (retarded operation) and NO_x (advanced operation) emissions. In [8, 9] the authors used detailed 3-D CFD analysis to study and better explain the in-cylinder phenomena that occurs in HCCI engine that uses a PFS strategy. They found in [8] that occurrence of single- or multi-stage combustion depends on the temperature distribution which is influenced by low temperature heat release (LTHR) and evaporative cooling from the liquid fuel spray. In [9] it was found that at 1 bar intake pressure since the premixed portion of the charge ignites before rich regions, the sequential autoignition occurs too fast and there is no reduction in the maximum pressure rise rate compared to HCCI operation. On the other hand, at 2 bar intake pressure the sequential autoignition occurs over a longer period as the premixed portion auto-ignites last, which reduces maximum pressure rise rate compared to HCCI operation.

Given the range of engine operating variables that affect PFS performance, prior studies have focused on in-cylinder operating variables without giving adequate consideration to the engine system. Moreover, the majority of the experimental studies focused solely on extending the high load operation limit, while the numerical ones focused on understanding the fundamental aspect and impact of a PFS strategy on HCCI combustion process. This paper presents a numerical study that focuses on mid-load engine operation, ~10.5 bar indicated mean effective pressure (IMEP), at 1,200 RPM. The present study bridges the gap between previous experimental and numerical studies by investigating an HCCI engine that employs a PFS strategy using a coupling of a detailed 3D-CFD and chemical kinetic combustion chamber model [10-13] with a 1D multi-cylinder engine system model [14, 15]. This approach builds off the approach which has been developed

by Babajimopoulos et al. [10], Embouazza et al. [11], Wang et al. [12], and Kong et al [13]. In [10], the coupling of CFD with chemical kinetics solvers is shown to be effective in capturing the effects of interacting fluid dynamics and chemical kinetics, in this case simulating a premixed charge compression ignition engine. In [11], tabulated chemistry is coupled to CFD in order to capture the interactions between chemistry and fluid dynamics. Wang et al. [12] coupled a CFD simulation to a complex 89 species gasoline mechanism to simulate directly-injected gasoline HCCI combustion, while Kong et al. [13] carried out a similar coupling between KIVA-3 and Chemkin, also to study directly injected gasoline HCCI combustion. In [14], Ogink et al. couple the AVL Boost engine simulation code, which is described in the AVL Boost Theory Manual [15], to SENKIN routines for calculating chemical kinetics; this combination is used to simulate gasoline HCCI combustion. The techniques which were pioneered in [11 – 14] are used in this work to investigate the behavior of a PFS operating strategy.

3D-CFD simulations incorporating chemical kinetic combustion models are an established method to study kinetically controlled combustion engines, demonstrated by both Zhou et al. [16] and Li et al. [17] in simulating Reactivity Controlled Compression Ignition combustion, with the former investigating biodiesel and methanol blends, and the latter investigating gasoline and biodiesel blends. In the first part of the present study, a detailed 3D-CFD simulation was used in performing a detailed parameter sweep to identify the PFS operating parameters leading to optimal engine performance. Operating parameters varied in this numerical study are intake pressure and temperature, DI timing, DI mass fraction, equivalence ratio and EGR fraction. By sweeping through a narrow range of controllable parameters, the sensitivity of engine performance (including

efficiency, emissions, ringing intensity (RI), etc.) to many of the operating variables is quantified. After identifying feasible operating points, which resulted in high indicated efficiency and low exhaust gas emissions, a full model cycle-simulation, featuring a detail description of a multi-cylinder engine charging and cooling elements, and entire intake and exhaust flow path, was run in order to assess the overall system efficiency and operating range in terms of intake temperature and engine speed.

2. Methods

2.1. Combustion Model

3D-CFD simulations in this work were performed using CONVERGE [18], a commercially available CFD code. The grid used for all CFD simulations is representative of the Sandia HCCI engine [4], as shown in **Figure 1**. CONVERGE employs a cut-cell Cartesian method for grid generation and generates a grid at each time step given the position of the boundary surfaces. The previous time step solution is mapped onto the new grid before solving for the next time step. This procedure naturally permits the use of moving surfaces, such as the piston and intake/exhaust valves. A uniform grid spacing of 2 mm was used, as this was the finest grid resolution that could be computationally afforded. This resulted in 140,881 in-cylinder cells at bottom dead center (BDC) and 13,859 in-cylinder cells at top dead center (TDC). A non-uniform, adaptive grid may be required to resolve a flame surface. However, a non-uniform grid is not required in this work as there is only sequential auto-ignition without flame propagation.

A schematic of the computational grid is shown in **Figure 1** and engine specifications are listed in **Table 1** (values in parentheses indicate experimental values that differ from the associated values used in the simulations). The geometric compression ratio (CR) of the computational mesh is 13.65:1, which is slightly lower than the 14:1 used in the experiment, in order to match the motored pressure trace as crevices and blow down are not meshed directly.

The engine has two intake ports, one port with a helical geometry and one port with a tangential geometry. The valve lift profiles are set using experimentally measured valve lift values. The piston has a deep central bowl to prevent piston wetting at late direct-injection timings. The head gasket crevice is included in the computational mesh and a piston ring crevice model is used rather than direct meshing of the piston ring crevices. Dimensional information about the piston rings and piston ring cutouts is used as input to the piston ring crevice model.

Simulations are run from 360 ° bTDC to 122 ° aTDC. This encompasses the full intake stroke to part way through the expansion, after combustion has completed and to the point at which the exhaust valve opens. The intake stroke is computed to create realistic thermal stratification and in-cylinder velocity profiles in the premixed intake gas. Chemistry is computed only for cells with temperatures in excess of 600 K and is not computed in the intake or exhaust ports while the respective valves are closed. This helps to reduce the computational cost of the simulations, as the time for computing chemistry accounts for a large fraction of the total computational effort.

Figure 1: Computational surface of Sandia HCCI engine at top dead center. The in-cylinder grid is shown with a grid spacing of 2 mm (the full grid includes the intake and exhaust ports).

Table 1. Sandia HCCI engine specifications

Displacement	0.981 L
Bore	102 mm
Stroke	120 mm
Connecting rod	192 mm
Geometric CR	13.65:1 (14:1)
Number of valves	4
IVO	360 °BTDC
IVC	158 °BTDC
EVO	122 °ATDC
EVC	368 °ATDC
Engine speed	1,200 rpm
Intake pressure	1.9 – 2.1 bar
Coolant temperature	100 °C

The gasoline used in the experiments by Dec et al. [4] and considered in this work is RD387 research grade gasoline, which contains a wide range of molecular components. Although the broad molecular composition has been quantified (i.e. amount of alkanes,

aromatics, and olefins), specifying the exact amount of each molecule in RD387 is not feasible. As a result, a surrogate containing a small number of pure components has been formulated by Mehl et al. [19] to mimic major properties of RD387. The targeted properties are the broad molecular composition, the H/C ratio, stoichiometric air-fuel ratio, and anti-knock index (AKI).

In this study, gasoline is modeled by the 4-component surrogate identified by Mehl et al. [19] comprised of 57% iso-octane (iC_8H_{18}), 16% n-heptane (nC_7H_{16}), 23% toluene ($C_6H_5CH_3$), and 4% 2-pentene (C_5H_{10-2}) by liquid volume. The chemistry is modeled using a 98-species reduced mechanism for the 4-component gasoline surrogate [20] including NO_x formation reactions and a well-mixed sub-grid chemistry model. A well-mixed sub-grid chemistry model indicates that the reaction rates are evaluated at the mean cell temperature (sub-grid temperature fluctuations are ignored). Despite this shortcoming, well-mixed sub-grid chemistry is common among RANS engine codes. Turbulence is modeled using the RNG k-epsilon turbulence model [21].

The fuel injector used in the experiment is an 8-hole gasoline direct injection (GDI) injector that has a hole diameter of 125 μm , included angle of 70° , and injection pressure of 120 bar. The injector is centrally mounted in the combustion chamber and oriented vertically downward. The simulated injection uses a Lagrangian spray model and a stochastic collision model (O'Rourke) with injected droplets following a Rosin-Rammler distribution [22] with a Sauter mean diameter (SMD) of 125 μm , 180 m/s injection velocity, and initial droplet temperature of 300 K. The liquid fuel droplet breakup is modeled using the Kelvin-Helmholtz Rayleigh-Taylor (KHRT) droplet breakup model [23]. The KHRT breakup model evaluates the growth rates of the KH and RT instabilities

to determine if a droplet breaks up due to one of these mechanisms during a computational time step.

The boundary conditions used for solid surfaces are constant temperature. Surface temperatures are set to the coolant temperature, except for the piston and head, which are set to the experimentally measured head temperature. For the intake ports, the initial conditions and the intake inflow compositions are set to the composition of the premixed charge. For the in-cylinder region and exhaust ports, the initial composition is the complete combustion products (CO_2 , H_2O , O_2 , N_2) of the premixed gas (not including additional DI fuel). The initial temperature of the intake region is the intake temperature, and the initial in-cylinder and exhaust temperatures are the experimental exhaust temperatures. The model used in this paper was previously validated and published in [9].

The agreement between experimental and simulated conditions can be seen in **Figure 2**, **Figure 3**, and **Figure 4**. **Figure 2** shows the agreement of in-cylinder pressure between the simulation and the experiment for four different injection strategies at 2 bar intake pressure. **Figure 3** and **Figure 4** show the simulated and experimental rates of heat release for the same four operating points. It can be seen from the three figures that generally good agreement exists between the model and the experiment. A more detailed discussion of the operating points and their specific agreements with experiments can be found in [9]. For this paper, the model was used nearly unaltered from that presented in [9], except for the use of the 98 species kinetic mechanism rather than the 96 species mechanism used in [9]. These mechanisms were both reduced from the same detailed mechanism, and have nearly identical behavior, with the primary difference between

215 them being the addition of the species NO and NO₂ to the 98 species mechanism, for the
 216 calculation of the concentrations of these pollutants.

217
 218 **Figure 2: Comparison of four experimental and simulated pressure traces at 2 bar intake pressure,**
 219 **for fully premixed (PM) and partially stratified (13% DI) cases at two different combustion phasings**
 220 **[9].**

221
 222 **Figure 3: Comparison of experimental and simulated rates of heat release at 2 bar intake pressure,**
 223 **for fully premixed (PM) and partially stratified (13% DI) cases [9].**

Figure 4: Comparisons between experimental and simulated rates of heat release at 2 bar intake pressure for two different injection timings for partially stratified (13% DI) cases [9].

2.2. Cycle-simulation

The simulation of the full engine performance was performed by a cycle-simulation model made in the AVL Boost. In order to obtain simulation results which accurately reflect the operation of the engine, the entire engine layout featuring four cylinders and a detailed description of the entire engine flow path was made, as seen in **Figure 5**. More detailed information on this model can be found in [24].

In the full engine model, combustion was calculated with a table heat release model which enables the calculation of many engine cycles in a reasonable time. These tables were created according to the rate of heat release profiles obtained from the detailed 3D-CFD simulation. The cylinder was modeled according to the cylinder geometry data, which is given in **Table 1**. In the AVL Boost, the effect of two intake and two exhaust valves is taken into account by setting the appropriate value for the scaling factor for the effective intake & exhaust flow area [15]. Heat transfer in the combustion chamber was calculated with Woschni's model. More on the heat transfer calculation in the

241 combustion chamber, as well as on the heat transfer and friction calculation in the pipe
 242 elements can be found in [15].

Figure 5: Full engine simulation model.

245 Variable geometry turbocharger (shown in **Figure 5** as TC1) performance was calculated
 246 with the full model charger calculation, which uses the turbine and compressor static
 247 performance maps that were obtained from the manufacturer. As these maps present just
 248 a narrow operating range of the charging device, extrapolation of the maps was
 249 performed by a BOOST Turbocharger Tool [15]. Vane position was set so that the
 250 appropriate pressure at the IVC is obtained. This was controlled automatically by a PID
 251 controller (shown in **Figure 5** as PID 1).

Besides the charging element, the full engine model features an intake air cooler (shown in **Figure 5** as CO1) and a system to bypass it. It also features a high pressure EGR loop with an EGR cooler (shown in **Figure 5** as CO2) and a system to bypass it. The intake air cooler is an air to air cooler, while the EGR cooler is an air to water cooler. Cooling performance was calculated with the target efficiency model, which requires the input of the cooler efficiency and the coolant temperature [15]. For the air to air cooler, the cooling efficiency was set to 65% [25], and the coolant temperature was set to 25°C, which corresponds to the temperature of the outside air. The air to water cooler efficiency was set to 90% [25], while its coolant temperature was set to 50°C. The cooler outlet temperature was controlled by the PID regulators (shown in **Figure 5** as PID 2 and PID 4, respectively), which control the flow through the cooler bypass, thus regulating the cooler outlet temperature. Cooler outlet temperature was set so that temperature at IVC matches the one obtained with the detailed 3D-CFD simulation. If no cooling is needed, cooler flow coefficients are set to zero, while the bypass restrictors are kept fully open, which means that the entire charge flows through the bypass. The external EGR loop is also equipped with a controllable valve (shown in **Figure 5** as R1) that enables the control of the amount of combustion products that are recirculated back to the intake manifold. The amount of EGR in the intake was controlled by a PID controller (shown in **Figure 5** as PID 6), and was set so that it matches the amount that was simulated with a detailed 3D-CFD model.

As classical species transport was used, a 4-component surrogate fuel (details can be found in the previous section) was created with a BOOST Gas Properties Tool [15]. The same injection strategy as in the 3D-CFD model was used, where a portion of the fuel

was injected into the intake port, while the remaining fuel was injected into the cylinder during the compression stroke. Fuel evaporation was modeled with a standard evaporation model available in AVL Boost [15].

The first set of the full engine simulations was performed to analyze the entire engine performance for the three most interesting points obtained with the 3D-CFD simulation. Operating parameters of these points are given in the next section.

The second set of the full engine simulations was performed to analyze the operability limit in terms of the simulation-boundary intake temperature (SB1 in **Figure 5**), for the most interesting points simulated with a detailed 3D-CFD model. These simulations were performed in order to determine the lowest and the highest intake temperature that would still be able to achieve the desired intake valve closure (IVC) temperature. The lowest intake temperature case presumes operation with uncooled EGR and intake air charge, while the highest intake temperature presumes the operation with the maximum cooling capacity. Hence, six simulations were run in this set, and the details regarding the operating parameters of these points are given in the next section.

Finally, the third set of the full-engine simulations was performed to analyze the engine's operability range in terms of engine speed (turbocharger matching). Eight simulations were run in this set, and details regarding the operating parameters of these points are given in the next section. Since these operation points were not simulated with a 3D-CFD model, due to lack of validation data, reference rates of heat release were not available. Hence, the combustion was calculated with a Vibe model, whose parameters were tuned so that the ROHR profile has a bell like shape; start of combustion was set to 4.9 degCA

ATDC and combustion duration was set to 6.3 degCA. These values were chosen according to the values obtained in the base operating point at 1,200 rpm. Moreover, as there is no data on the fuel mass for these operating points, fueling was set so that the overall charge to fuel ratio (C/F ratio) was approximately 34.5. This C/F ratio was chosen so that it equals the one that was simulated in the base operating point at the engine speed of 1,200 rpm. Since there is no data on the EGR fractions for these operating points, simulated EGR fractions were set to 28.6% for the first sweep, and 20% for the second sweep. Also, as there was no data on the IVC pressure and temperature for these operating points, the desired values were set according to the base operating point at 1,200 rpm (2.01 bar and 384 K).

3. Results and Discussion

3.1. Combustion Model Results

Table 2. Key engine operation variables.

Variable	Unit	Range
Intake Pressure	Bar	1.9 - 2.1
Intake Temperature	Kelvin	336 - 377
Direct Injection Timing	CAD bTDC	25 - 130
Direct Injection Fraction	-	0.08 - 0.19
Charge-Mass Equivalence Ratio	-	0.4 - 0.45
Exhaust Gas Recirculation Fraction	-	0.2 - 0.4

310

311 **Table 3. Detailed intake parameters and results for five engine operating conditions. Note that points**

312 **3 and 4 are only used for Ringing Analysis.**

Variable	Unit	Base Case	P1	P2	P3	P4
Intake Temperature	K	359	349	362	372	367
Intake Pressure	Bar	2.00	1.98	1.90	2.10	1.90
Injection Timing	dADTC	-75	-100	-100	-100	-75
EGR	%	29	14	22	29	22
Engine Speed	RPM	1,200	1,200	1,200	1,200	1,200
IMEP	bar	10.6	10.63	10.6	10.44	9.87
Direct Injected Fraction	-	0.13	0.08	0.20	0.08	0.14
Equivalence Ratio	-	0.43	0.41	0.44	0.4	0.41
UHC	g/kW-hr	0.3	0.27	0.31	0.33	0.34
O2	g/kW-hr	456	730	510	518	586
CO	g/kW-hr	1.68	1.65	1.67	1.82	1.8
CO2	g/kW-hr	968	760	860	982	874
NOx	g/kW-hr	0.98	0.14	2.72	0.14	1.82
Efficiency	-	43.0	43.4	43.2	43.3	43.5
CA10	Degrees	6.8	9	7.2	6.8	5.8
CA50	Degrees	11.1	12.7	10.2	9.6	9.5
CA90	Degrees	13.6	15	12.8	11.3	11.6
Combustion Duration	Degrees	8.3	9.2	6.9	6.3	7.2
Ringing Intensity	MW/m ²	3.9	4.6	3.7	11.6	6.3
Maximum Cylinder Pressure	Bar	112	109	113	123	112
Maximum Cylinder Temperature	K	2,538	2,190	2,572	2,230	2,583
Maximum Average Cylinder Temperature	K	1,928	1,901	1,983	1,921	1,943

The combustion modeling portion of this study focused on conducting a parametric variable sweep around an experimentally-validated simulated engine operating point. Important engine operating parameters, listed in **Table 2**, were varied in an effort to assess the performance of the original operating point against similar operating points that would provide similar load levels. These operating points characterize mid-load operation (IMEP of approximately 10.5 bar) at low engine speed (1,200 RPM). In total, 212 simulations were run, using approximately 150,000 CPU hours.

A primary focus of this study is the closed-cycle efficiency, as this will have a significant influence on efficiency and exhaust gas emissions. In **Figure 6** and **Figure 7**, cycle efficiency is plotted against combustion phasing and combustion duration, respectively. In these figures, three possible operating points are indicated by special markers, while all other operating points are indicated by round black markers. The specific conditions relating to these three operating points, as well as two others that will be discussed later, can be found in **Table 3**. It can be seen from **Figure 6** and **Figure 7** that the peak efficiencies indicated by the simulations range between 43 – 44%. Also, it can be seen in **Figure 6** that a relatively wide range of combustion phasings yield high indicated efficiencies, with phasings ranging from the earliest tested (4 CAD aTDC) to approximately 14 CAD aTDC yielding efficiencies above 42%. This trend of efficiencies is driven by the effects of heat transfer, thermodynamics, and combustion efficiency. In this case, early combustion phasings yielded high combustion efficiency and high thermodynamic efficiency, while not incurring enough of a heat transfer penalty to reduce efficiency. As will be discussed later, earlier combustion phasing was not tested due to limits placed on Ringing Intensity. At later combustion phasings, there is a

thermodynamic penalty on efficiency, as a portion of the expansion stroke is left effectively unused, and this accounts for the majority of the drop in overall efficiency at later combustion phasings. At extremely late combustion phasings, a combustion efficiency penalty may occur as well, though as can be seen in **Figure 8** and **Figure 9**, unburned hydrocarbon (UHC) and carbon monoxide (CO) emissions are similar under most conditions, as cases with partial or full misfires have been omitted from these results as they are unfeasible for engine operation. In the PFS combustion system, and at the mid-load conditions that are studied here, the UHC and CO emissions are primarily from charge that is present in the crevice and near-wall regions during combustion. This can be seen most clearly in **Figure 8**; as combustion phasing is delayed, a lower fraction of the charge is present in the crevice and near-wall regions (due to combustion occurring later in the expansion stroke, resulting in a higher combustion chamber volume), resulting in slightly decreased carbon monoxide emissions.

Figure 6: Indicated efficiency plotted against combustion phasing for all operating conditions evaluated using the combustion model.

Figure 7: Indicated efficiency plotted against combustion duration for all tested operating conditions.

Figure 8: Carbon Monoxide (CO) emissions as a function of combustion phasing.

Figure 9: Unburned Hydrocarbons (UHC) emissions as a function of combustion phasing.

The sweep of engine operating conditions identified two potential engine operating points in addition to the base case from which the permutations were based on. Point 1 improves upon the base case in the main metrics used for this study: indicated efficiency and NO_x emissions, while maintaining acceptable RI. Thus, Point 1 would be a preferred operating point to the Base Case under most circumstances. Point 2, on the other hand, reduced RI and increased control over combustion phasing at the expense of NO_x production and indicated efficiency. Therefore, this operation point may be favored in an environment in which Point 1 cannot be achieved due to the lower required intake temperature or in which RI must be reduced. From **Figure 6** and **Figure 7** it can be seen that points exist with greater efficiencies than the three points that were chosen. However, it will be seen in figures presented later in this paper that these points were unsuitable either from a standpoint of RI, or NO_x emissions, or both.

NO_x emissions are a particular problem for lean burn engines that cannot make use of a 3-way catalyst. As seen in **Figure 10**, NO_x emissions were highly dependent on combustion phasing, and fuel inhomogeneity (in this case a function of DI fractions and injection timings). This trend is easily explained by the thermal dependence of NO_x via the Zeldovich mechanism [26]. As seen in **Figure 10**, the NO_x emissions correspond closely with the peak cell temperature encountered by each operating condition. The peak temperature, in turn, was influenced by both the mixture preparation and the combustion phasing. In general, the higher DI fractions had significantly higher NO_x than the lower DI fractions for a given combustion phasing. This is caused by a larger amount of fuel mass close to stoichiometric conditions at the point of auto-ignition, which results in higher localized post-combustion temperatures. Therefore, to keep NO_x emissions low

while using a more aggressive DI strategy (possibly as a control strategy), a later combustion phasing is required. However, combustion phasings late enough to ensure low NO_x also begin to reduce the indicated efficiency of the high pressure cycle, and push the operating point closer to the misfire limit. Therefore, reducing the DI fraction seems more favorable for this mid-load condition.

Figure 10: Plot of NOx emissions against combustion phasing for all operating points.

Figure 11: Plot of NOx emissions against the maximum temperature reached in the cylinder during the complete engine cycle.

A critical index for safe engine operation of advanced combustion engines (ACE's) is the RI. This parameter quantifies the degree of engine-

damaging pressure oscillations that are occurring in the cylinder due to combustion. **Figure 12** shows the level of ringing that occurs in the simulated engine operating points. In this figure a threshold is marked at 5 MW/m^2 , which has become the accepted maximum level of RI for production engines [27]. From **Figure 12** it can be seen that RI is closely related to combustion phasing, with later combustion phasings reducing the RI. Late combustion phasing helps reduce ringing because as phasing is delayed, the piston is more rapidly increasing the volume of the combustion chamber as combustion occurs, which in turn reduces the pressure rise rate and maximum temperature in the cylinder. **Figure 12** also shows that for a given combustion phasing, RI can vary on the order of 50%, indicating that factors other than combustion phasing can have a large influence on ringing.

Figure 12: Plot of RI against combustion phasing for all operating points. An upper limit of ringing for this engine is shown by the red line.

Figure 13 plots the rate of heat release for four cases: the two cases of interest, and two additional cases, Point 3 and Point 4. Point 3 and Point 4 have nearly identical CA50's, of $\sim 9 \text{ CAD aTDC}$, however their RI's are 11.6 and 6.3 MW/m^2 , respectively. The cause of these radically different RI's can be found in their ROHR profiles, as seen in **Figure 13**. These figures show the different manners in which Points 3 & 4 burn. Point 4 has a

412 significantly lower peak ROHR; this peak ROHR is caused by the ignition of the nearly-
 413 homogeneous bulk-charge. Point 4 has a smaller bulk charge mass than Point 3. This is
 414 due to the greater fuel stratification found in Point 4, which is caused by a later direct
 415 injection which contains more fuel than the direct injection strategy employed in Point 3.
 416 As seen in **Figure 14**, the higher peak ROHR results in a higher pressure rise rate, which
 417 in turn raises the RI.

418
 419 **Figure 13: Rate of heat release profiles for five operating points.**

420
 421 **Figure 14: In-cylinder pressure traces for five operating points of interest.**

Finally, direct-injection timing and mass sweeps were conducted to assess the ability to use the directly-injected fuel to help control combustion phasing during engine operation.

Figure 15 and **Figure 16** show the results of these sweeps.

Figure 15: Combustion phasing (CA50) plotted against direct injection timing. In this sweep, the mass of both the aspirated and directly injected fuel is held constant for each case.

Figure 16: Combustion phasing (CA50) plotted against direct injection mass fraction.

In this sweep, the mass of the aspirated fuel is held constant while the mass of the directly injected fuel varies. It can be seen in **Figure 15**, that the Base Case and Point 2 both demonstrate significant control over combustion phasing with the injection timing sweep. However, Point 1 shows significantly reduced control over combustion phasing relative to the other points, as indicated by the difference in slope of the Point 1 dataset compared

to Points 2 and 3. This is likely due to the lower direct-injection mass fraction in Point 1, resulting in a smaller perturbation of the in-cylinder conditions at TDC, reducing the effect on combustion phasing. This effect is obviously nonlinear, as the Base Case and Point 2 show similar control over combustion phasing even though they have different direct-injection mass fractions. However, it can be seen in **Figure 16** that when the direct-injection mass fraction is altered, all three points have similar changes in CA50. It should be noted, however, that in the range tested, the timing sweep has a larger effect on CA50 while not altering the fuel energy level, though in the case of Point 1, the timing sweep could not alter the combustion phasing over a wide range.

3.2. Cycle Simulation Results – Complete Engine Performance

Complete engine performance was estimated for only three operating points (as seen in **Table 4**).

Table 4. Simulated key operating points.

Operating Point	Engine Speed (rpm)	Pressure at IVC (bar)	Temperature at IVC (K)	EGR (%)
Base	1,200	2.01	384	28.6
Point 1	1,200	1.99	371	20
Point 2	1,200	1.93	382	20

In the full engine performance calculation, combustion was calculated with a table heat release model, with tables that were created according to the rate of heat release profiles obtained from the detailed 3D-CFD simulation. **Figure 17** shows that there is a good fit between the pressure traces obtained in the 3D-CFD and Boost simulation.

452 The variable geometry turbocharger was chosen so that the compressor operates in the
 453 optimum area in the engine operating points of interest, as seen in **Figure 18**. The engine
 454 performance results obtained with the full engine simulation are presented in **Table 5** and
 455 **Table 6**.

456
 457 **Figure 17: Comparison between the CFD and Boost simulation**

458 From the simulation results, it can be seen that similar overall performance was obtained
 459 in all three operating points of interest. The highest output work was obtained in the base
 460 operating point, which comes as a combined result of the higher fueling rate and the
 461 overall system performance. Lower brake efficiency was obtained in operating Point 1,
 462 compared to the Base and Point 2 operating points. This is a result of higher pumping

losses, which is evident from a higher exhaust backpressure, which comes as a consequence of the lower turbine inlet temperature (**Table 6**).

Figure 18. Compressor performance map with the referenced operating points plotted in it.

Table 5. The engine performance obtained with the full engine simulation (η_E – the engine brake efficiency; p_{INT} – intake pressure; p_{EXH} – exhaust pressure)

Operating Point	BMEP (bar)	η_E (%)	p_{INT} (bar)	p_{EXH} (bar)
Base	9.57	38.96	1.94	2.45
Point 1	9.44	38.8	1.97	2.5
Point 2	9.52	38.96	1.91	2.32

Table 6. The system performance obtained with the full engine simulation ($P_{EGR, COOLER}$ – the calculated heat flux at EGR cooler; $P_{INTERCOOLER}$ – the calculated heat flux at intercooler)

Operating Point	$P_{EGR, COOLER}$ (W)	$P_{INTERCOOLER}$ (W)	T_{EXH} (K)	η_{COMP} (%)	η_{TC} (%)
Base	7,486	0	789	73.7	39
Point 1	6,447	934	777	72.9	37.8
Point 2	4,069	0	799	73	37.5

The overall turbocharger efficiency was quite high in all three operating points. The base operating point had the highest turbocharger efficiency, which is a result of the position of the turbine stator blade that the turbine was operated at. Point 1 and Point 2 were operated with more opened turbine stator blades which is a result of the lower EGR requirement and hence higher exhausts mass flow at the turbine. Slightly lower turbocharger efficiency in these operating points is a result of the lower turbine efficiency at this stator blade position. Although Point 2 has lower turbocharger efficiency compared to the Base point, this point has the lowest pumping losses, which comes as a result of the highest exhaust (turbine inlet) temperature, which enables the operation with lower exhaust (turbine inlet) pressure.

The highest cooling requirement was in the base operating point, as a result of the highest amount of EGR that had to be cooled. It can be seen that intercooler only had to be applied in the Point 1 operating point. This is a result of low EGR rate (20%) in the combination with a low IVC temperature requirement **Table 6**.

3.3. Cycle Simulation Results – Engine Operability Range

The low and high intake temperature range was analyzed with a table heat release model at the engine speed of 1,200 rpm. The operating points that were analyzed are specified in **Table 7**.

The engine could be operated in all three operating points at -70°C and 70°C. To obtain the desired IVC temperature at -70°C, the EGR cooler had to be engaged, which means that even lower intake temperature could be obtained. However, as such low temperatures are not expected, -70°C was set as a low temperature limit. Also, at 70°C the desired IVC

temperature was obtained without the intercooler operating at full capacity, which means that even higher intake temperature could be obtained. However, as such high temperatures are not expected, 70°C was set as a high temperature limit.

Table 7. Simulated operating points for high- and low-temperature limits.

Operating Point	Engine Speed (rpm)	Pressure at IVC (bar)	Temperature at IVC (K)	EGR (%)
Base – Low	1,200	2.01	384	28.6
Base – High	1,200	2.01	384	28.6
Point 1 – Low	1,200	1.99	371	20
Point 1 – High	1,200	1.99	371	20
Point 2 – Low	1,200	1.93	382	20
Point 2 – High	1,200	1.93	382	20

The operability limit in terms of the engine speed was analyzed with a Vibe model as 3D-CFD results were not available. These simulations were performed to test if the chosen turbocharger and coolers could accommodate the engine operation at various engine speeds. The tested engine speed range was from 800 rpm to 2,400 rpm, which represents the idle and top speed for a common heavy-duty engine. The analyzed operating points are specified in **Table 8**.

As can be seen from the compressor performance map, shown in **Figure 19**, the engine could be operated in this speed range with a few limitations. At 800 rpm, the desired level

of SHP pressure could not be obtained while achieving both the desired EGR rates, due to the combination of the low exhaust gas enthalpy and low exhaust gas mass flow at the turbine. As expected, higher intake pressure could be obtained with a 20% EGR rate due to higher mass flow at the turbine.

Table 8. Target conditions for simulated operating points over the engine speed range

Operating Point	Engine Speed (rpm)	Pressure at IVC (bar)	Temperature at IVC (K)	EGR (%)
Idle	800	2.01	384	30
Speed 1	1,600	2.01	384	30
Speed 2	2,000	2.01	384	30
Top Speed	2,400	2.01	384	30
Idle	800	2.01	384	20
Speed 1	1,600	2.01	384	20
Speed 2	2,000	2.01	384	20
Top Speed	2,400	2.01	384	20

Figure 19. Compressor performance map with the referent operating points plotted in it.

On the opposite side of the engine speed range, at 2,400 rpm, the compressor was operated in the low efficiency area which led to a considerable deterioration in the engine

breathing (**Table 9**), which can be seen through the increased exhaust backpressure. As can be seen, at 2,400 rpm the desired IVC pressure could not be obtained with a 20% EGR rate due to significantly deteriorated compressor efficiency.

Table 9. The engine performance obtained in the operability range simulation

Operating Point	Engine speed (rpm)	P_{INT} (bar)	P_{EXH} (bar)	EGR (%)	Power (kw)	Torque (N*m)
Idle	800	1.32	1.61	30	14.7	175.7
Speed 1	1600	2	2.58	30	50.3	300.5
Speed 2	2000	2	2.96	30	60.9	290.7
Top Speed	2400	1.98	3.68	30	61.8	245.7
Idle	800	1.48	1.85	20	16.7	199.8
Speed 1	1600	2	2.67	20	49.8	296.6
Speed 2	2000	1.98	3.42	20	56.4	271.7
Top Speed	2400	1.75	3.45	20	53.5	215.2

4. Conclusions

This study used a RANS 3D-CFD model of an experimental medium duty Homogeneous Charge Compression Ignition engine operating with a Partial Fuel Stratification strategy to optimize a mid-load engine operation point at both the combustion and system levels. It was found that relative to a baseline operating point, the closed-cycle engine operation was improved in terms of indicated efficiency (0.4%), NO_x emissions (0.8 g/kw*hr) at the expense of RI (0.7 MW/m²) by carefully tuning intake and direct injection parameters. A second operating point was identified which increased efficiency (0.2 %) and reduced RI (0.2 MW/m²) at the cost of NO_x emissions (1.7 g/kw*hr). In assessing the parametric sweep of operating conditions, it was found that the largest influence on combustion system performance came from the combustion phasing of the engine operating point, with late combustion phasing being required to reduce RI and reduce

NO_x production. However, excessively late combustion phasing reduced efficiency. The sweep of engine operating points indicated that if NO_x is to be kept at a low level with efficiency kept high, HCCI-like conditions are most desirable. However, these conditions reduce the ability to control combustion phasing with the direct injection. In the case of higher DI fractions, very late combustion phasing was required to keep NO_x low, which reduced efficiency.

Following the optimization of the in-cylinder engine operating point, a 0D/1D engine model was created to determine the feasibility and operability limits of the engine, in regards to turbomachinery and thermal regulation. Highlighting the importance of a system level approach, the most efficient point over the high-pressure cycle did not result in the highest overall efficiency of the engine at the system level. This was due to the changes to the turbine inlet conditions caused by the reduced use of EGR and lower intake pressure. By reducing EGR and intake pressure, the mass flow through the turbine increased, while the pressure ratio decreased, which moved the turbine into a lower efficiency range. The peak brake efficiency for the engine, when considering turbomachinery sized for operation of the base case between 800 RPM and 2,400 RPM, was found to be 39%. For this operating condition, intake temperatures in the range of -70 °C to 70 °C were found to permit the correct IVC conditions through charge and EGR cooling. These results indicate that the engine operating points are viable and competitive with traditional diesel operation, and are more efficient than current spark-ignition operating points at a similar load level. In addition, the move from HCCI to PFS offers the ability to implement additional control over the engine, through altering direct-injection timing or injected fuel mass, as investigated in this paper. These additional

control possibilities make PFS a more realistic combustion system than HCCI, while retaining the high-efficiency, low-emissions operation that is characteristic of lean-burn compression ignition engines.

5. Acknowledgements

This work at the University of California-Berkeley was partially supported by NSF/DOE Award No. CBET-1258653 entitled “Advancing Low Temperature Combustion and Lean Burning Engines for Light- and Heavy-Duty Vehicles with Microwave Assisted Spark Plugs and Fuel Stratification.”

This research used resources of the National Energy Research Scientific Computing Center, a DOE Office of Science User Facility supported by the Office of Science of the U.S. Department of Energy under Contract No. DE-AC02-05CH11231.

6. Nomenclature

ACE – advanced combustion engine
aTDC – after top dead center
BDC – bottom dead center
BMEP – brake mean effective pressure
bTDC – before top dead center
CA50 – crank angle at which 50% of total heat release has occurred
CFD – computational fluid dynamics
CO – carbon monoxide
CR – compression ratio

574	DI – direct injection
575	EGR – exhaust gas recirculation
576	GDI – gasoline direct injection
577	H ₂ O – water
578	HC – hydrocarbons
579	HCCI – homogeneous charge compression ignition
580	IMEP – indicated mean effective pressure during the closed cycle
581	IMEP _{GE} – indicated mean effective pressure during the gas exchange phase
582	IVC – intake valve closure
583	IVO – intake valve opening
584	KHRT – Kelvin-Helmholtz Rayleigh-Taylor
585	LTC – low temperature combustion
586	N ₂ – nitrogen
587	NO _x – oxides of nitrogen
588	O ₂ – oxygen
589	OP – operating point
590	p _{INT} – intake pressure
591	P _{EGR,COOLER} – required cooling power of the EGR cooler
592	p _{EXH} – exhaust pressure
593	PFI – port fuel injection
594	PFS – partial fuel stratification
595	PM – fully premixed
596	RANS – Reynolds averaged Navier Stokes

597 RPM – revolutions per minute
 598 SHP – start of high pressure cycle
 599 SMD – Sauter mean diameter
 600 TDC – top dead center
 601 T_{EXH} – exhaust gas temperature
 602 VP – turbine vane position
 603 η_E – brake efficiency
 604 η_T – turbine efficiency
 605 η_{TC} – overall variable geometry turbocharger efficiency

606 **7. References**

- 607 [1] Saxena S, Bedoya ID. Fundamental phenomena affecting low temperature
 608 combustion and HCCI engines, high load limits and strategies for extending these
 609 limits. Prog Energy Combust Sci 2013; 39(5): 457-88.
 610 [dx.doi.org/10.1016/j.pecs.2013.05.002](https://doi.org/10.1016/j.pecs.2013.05.002).
- 611 [2] Dec JE, Yang Y, Dernotte J, Ji C. Effects of Gasoline Reactivity and Ethanol
 612 Content on Boosted, Premixed and Partially Stratified Low-Temperature Gasoline
 613 Combustion (LTGC). SAE Int J Engines 2015; 8(3): 935-55.
 614 [dx.doi.org/10.4271/2015-01-0813](https://doi.org/10.4271/2015-01-0813).
- 615 [3] Sjöberg M, Dec JE. Smoothing HCCI Heat-Release Rates Using Partial Fuel
 616 Stratification with Two-Stage Ignition Fuels. SAE Technical Paper 2006.
 617 [dx.doi.org/10.4271/2006-01-0629](https://doi.org/10.4271/2006-01-0629).

- 618 [4] Dec JE, Yang Y, Dronniou N. Boosted HCCI - Controlling Pressure-Rise Rates for
619 Performance Improvements using Partial Fuel Stratification with Conventional
620 Gasoline. SAE Int J Engines 2011; 4(1): 1169-89. dx.doi.org/10.4271/2011-01-
621 0897.
- 622 [5] Yang Y, Dec JE, Dronniou N, Sjöberg M. Tailoring HCCI heat-release rates with
623 partial fuel stratification: Comparison of two-stage and single-stage-ignition fuels.
624 Proc Combust Inst 2011; 33(2): 3047-55. dx.doi.org/10.1016/j.proci.2010.06.114.
- 625 [6] Yang Y, Dec JE, Dronniou N, Sjöberg M, Cannella W. Partial Fuel Stratification
626 to Control HCCI Heat Release Rates: Fuel Composition and Other Factors
627 Affecting Pre-Ignition Reactions of Two-Stage Ignition Fuels. SAE Int. J. Engines
628 2011; 4 (1): 1903-20. dx.doi.org/10.4271/2011-01-1359.
- 629 [7] Yang D, Wang Z, J-X Wang JX, Shuai S. Experimental study of fuel stratification
630 for HCCI high load extension, Appl Energy 2011; 88(9): 2949–54.
631 dx.doi.org/10.1016/j.apenergy.2011.03.004.
- 632 [8] Wolk B, Chen JY. Computational Study of Partial Fuel Stratification for HCCI
633 Engines Using Gasoline Surrogate Reduced Mechanism. Combust Sci Technol
634 2014; 186 (3): 332-54. dx.doi.org/10.1080/00102202.2013.870161.
- 635 [9] Wolk B, Chen JY, Dec JE. Computational study of the pressure dependence of
636 sequential auto-ignition for partial fuel stratification with gasoline. Proc Combust
637 Inst 2015; 35(3): 2993-3000. dx.doi.org/10.1016/j.proci.2014.05.023.
- 638 [10] Babajimopoulos A, Assanis DN, Flowers DL, Aceves SM, Hessel RP. A fully
639 coupled computational fluid dynamics and multi-zone model with detailed
640 chemical kinetics for the simulation of premixed charge compression ignition

engines. Int J Engine Res 2005; 6(5): 497-512.
dx.doi.org/10.1243/146808705X30503.

[11] Embouazza M, Haworth D, Darabiha N. Implementation of Detailed Chemical
Mechanisms into Multidimensional CFD Using in situ Adaptive Tabulation:
Application to HCCI Engines. SAE Technical Paper 2002.
dx.doi.org/10.4271/2002-01-2773.

[12] Wang Z, Wang J, Shuai S, Zhang F. Numerical Simulation of HCCI Engine With
Multi-Stage Gasoline Direct Injection Using 3D-CFD With Detailed Chemistry.
SAE Technical Paper 2004. 2004-01-0563. dx.doi.org/10.4271/2004-01-0563.

[13] Kong SC, Reitz RD. Application of detailed chemistry and CFD for predicting
direct injection HCCI engine combustion and emissions. Proc Combust Inst 2002;
29(1): 663–69. dx.doi.org/10.1016/S1540-7489(02)80085-2.

[14] Ogink R, Golovitchev V. Gasoline HCCI Modeling: Computer Program
Combining Detailed Chemistry and Gas Exchange Processes. SAE Technical
Paper 2001. 2001-01-3614. dx.doi.org/10.4271/2001-01-3614.

[15] AVL Boost v2013.2 - Theory, AVL LIST GmbH, 2013.

[16] Zhou DZ, Yang WM, An H, Li J. Application of CFD-chemical kinetics approach
in detecting RCCI engine knocking fuelled with biodiesel/methanol. Appl Energy
2015; 145: 255-64. dx.doi.org/10.1016/j.apenergy.2015.02.058.

[17] Li J, Yang WM, An H, Zhao D. Effects of fuel ratio and injection timing on
gasoline/biodiesel fueled RCCI engine: A modeling study. Appl Energy 2015; 155:
59-67. dx.doi.org/10.1016/j.apenergy.2015.05.114.

[18] Richards KJ, Senecal PK, Pomraning E. CONVERGE 2.1.0. 2013.

- 664 [19] Mehl M, Chen JY, Pitz WJ. An approach for formulating surrogates for gasoline
665 with application towards a reduced surrogate mechanism for CFD engine
666 modeling. *Energy Fuels* 2011; 25(11): 5215-5223. [dx.doi.org/10.1021/ef201099y](https://doi.org/10.1021/ef201099y).
- 667 [20] Wolk BM. *Fundamental interactions in gasoline compression ignition engines*
668 *with fuel stratification* (Doctoral dissertation, University of California at Berkeley,
669 published in Berkeley, California, USA).
- 670 [21] Orszag SA, Yakhot V, Flannery WS, Boysan F, Choudhury D, Maruzewski J,
671 Patel B. Renormalization group modeling and turbulence simulations. Near wall
672 turbulent flows, 1993: 1031-46.
- 673 [22] Rosin P, Rammler E. The laws governing the fineness of powdered coal. *J Inst of*
674 *Fuel* 1993; 7: 29-36.
- 675 [23] Patterson M, Reitz RD. Modeling the effects of fuel spray characteristics on diesel
676 engine combustion and emission. SAE Technical Paper 1998.
677 [dx.doi.org/10.4271/980131](https://doi.org/10.4271/980131).
- 678 [24] Taritas I, Kozarac D, Sjeric M. Numerical Study of Boosting Configurations and
679 Valve Strategies for High Load HCCI Engine in Wide Range of Engine Speed.
680 SAE Technical Paper 2014. [dx.doi.org/10.4271/2014-01-1267](https://doi.org/10.4271/2014-01-1267).
- 681 [25] Garret Turbochargers, “What is intercooler effectiveness?”
682 www.turbobygarrett.com/turbobygarrett/intercooler-effectiveness , retrieved April
683 2016.
- 684 [26] Zeldovich YB. The oxidation of nitrogen in combustion and explosions. *Acta*
685 *Physicochim. URSS*. 1946; 21(4): 577-628.

686 [27] Dernotte J, Dec JE, Ji C. Investigation of the Sources of Combustion Noise in HCCI
687 Engines. SAE Int. J. Engines 2014; 7(2): 730-61. [dx.doi.org/10.4271/2014-01-](https://doi.org/10.4271/2014-01-1271)
688 1271.