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Premixed Compression Ignition (PCI) Combustion with Modeling-Generated Piston Bowl Geometry in a Diesel Engine

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ABSTRACT

Sustainable PCI combustion was achieved in a light-duty diesel engine through the installation of a 120° spray angle nozzle and modeling-generated piston bowl geometry developed for compatibility with early start-of-injection timings. Experimental studies were conducted to determine favorable settings for boost pressure, SOI timing, and EGR rate at 2000 rev/min, 5 bar BMEP. An optimal SOI timing was discovered at 43° BTDC where soot and NOx emissions were reduced 89% and 86%, respectively. A 10% increase in fuel consumption was attributed to increased HC and CO emissions as well as non-optimal combustion phasing. Combustion noise was sufficiently attenuated through the use of high EGR rates. The maximum attainable load for PCI combustion was limited by the engine's peak cylinder pressure and cylinder pressure rise rate constraints.

INTRODUCTION

PCI combustion can be characterized by fuel delivery occurring early in the compression stroke when cylinder pressure and temperature are below the flammability limits of diesel fuel [1]. Substantial mixing time is provided which leads to a reduction in local equivalence ratios. The subsequent combustion occurs nearly simultaneously throughout the cylinder resulting in the absence of diffusion combustion, which greatly reduces soot and NOx production.

PCI combustion is fundamentally different from Homogeneous Charge Compression Ignition (HCCI) combustion in that variations in local equivalence ratio are present throughout the cylinder ranging from near stoichiometric conditions in the combustion bowl to very lean conditions elsewhere in the cylinder [2]. True HCCI combustion requires a uniform equivalence ratio throughout the mixture. This is typically accomplished through port fuel injection or fuel-air mixing upstream of the intake port. However, port fuel injection of diesel fuel is difficult, because the fuel's high boiling point results in poor vaporization and elevated hydrocarbon (HC) and carbon monoxide (CO) emissions [3]. Therefore, PCI combustion represents a more feasible approach to reducing diesel engine emissions through the utilization of recent technological advancements in common-rail fuel systems.

PCI combustion has shown promising results toward simultaneously reducing particulate matter (PM) and nitric oxide (NOx) emissions [4]. However, PCI combustion poses several challenges for engine development engineers. The first challenge of PCI combustion is its extreme sensitivity to incremental changes in Exhaust Gas Recirculation (EGR) rate [5]. As a result, the development of transient engine control strategies is difficult. Current limitations in EGR rate control can be further complicated by unequal EGR distributions existing between individual intake runners, resulting in combustion instability. The second challenge of PCI combustion is increased fuel consumption due to increased HC and CO emissions stemming from incomplete combustion [1]. The final challenge of PCI combustion is elevated combustion noise. Because the fuel/air mixture ignites nearly simultaneously throughout the cylinder, the rate of cylinder pressure rise can be substantial. Fortunately, heavy EGR rates can be used to combat combustion noise by delaying the start of combustion [6].

One objective of this research is to eliminate the threat of engine oil dilution by diesel fuel so that the underlying challenges of PCI combustion can be safely investigated and quantified. This is accomplished through the installation of a piston having a modeling-generated combustion bowl geometry receptive of early SOI timings. This new piston geometry is mated with a narrow spray angle nozzle to greatly expand the range of early SOI timings that can be safely explored without causing engine oil degradation.

The primary objective of this research is to experimentally verify that engine simulation software and micro-genetic algorithms can be successfully leveraged to create a versatile piston bowl geometry [7] that will allow engine operation in either PCI or diffusion combustion modes.

An engine that is capable of utilizing both combustion modes offers the advantage of low soot and NOx production in PCI combustion mode while still retaining its high load capability in diffusion combustion mode. Because the details of diffusion combustion have been thoroughly documented elsewhere [8, 9, 10], this paper will focus primarily on PCI combustion results.

EXPERIMENTAL SET- UP

ENGINE AND DYNAMOMETER

The test engine used in this research is a single-cylinder version of Fiat's 1.9L four-cylinder High Speed Direct-Injection (HSDI) diesel engine. The engine is fully instrumented and coupled to a 50 hp dynamometer controlled by a Dyne-Loc IV digital dynamometer controller. The engine has a maximum speed of 4200 rev/min and a maximum power of 22kW at 3800 rev/min. The cylinder head and jug were manufactured by Fiat. The engine block is a Hydra model acquired from Ricardo Research. The specifications of the engine are presented in Table 1.

Table 1: Engine specifications.

Engine Type	Single-cylinder HSDI diesel
Displacement	477cc
Bore x Stroke	82.0 x 90.4 mm
Compression ratio	Production piston - 18.9 : 1
	New Piston - 16.0 : 1
Intake ports	1 - Helical
	1 - Directed
Piston type	Production piston - re-entrant bowl
	New piston - open crater
IVO	10° BTDC
IVC	38° ABDC
EVO	38° BBDC
EVC	8.5° ATDC

INTAKE AND EXHAUST SYSTEM

The intake and exhaust systems are configured for simulated turbocharging. Compressed intake air is supplied to the test cell at a pressure of 620 kPa. A pressure regulator is used to control the upstream pressure of four critical flow orifices which allow for an accurate measurement of mass flow. The critical flow orifices are arranged in parallel with individual valves to ensure choked flow. After passing through the selected orifices, the intake air is routed through a 3 kW heater. The heater's temperature setting is adjustable, and a PID controller is used to cycle power to the heater as needed. The electrical current demand is determined through a temperature feedback circuit connected to the intake surge tank tank.

In an engine equipped with a turbocharger, back-pressure is applied to the engine via the flow restriction created by the turbine. To simulate this back-pressure, a valve was installed in the exhaust plumbing and was controlled with a pneumatic actuator. Figure 1 shows a schematic of the engine set-up developed by Tennison [11].

EXHAUST GAS RECIRCULATION SYSTEM

EGR is attained through a direct line which connects the intake and exhaust surge tanks. The pressure differential between the two surge tanks is used to control the flow of EGR. The exhaust surge tank pressure is adjusted based on the intake surge tank pressure setting. In all cases, the exhaust pressure is higher than the intake

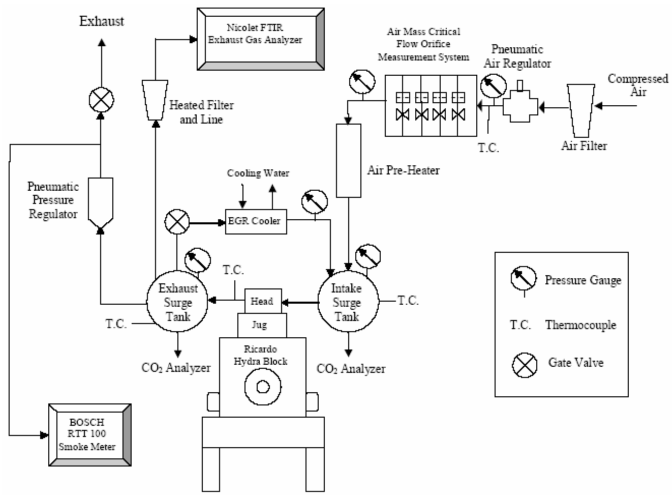

Figure 1: Experimental set-up schematic.

pressure to simulate back-pressure induced by a turbine and to drive the flow of EGR toward the intake surge tank. EGR rate is calculated with the following equation:

$$\%EGR = \frac{\%CO_2\,(intake) - \%CO_2\,(ambient)}{\%CO_2\,(exhaust) - \%CO_2\,(ambient)} \times 100$$

EMISSIONS MEASUREMENTS

Measured emissions include soot, NOx, CO, HC and CO2. Gaseous emissions are monitored with a Thermo Nicolet NEXUS 670 FT-IR emissions analyzer. Samples are drawn from an insulated stainless steel probe inserted into the exhaust surge tank. From there, the gases travel to the analyzer through a 170° C heated line to prevent the condensation of emissions species.

Soot emissions measurements are made with a Bosch RTT100 smoke opacimeter. The unit self-calibrates from 0 to 100% opacity before each measurement. Through the use of an internal conversion table, the instrument converts visual opacity into mass concentration. The smoke meter displays mass concentration in terms of mg/m3, so a conversion is required to achieve units of g/kg-fuel.

EMISSIONS TARGETS

An objective of this research was to provide data for the development of a European passenger car engine. Therefore, the New European Drive Cycle (NEDC) was used to select appropriate operating conditions. By reducing the NEDC to steady-state operating points through statistical methods, the research sponsor was able to generate three weighted operating points. Based on these weighting factors and Euro 4 standards, emissions targets were developed for each of the three operating points.

Table 2: Emissions targets [g/kg-fuel] [12].

Soot	NOx	HC	CO
0.58	1.0	7.7	77

Table 2 shows the calculated emissions targets based on Euro 4 emissions standards for the 2000 rev/min, 5 bar steady-state point. Data points representing these values appear in several plots throughout the paper and are labeled as "Target" in each occurrence.

FUEL DELIVERY SYSTEM

A common-rail system with a maximum injection pressure of 1600 bar is used to deliver commercial grade # 2 diesel fuel to the engine. This system is capable of delivering multiple injections per cycle with an electro-hydraulically controlled injector. Start-of-injection timing, duration, and rail pressure are set through a graphical user interface (GUI) on a personal computer. The 8-hole minisac nozzles used in this research were manufactured by the Robert Bosch Corporation. Additional details are provided in Table 3.

In this paper, the term ‘spray angle’ refers to the included angle between the centerlines of opposing spray plumes emitted from the fuel injector’s nozzle, as shown in Figure 2. Nozzle spray visualization experiments were conducted by Lee [13] using a pressurized bomb and high-speed camera. Lee’s results verified that the nozzles used in this research had symmetrical, evenly distributed spray plumes.

The term ‘shim height’ is used to refer to the height of the copper shim used for sealing the surfaces between the common-rail injector and cylinder head. The height of the shim dictates the nozzle’s protrusion into the combustion chamber and, therefore, affects spray targeting of the combustion bowl.

A designed experiment was used to determine the most favorable combination of injector spray angle and shim height. The test parameters and settings are listed in Table 4. Figure 3 shows the injector spray plume trajectories for the five nozzles tested and the production piston profile at TDC. Figure 4 shows the soot response surface to injector shim height and spray angle at 2000 rev/min, 5 bar BMEP. Because improved spray targeting leads to improved mixing and lower soot emissions, soot emissions were chosen as the primary spray targeting metric. Therefore, a spray-angle of 148° with a shim height of 1.8 mm was selected.

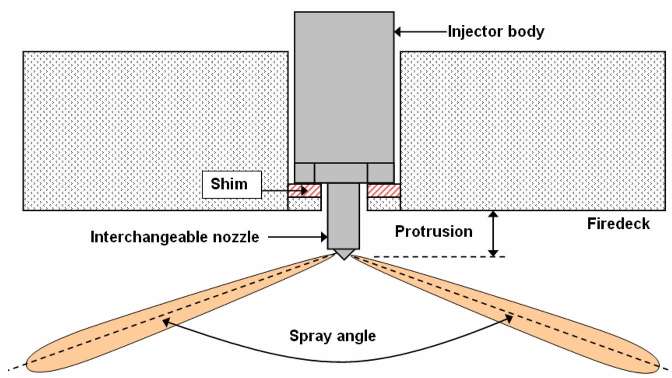

Figure 2: Common-rail injector terminology.

Table 3: Common-rail injection system specifications.

Injector type	electro-hydraulically controlled
Injection pressure	variable up to 1600 bar
Nozzle types	minisac
Nozzle hole configuration	8-hole
Included spray angles	110º, 120º, 130º, 140º, 145º, 148º, 151º, and 154º

Table 4: Test parameters and settings for production piston spray targeting study.

Parameter	Spray targeting settings
Engine speed [rev/min]	2000
Load [BMEP]	5
SOI [º bTDC]	7
Injection pressure [bar]	750
Spray angle [deg]	140-154
Shim thickness [mm]	1.0-2.5
Nozzle hole number	8
Boost pressure [kPa]	130.3
Exhaust pressure [kPa]	140
Intake temp [C]	90
EGR [%]	0

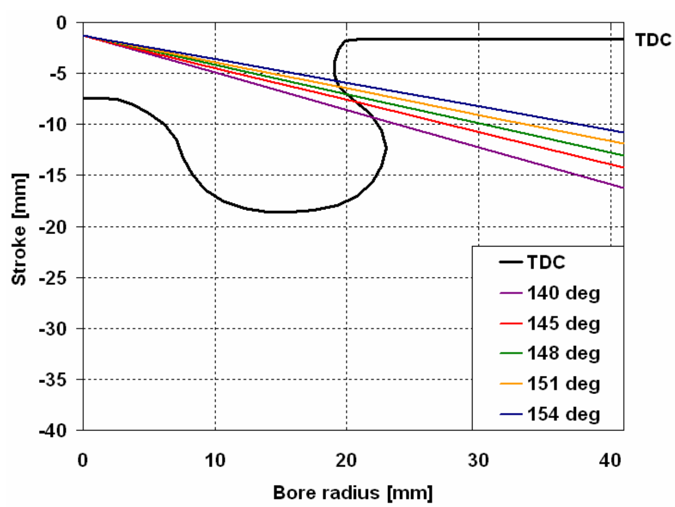

Figure 3: Injector spray plume trajectories with production piston profile at TDC.

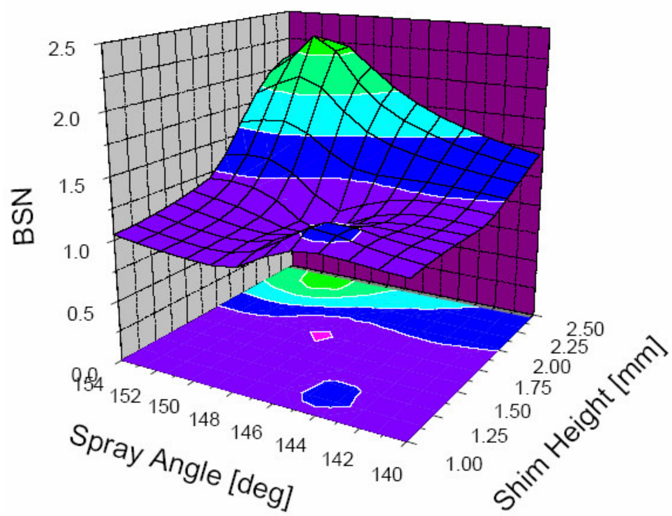

Figure 4: Bosch Smoke Number (BSN) response to injector shim height and spray angle.

CYLINDER PRESSURE MEASUREMENT

Cylinder pressure data is acquired through a piezo-electric pressure transducer (Kistler, model 6125A) mated to a charge amplifier (Kistler, model 5010B). The signal is routed to a National Instruments data acquisition board which is connected to a PC. The board also receives locating signals from a shaft encoder on the crankshaft and a half-speed signal from an optical interrupter on the camshaft. The data acquisition program, which was developed by Thiel [14], incorporates an equation for apparent heat release rate [15]. Cylinder pressure data is collected at 0.25° increments. All cylinder pressure and heat release information displayed in this paper represents an average of 200 engine cycles.

EARLY SOI TIMING LIMITATIONS OF PRODUCTION PISTON BOWL

In order to determine a safe SOI timing range with the production piston and matched spray angle, an injection timing sweep was conducted. The parameters and settings for the test are shown in Table 5. Based upon the engine manufacturer's recommendations, a peak cylinder pressure constraint of 130 bar and maximum cylinder pressure rise rate constraint of 15 bar/deg were imposed. If either of these limits were in jeopardy of being exceeded, the fueling rate was reduced at that test point. Due to these constraints, the prescribed BMEP could not be achieved at advanced SOI timings, as shown in Figures 5 and 6.

The broad SOI timing sweep was conducted without EGR. Emissions and fuel consumption data were recorded at all test points. As the SOI timing is advanced from 4° ATDC towards 25° BTDC, a traditional Maximum Brake Torque (MBT) curve is observed. As the injection timing is advanced beyond 25° BTDC, violation of the cylinder pressure rise rate constraint occurs. As a result, the necessary quantity of fuel required to achieve 5 bar BMEP could not be delivered. In these cases, the fuel flow rate was incrementally reduced until the cylinder pressure rise rate fell within the predetermined test constraint.

As the injection timing is advanced beyond 30° BTDC, the quantity of fuel delivered prior to observing a violation of the cylinder pressure rise rate constraint increases. The explanation for this trend can be found in the HC emissions presented in Figure 5. As the injection timing is advanced beyond 25° BTDC, a dramatic upswing in HC emissions occurs. This indicates that significant spray-wall impingement is occurring at advanced injection timings as a result of poor spray targeting.

Spray-wall impingement must be avoided to limit lubrication oil dilution by unburned fuel. In early injection cases, a portion of the poorly targeted fuel adheres to the cylinder wall and eventually enters the crankcase where it acts as a viscosity reducer. The result is typically main bearing failure. To avoid this catastrophic outcome, SOI timings earlier than 25° BTDC are deemed unsustainable with this hardware configuration.

Table 5: Test parameters and settings for production piston SOI timing sweep.

Parameter	SOI timing sweep settings
Engine speed [rev/min]	2000
Load [BMEP]	5
SOI [º bTDC]	-4, 0, 5, 10, 15, 20, 25, 30, 35, 40
Injection pressure [bar]	750
Spray angle [deg]	148
Protrusion [mm]	1.8
Nozzle hole number	8
Boost pressure [kPa]	130.3
Exhaust pressure [kPa]	140
Intake temp [C]	90
EGR [%]	0

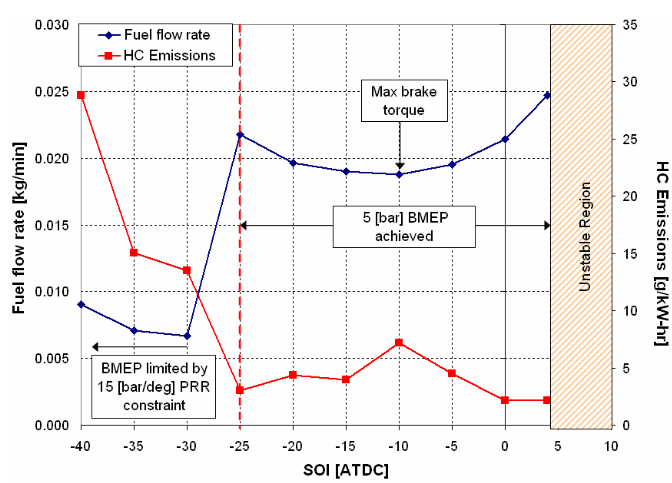

Figure 5: SOI timing sweep with production piston and matched injector spray angle.

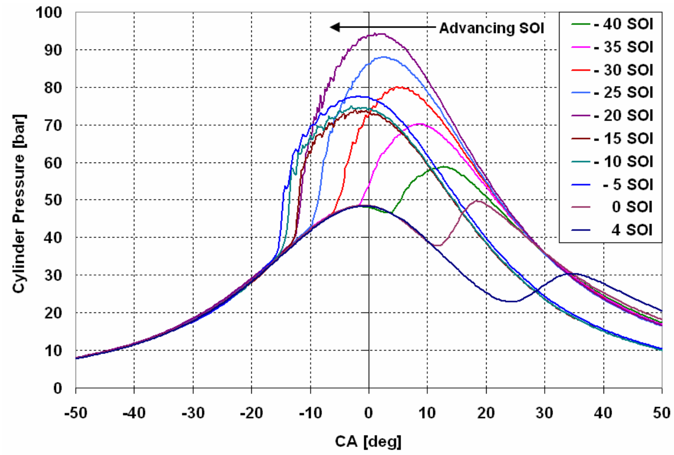

Figure 6: Pressure traces from SOI timing sweep test.

It quickly became apparent that constraining SOI timings to 25° BTDC would not provide the necessary freedom to conduct a thorough study of PCI combustion. One potential solution was to install a narrow angle nozzle to limit spray impingement on the cylinder walls. However, the production piston bowl was not designed for narrow spray angles. Preliminary spray targeting tests revealed an incompatibility with spray angles deviating even slightly from the production spray angle. Therefore, changing only the spray angle was not a viable option.

DEVELOPING A NEW PISTON BOWL GEOMETRY

The piston's combustion bowl and injector spray characteristics represent the two most important components of a diesel combustion system. Therefore, it is crucial that they be designed in tandem.

The impetus for creating a new piston geometry stemmed from the desire to develop a versatile combustion system that could perform well with either early or conventional SOI timings. In theory, this would provide the flexibility of operating with PCI combustion at low or medium loads and diffusion combustion at high loads. Through this strategic combination of combustion modes, the engine would benefit from enhanced low emissions capability while still retaining high load potential. This would ensure that no speed or load compromises would exist when compared to the production engine.

Diwakar [16] coupled micro-genetic algorithms with the multidimensional KIVA code to generate a new piston bowl geometry. To ensure the versatility of the combustion system, special considerations were taken when establishing the model's constraints. For instance, the injector spray angle was constrained to a maximum value of 130 degrees to guarantee that early SOI capability would exist at the conclusion of the optimization. In addition, the simulation was run at 4200 rev/min, full-load to ensure the combustion system would be capable of producing high power.

Over the course of the simulation, the model sought the widest allowable spray angle as an avenue for improving air utilization during the high speed, high load optimization. Figure 7 shows the resulting modeling-generated piston bowl profile overlaid upon the production piston profile. The most noteworthy difference is the change from a re-entrant to an open crater combustion bowl. The compression ratio was actively reduced from 18.9 to 16.0 to facilitate lower NOx emissions through decreased peak cylinder pressures and temperatures.

BASELINING THE NEW PISTON

SPRAY TARGETING

Following the installation of the new modeling-generated piston, spray targeting studies were performed to determine the ideal injector spray angle. Conventional diffusion combustion was chosen for the targeting studies, because it is much more sensitive to spray angle values than early injection PCI combustion. This also ensured that diffusion combustion could be effectively engaged at high load conditions where PCI combustion would not be sustainable.

The simulation had converged to a spray angle of 130°, so the experimental validation began at this point. EGR sweep tests with fixed LPP and BMEP were run for three different spray angles, as shown in Figure 8. The 110°, 120°, and 130° nozzles were all 8-hole minisac type. The test parameters and settings are listed in Table 6.

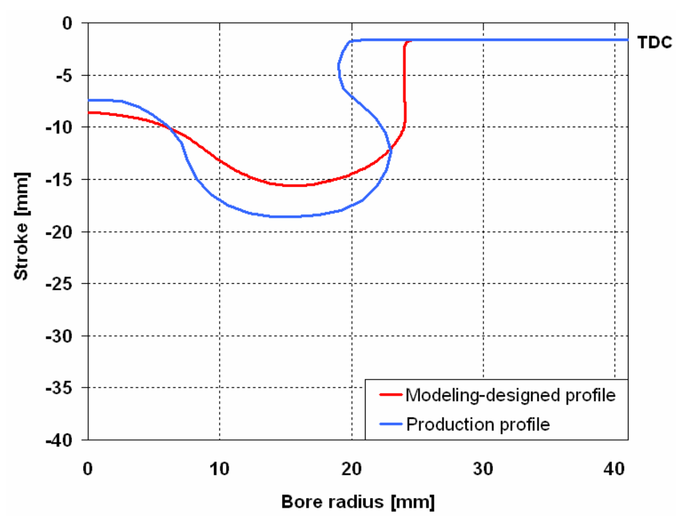

Figure 7: Production and modeling-generated piston bowl profiles.

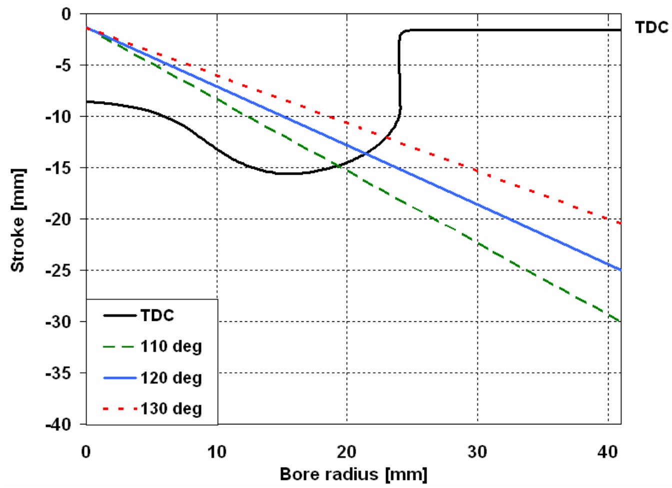

Figure 8: Injector spray plume trajectories with modeling-generated piston profile at TDC

The soot-NOx tradeoff curves generated from the spray targeting study are displayed in Figure 9. The results are evaluated primarily on the basis of soot emissions, because lower soot emissions are indicative of improved mixing and spray targeting. The 110° and 130° nozzles produce relatively higher soot levels when compared to the 120° nozzle. As a result, the 120° nozzle is implemented in all subsequent testing.

Table 6: Test parameters and settings for spray targeting of modeling-generated piston.

Parameter	Case 1	Case 2	Case 3
Engine speed [rev/min]	2000	2000	2000
Load [BMEP]	5	5	5
SOI [° bTDC]	4, 5, 7, 9	4, 5, 7, 8	5, 6, 8, 10
Injection pressure [bar]	1500	1500	1500
Spray angle [deg]	110	120	130
Protrusion [mm]	1.8	1.8	1.8
Nozzle hole number	8	8	8
Boost pressure [kPa]	130.3	130.3	130.3
Exhaust pressure [kPa]	140.0	140.0	140.0
Intake temp [C]	90	90	90
EGR [%]	21, 30, 36, 46	22, 29, 36, 38	23, 31, 38, 47

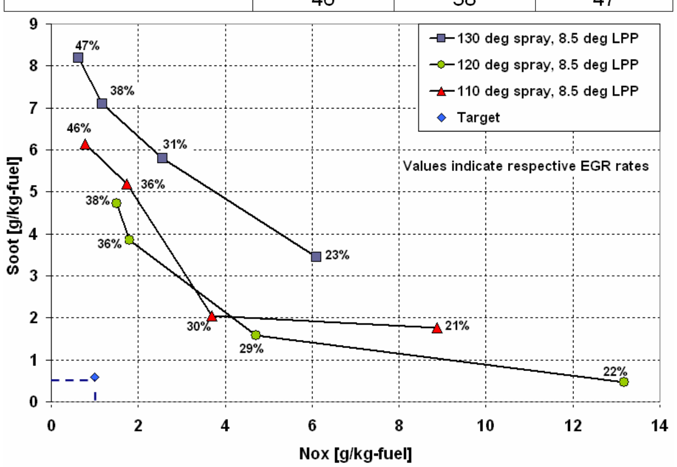

Figure 9: Soot and NOx results for spray targeting study of modeling-generated piston.

FUEL CONSUMPTION BASELINE

Tests were carried out with the modeling-generated piston to determine the Location of Peak Pressure (LPP) corresponding to Maximum Brake Torque (MBT). No EGR was used and BMEP was fixed at 5 bar. At this operating condition, MBT is located at 8.5° ATDC. The minimum ISFC corresponding to MBT is 211 g/kW-hr, as shown in Fig. 10.

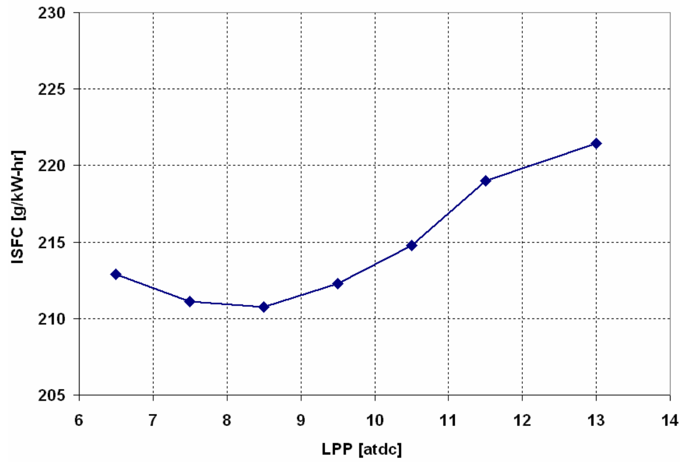

Figure 10: ISFC vs. LPP for SOI timing sweep with modeling-generated piston.

EMISSIONS BASELINE

A SOI timing sweep from 1° to 13° BTDC established baseline soot and NOx emissions for the new piston at 2000 rev/min, 5 bar BMEP. NOx emissions are substantial due to diffusion combustion amid excess air without EGR. The results are shown in Fig 11.

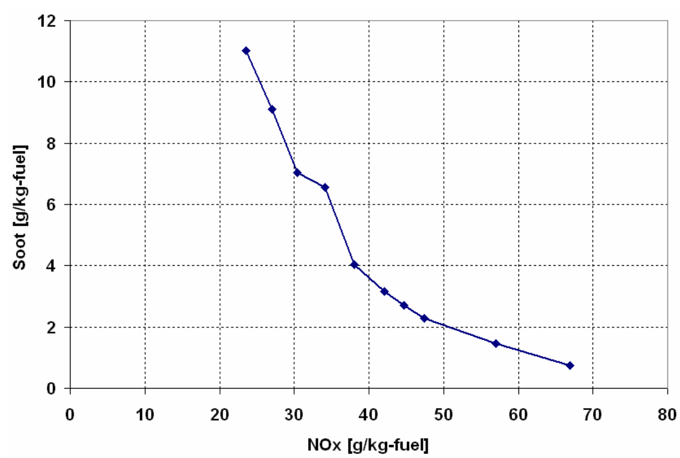

Figure 11: Baseline soot and NOx emissions for the modeling-generated piston.

PCI COMBUSTION AT MEDIUM SPEED, MEDIUM LOAD

The migration from conventional diffusion combustion to early-injection PCI combustion was accomplished through incremental changes in engine operating parameters such as SOI timing, boost pressure, and EGR rate. The objective was to understand each parameter's effect on PCI combustion while minimizing deviations from diffusion combustion settings, thereby enabling rapid combustion mode switching. However, the necessary parameter changes were found to be significant and time intensive with respect to engine transient time scales. EGR response clearly emerged as the principal limiting parameter.

INJECTION TIMING EFFECT

The first parameter investigated during the migration toward PCI combustion was injection timing. A broad SOI timing sweep was conducted to demonstrate its impact on emissions. The SOI timing was swept from 15° to 51° BTDC in 3° increments as shown in Table 7.

Table 7: Test settings for SOI timing sweep with fixed EGR rate and modeling-generated piston.

Parameter	Fixed EGR
Engine speed [rev/min]	2000
Load [BMEP]	5
SOI [º bTDC]	15, 18, 21, 24, 27, 30, 33, 36, 39, 42, 43, 44, 45, 48, 51
Injection pressure [bar]	1500
Spray angle [deg]	120
Protrusion [mm]	1.8
Nozzle hole number	8
Boost pressure [kPa]	130.3
Exhaust pressure [kPa]	140.0
Intake temp [C]	90
EGR [%]	40

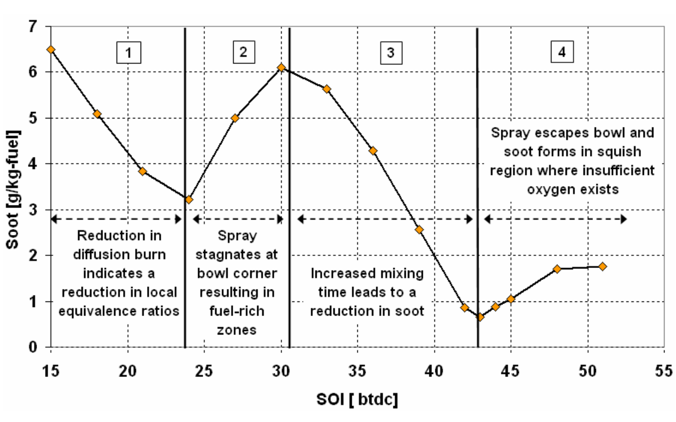

Figure 12: Soot response to SOI timing sweep with 40% EGR rate.

Based upon prior testing results, a fixed EGR rate of 40% was selected for the injection timing sweep. As the injection timing is advanced from 15° BTDC towards 23° BTDC, the soot emissions decrease from 6.5 to 3.25 g/kg-fuel, as shown in Figure 12. This is attributed to a reduction in the diffusion burn as more of the fuel participates in the premixed burn. This was verified through an examination of the heat release data.

As the SOI timing is advanced beyond 23° BTDC, the soot emissions increase rapidly. This is thought to be a result of spray stagnation in the corner region of the piston bowl which produces local fuel-rich zones, depicted in Figure 13. This conclusion is supported through computational data provided by Kim et al. [17].

As the injection timing is advanced beyond 30° BTDC, a significant reduction in soot is observed with a minimum occurring at 43° BTDC. This is attributed to an increase in mixing time afforded to the air-fuel charge by early SOI timings. In all cases where SOI timing occurs before 33° BTDC, ignition begins at approximately 12° BTDC as shown in Figure 14. This confirms that combustion is being controlled solely by chemical kinetics. Consequently, in this timing range, advances in SOI timing translate directly to longer mixing times.

As the injection timing is advanced beyond 43° BTDC, the soot emissions begin to rise. This is a result of poor spray targeting as the spray plumes escape the piston bowl, as shown in Figure 13. The end result is fuel rich zones in the squish region which cannot be fully oxidized.

Based upon these results, 43° BTDC is deemed the best injection timing for this combustion bowl and EGR level at 2000 rev/min, 5 bar. Accordingly, this SOI timing is fixed for all subsequent parametric tests.

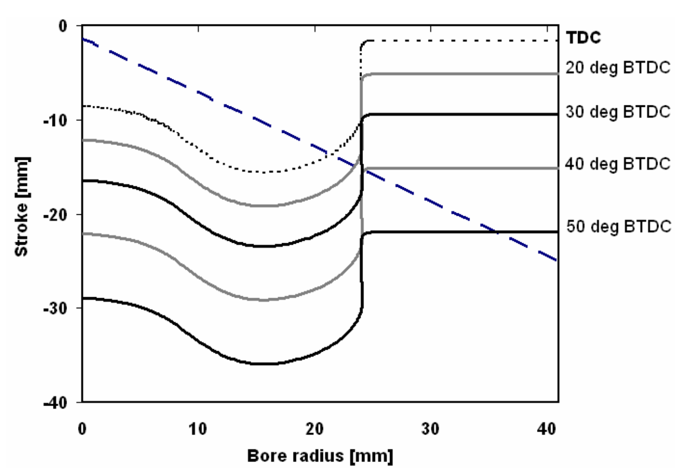

Figure 13: Piston-spray interaction diagrams with 120° spray trajectory and modeling-generated piston profile at the four stages shown in Fig. 12 during the compression stroke.

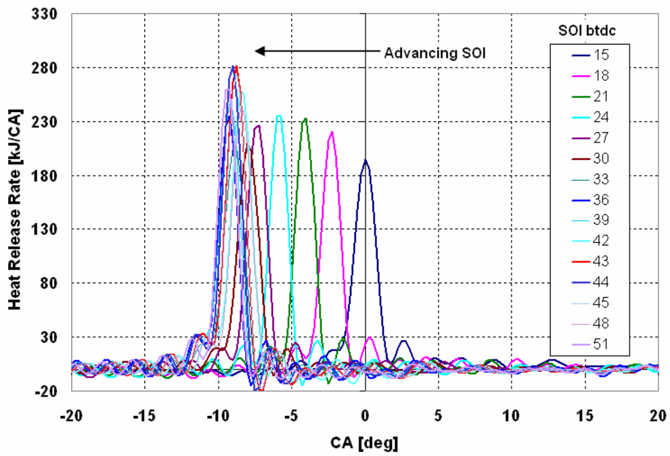

Figure 14: Heat release rates for SOI timing sweep.

BOOST PRESSURE AND EGR EFFECTS

The effect of boost pressure was evaluated through EGR sweep tests conducted at five boost pressure settings ranging from 130.3 kPa to 185.5 kPa. Backpressure settings are adjusted at each boost pressure setting to achieve the desired EGR rates. Despite high equivalence ratios, emissions targets are met with a boost pressure of 164.8 kPa over a range of 55-70% EGR, as shown in Figure 15. At the 61% EGR point, soot and NOx emissions are reduced by 89% and 86% from the baseline emissions, respectively.

In order to avoid violating the maximum allowable cylinder pressure constraint, the EGR rate for the 164.8 kPa boost pressure case cannot be decreased below 49% as depicted in Figure 16. Further reductions in EGR rate cause an advance of the heat release rate in the compression stroke, leading to exceedingly high peak cylinder pressures.

Table 8: Test parameters and settings for EGR sweep test conducted at four boost pressure cases.

Parameter	Case 1	Case 2	Case 3	Case 4
Engine speed [rev/min]	2000	2000	2000	2000
Load [BMEP]	5	5	5	5
SOI [º bTDC]	43	43	43	43
Injection pressure [bar]	1500	1500	1500	1500
Spray angle [deg]	120	120	120	120
Protrusion [mm]	1.8	1.8	1.8	1.8
Nozzle hole number	8	8	8	8
Boost pressure [kPa]	130.3	144.1	151.0	164.8
Exhaust pressure [kPa]	140.0	153.8	160.6	174.4
Intake temp [C]	90	90	90	90
EGR [%]	42, 43, 45, 49	45, 47, 48, 52	48, 49, 52, 53	49, 51, 52, 54, 57, 61, 65, 68

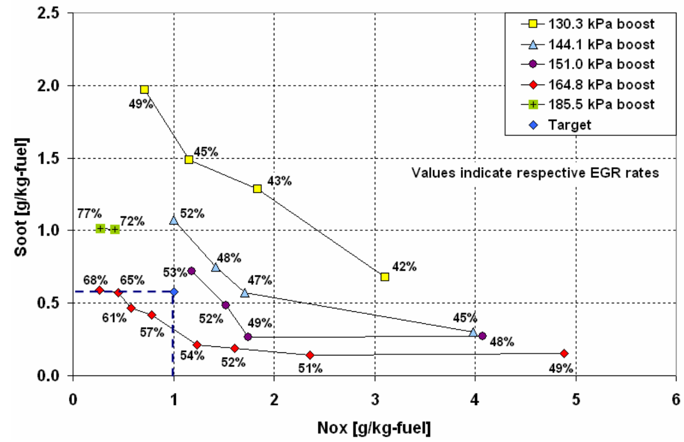

Figure 15: Soot and NOx results for EGR sweep tests with varying levels of intake boost pressure.

From the corresponding heat release rates plotted in Figure 17, the effects of high EGR rates are apparent. As the EGR rate is increased beyond 55%, the start of combustion is delayed considerably. This is marked by a decrease in the peak value of the heat release rate as well as an overall broadening of the trace. This broadening contributes to greater heat transfer losses as the combustion occurs over a longer crank-angle duration.

As the boost pressure is increased beyond 164.8 kPa, large amounts of EGR are required for sufficient start-of-combustion delay. At the 185.5 kPa boost pressure case, an EGR rate less than 70% is not allowable due to violation of the 130 bar maximum cylinder pressure constraint. The result of this constraint can be observed in Figure 18. As a consequence of the EGR rates being constrained to such high values, further reductions in soot emissions are not possible. If the peak allowable cylinder pressure constraint was extended to 140 or 150 bar, it is expected that a soot-NOx tradeoff curve would appear below the target emissions values.

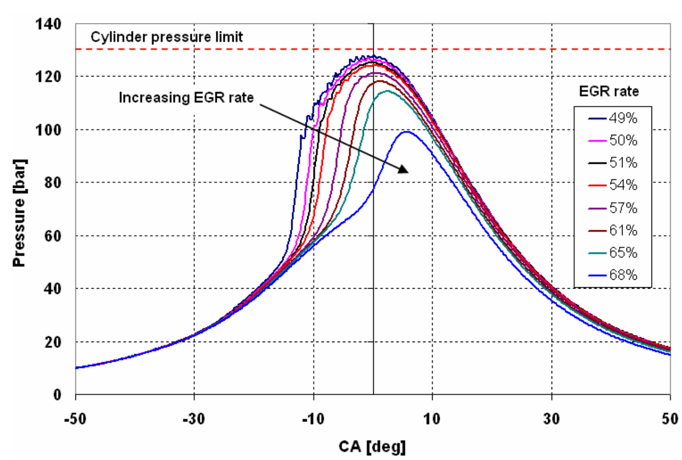

Figure 16: Cylinder pressure response to EGR rate at 164.8 kPa intake boost pressure.

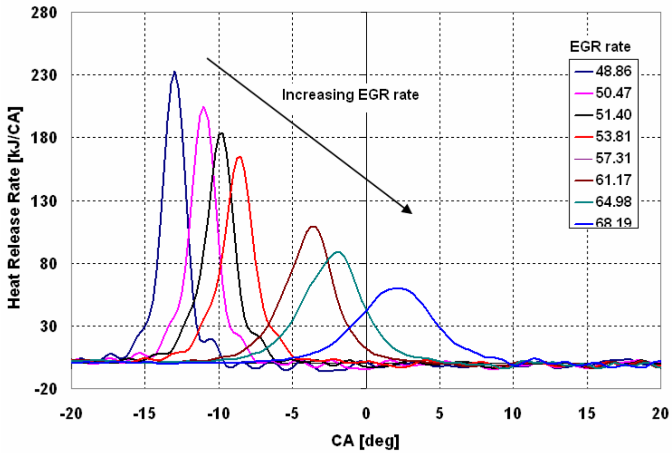

Figure 17: Heat release rate response to EGR rate at 164.8 kPa intake boost pressure.

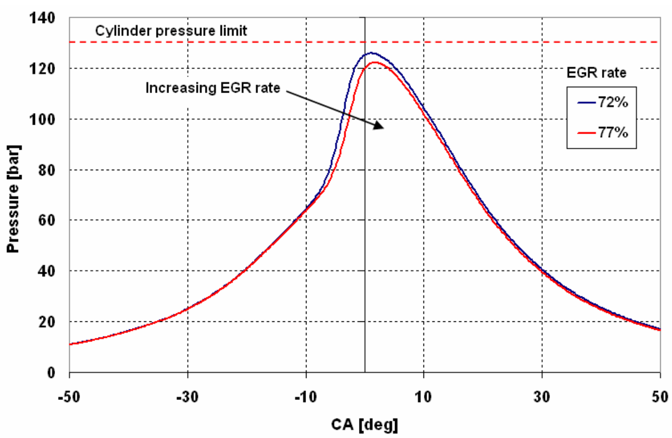

Figure 18: Cylinder pressure response to EGR rate at 185.5 kPa intake boost pressure.

HC AND CO EMISSIONS

In addition to monitoring soot and NOx emissions, it was also important to monitor HC and CO emissions. These two emissions species are generally quite low for diffusion combustion because of low equivalence ratios and high cylinder temperatures. However, they become a concern when equivalence ratios increase and bulk gas temperatures decrease. Consequently, elevated HC and CO emissions are common during PCI combustion where these conditions are difficult to avoid.

Figure 19 shows that decreasing the equivalence ratio was required to reduce HC and CO emissions, indicating that if more oxygen is available, HC and CO emissions are more likely to convert to less harmful product gases. Additionally, as the equivalence ratio decreases, the high specific heat inert gas composition also decreases due to the decreasing EGR rate. This results in higher cylinder temperatures which are critical for ensuring the complete conversion of these intermediate species.

COMBUSTION NOISE

Combustion noise became a major concern when SOI timings were advanced significantly beyond TDC. The derivative of the cylinder pressure trace was used to evaluate combustion noise and is referred to as the cylinder pressure rise rate. The peak rate of cylinder pressure rise was a primary concern pertaining to both engine hardware durability and combustion noise. A value of 15 bar/deg was selected as an allowable value for $dP/d\theta$. In cases where this preset value was exceeded, test durations were made as short as possible to avoid engine damage while still allowing for sufficient data collection. Figure 20 shows the significant effect EGR rate has on reducing the peak rate of cylinder pressure rise. As the EGR rate surpasses 55%, $dP/d\theta$ values become acceptable. With the addition of more EGR, the noise generated from PCI combustion is at or below the level of conventional combustion. As a result, the points with the highest EGR rate produced the quietest combustion as well as the most significant reductions in soot and NOx emissions.

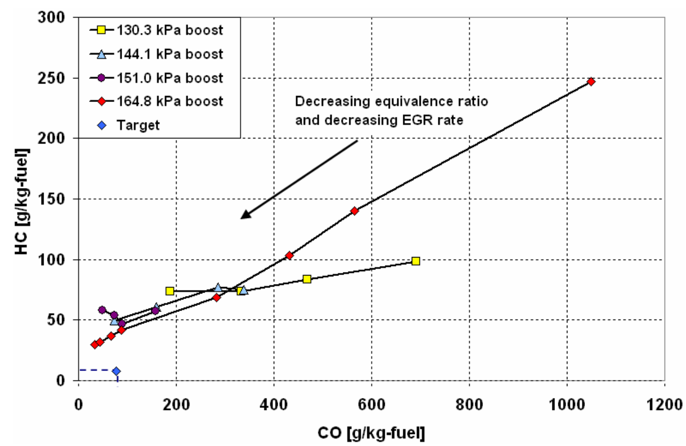

Figure 19: HC and CO results for EGR sweep tests at varying levels of intake boost pressure.

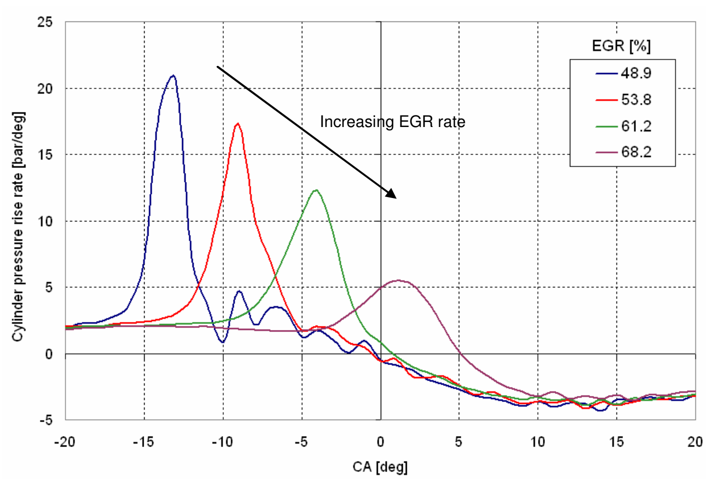

Figure 20: Cylinder pressure rise rate response to EGR rate at 164.8 kPa intake boost pressure.

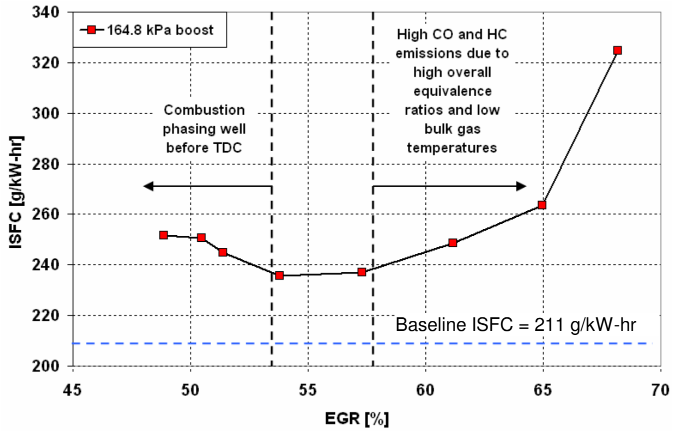

Figure 21: ISFC versus EGR rate at 164.8 kPa intake boost pressure

FUEL CONSUMPTION

Fuel consumption was monitored closely during all testing. Figure 21 shows how ISFC responds to EGR rate for the test conditions listed in Case 4 of Table 8. Several factors contribute to these results, with the most significant being high HC and CO emissions at test points relying on high EGR rates. The other contributing factor was advanced heat release rates due to non-optimal combustion phasing.

Minimum fuel consumption appears between 54 and 58% EGR. With these EGR rates, a 10% fuel consumption penalty over the 211 g/kW-hr baseline is recorded and is visible in Figure 21. If the EGR rate is decreased from the aforementioned values, fuel consumption increases due to an advancement of combustion phasing.

CONCLUSIONS

The present modeling-generated piston bowl geometry, when mated with a 120° spray angle nozzle, allowed for both sustainable PCI combustion and conventional diffusion combustion.

The crux of this research focused on quantifying the capability of early-injection PCI combustion. The load limitations of PCI combustion were found to hinge heavily upon the mechanical limitations of the engine. Cylinder pressure and cylinder pressure rise rate constraints limited the maximum attainable load in the single-cylinder engine to 5 bar BMEP. Short crank-angle duration combustion events, which were phased slightly before TDC, contributed to high peak cylinder pressures.

Parametric studies including variations in boost pressure, injection timing, and EGR rate were used to improve exhaust emissions at 2000 rev/min, 5 bar BMEP. An interesting trend in soot emissions was discovered when SOI timing was swept from 15° to 51° BTDC. An optimum SOI timing was discovered at 43° BTDC, so this timing was fixed for all subsequent testing.

Soot and NOx emissions reductions of 89% and 86% were observed at the final engine settings, respectively. A 10% increase in fuel consumption was attributed to a combination of factors including an increase in HC and CO emissions as well as non-optimal combustion phasing.

The effects of boost pressure and EGR rate were also investigated. These two parameters were used to control combustion phasing and, consequently, combustion noise. The balance between boost pressure and EGR rate dictated the equivalence ratio and had a significant effect on engine-out emissions.

It is concluded that future engine hardware, if developed to withstand greater peak cylinder pressures, will enable PCI combustion to be extended toward higher loads. Furthermore, if injection timings are scaled with engine speed to provide sufficient mixing times, higher engine speeds may also be attainable through PCI combustion.

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