2005-01-0167

Evaluation of a Narrow Spray Cone Angle, Advanced Injection Timing Strategy to Achieve Partially Premixed Compression Ignition Combustion in a Diesel Engine

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ABSTRACT

Simultaneous reduction of nitric oxides (NO_x) and particulate matter (PM) emissions is possible in a diesel engine by employing a Partially Premixed Compression Ignition (PPCI) strategy. PPCI combustion is attainable with advanced injection timings and heavy exhaust gas recirculation rates. However, over-advanced injection timing can result in the fuel spray missing the combustion bowl, thus dramatically elevating PM emissions. The present study investigates whether the use of narrow spray cone angle injector nozzles can extend the limits of early injection timings, allowing for PPCI combustion realization. It is shown that a low flow rate, 60-degree spray cone angle injector nozzle, along with optimized EGR rate and split injection strategy, can reduce engine-out NO_x by 82% and PM by 39%, at the expense of a modest increase (4.5%) in fuel consumption. This PPCI strategy has the potential for meeting upcoming stringent fuel specific NO_x emission levels of less than 1 g/kg-fuel and fuel specific PM levels less than 0.25 g/kg-fuel.

INTRODUCTION

Homogeneous Charge Compression Ignition (HCCI) combustion in diesel engines promises to deliver thermal efficiencies similar to conventional modes of combustion, but with dramatically lower levels of NO_x, PM emissions [1]. The need for such a novel mode of combustion stems from the legislated, ultra low, future emission standards for passenger vehicles, while maintaining high fuel economy specifications. For instance, upcoming stringent emission regulations require a 93% reduction in NO_x emission and an 87.5% reduction in PM emission relative to present-day standards.

Although HCCI engine operation was first investigated using gasoline as the fuel [2-4], attention to diesel HCCI has been drawn since Thring's published work in 1989 [5]. The high ignitability of diesel fuel makes it compatible with the compression ignition aspect of HCCI. However, in order to overcome the low evaporability of diesel fuel, several researchers have proposed alternative approaches to implementing a diesel HCCI combustion strategy. For example, Ryan et al. developed a port-fuel injected system, where diesel fuel was injected into the air stream and an intake air heater was added to assist with the fuel vaporization [6].

The true implementation of an HCCI strategy in a diesel engine meets the same resistance found in a gasoline engine, where compression ratio and load are limited by uncontrolled auto-ignition of the fuel. Thus attention has shifted to implementing premixed compression ignition (PCI), or partially premixed compression ignition (PPCI) strategies. Significant advances in electronically-controlled, high pressure, multiple injection fuel systems have accelerated the development of diesel-fuelled PCI combustion strategies [7]. Staying true to the direct-injection feature of most conventional diesel engines, mixture preparation for PCI or PPCI usually occurs through direct injection of the fuel into the cylinder. Strategies as reported in [8], [9], [10], and [11] all use direct injection methods to prepare a premixed charge.

A number of the direct injection PCI strategies have employed early injection timings to extend the ignition delay to a long enough period for complete fuel vaporization to occur prior to combustion. For instance, one proposed strategy uses two injections; an early injection that introduces 50% of the total fuel prepares an ignitable mixture lit off by the second injection of the remaining fuel closer to top dead center (TDC) [12]. Similarly, others have demonstrated a split injection strategy where an early first pulse injected fuel 54° to 4° before TDC, with a late second pulse occurring 13° after TDC [13]. However, conventional diesel fuel injectors with cone angles of 120- to 160-degrees about the vertical centerline commonly direct fuel towards the cylinder liner at the needed early timings. Hence, unless special measures are taken, early injection strategies result in excessive spray-wall interaction.

Various attempts have been made to mitigate problems encountered with these early injection strategies. One attempt utilizes three injectors, recently upgraded to pintle-type injectors [14], to avoid spray-wall interaction. Other strategies use heavy amounts of EGR, high swirl, and late fuel injection timings (near or after TDC) to achieve PCI combustion [11, 15-16]. Another alternative uses narrow spray cone angle fuel injector nozzles (less than 100-degrees) to avoid spray-wall interaction at early injection timings [10]. This strategy shows promise to achieving diesel PPCI. Nevertheless, Docquier reported considerable in-homogeneities within the fuel/air mixture, poor fuel vaporization and high liquid fuel penetration, when using an 80-degree spray angle nozzle [17].

The present study builds upon the narrow spray cone angle concept and explores further their use in conjunction with early injection to assist with PPCI mixture preparation. Furthermore, this study attempts to eliminate the spray penetration issue by utilizing smaller diameter injector holes. A split injection strategy that resembles the double injection strategies described in [12] and [13] is adopted for this assessment. An experimental investigation of three different spray angles, complemented with computational fluid dynamics analysis (CFD), reveals that narrow spray cone angle nozzles indeed provide an opportunity to achieve simultaneous reductions in NO_x and PM, with minor penalties in fuel economy.

EXPERIMENTAL SET-UP

All of the experimental data presented in this study were collected from a 1.7L in-line, 4-cylinder diesel engine. The production counterpart of this prototype engine was developed for European passenger car applications. Current hardware configurations are capable of meeting Euro IV emission standards [18]. A common rail fuel system allows for variable pressure (up to 1600 bar), timing, and number of injections. A variable geometry turbocharger (VGT), along with an EGR valve, supports high flow rates of exhaust gas recirculation. The engine's compression ratio was decreased from the production setting of 19:1 to 16:1 for developing the combustion strategy of this study.

The engine is installed in a test cell of the General Motors Collaborative Research Laboratory at the University of Michigan. A Horiba Series 23 emissions bench provided gaseous emission measurements, while an AVL 413D smoke meter provided exhaust carbon smoke levels. The MIRA correlation provided the conversion of smoke number to fuel specific PM emission [19]. Kistler 6041A water-cooled pressure transducers provided in-cylinder pressure measurements in all four cylinders.

Since pressure data was collected from all four cylinders, reported heat release curves were calculated from cylinder averaged pressure data. In addition to the cylinder averaging, cylinder pressure was averaged for 16 consecutive engine cycles. Furthermore, engine air flow, fuel flow, and EGR rates were averaged for each cylinder.

All results reported in this study were collected at an engine operating condition of 1400 RPM and a fueling rate of 15 mm^3/stroke. This equates to a nominal load condition of BMEP=3.5 bar, which was chosen because it covers roughly 62% of the European Transient Cycle-5. All studies used diesel #2 fuel (~C_7H_{13}), with a Cetane number of 50 and a heating value of 42.9 MJ/kg.

Three injector nozzles with varying spray cone angles, i.e. 115-, 80-, and 60-degrees, were used to explore whether PPCI can be achieved using a narrow cone angle nozzle in conjunction with early injection timings. All three nozzles have a nominal flow rate of 390 cc/30s @ 100 bar. In addition to this flow rate, 60-degree nozzles with nominal flow rates of 320 cc/30s @ 100 bar were utilized to further improve the feasibility of a PPCI combustion strategy. Note that the flow rate of the 60-degree nozzle was altered by reducing the diameter of the six injection holes, with all other design details unchanged.

During evaluation of the injector nozzles, a fuel injection pressure of 1000 bar was chosen to assist with fuel atomization, mixing, and complete injection prior to burning. Exhaust gas recirculation rates varied between 35% and 45% for the different test cases. The EGR rate was used to meet a targeted NO_x value for each nozzle, thus resulting in different EGR flows. Once the target NO_x value had been met, the EGR rate remained constant for the sweep of pilot injection timing. The objective of the investigation was to explore very advanced injection timings to simultaneously obtain low NO_x (EI-NO_x < 1 g/kg-fuel) and low PM (EI-PM < 0.25 g/kg-fuel).

COMPUTATIONAL FRAMEWORK

CFD analyses were conducted to support the experimental development of using narrow spray cone angle nozzles. The experimental study seeks to establish PPCI combustion, which depends heavily on the injection and mixing processes; therefore the CFD model focuses on these particular influences.

KIVA 3-V [20-22] along with a 37,000-cell grid provided the numerical simulation of the fuel injection events within the entire cylinder. Results shown highlight a slice through the centerline of one spray plume. Roughly 1,500 parcels represent the spray droplets for each injection event; each parcel contains droplets with the same properties. Tetradecane ($C_{14}H_{30}$) serves as the fuel in the computations, which has evaporation properties that closely match diesel fuel.

The WAVE breakup model developed by Reitz and Diwakar [23-24] simulates the fuel spray breakup, using an empirically determined B_1 constant of 4.5. This constant accounts for uncertainties associated with the nozzles. The impingement model developed by Grover et al. captures the spray impingement and wall-film formation of the fuel [25]. Finally, the original KIVA evaporation model simulated the fuel evaporation. The model was calibrated and validated to the engine motoring pressure. The success of the model's prediction capability has been demonstrated on a medium-size diesel engine [26].

RESULTS AND DISCUSSION

EFFECT OF INJECTION TIMING

Injecting fuel very early in an engine cycle helps to create a PPCI combustion pattern. However, too early of an injection leads to poor fuel evaporation, piston bowl spray targeting issues, and over-advanced combustion (fuel heat release too early in the cycle). An injection-timing sweep with all three nozzle configurations provides insight into these issues and their impact on PPCI.

115-Degree Spray Cone Angle Nozzle

The first set of data highlights results from the 115-degree spray cone angle nozzles. For this set, EGR constituted 45% of the total intake charge, injection rail pressure was set to 1000 bar, and a split injection strategy was employed, where approximately 50% of the fuel was injected in a pilot. Start Of Pilot (SOP) injection timing was swept from 47° to 30° BTDC, while the remaining fuel was injected at a fixed timing of 2° BTDC. Fuel sprays begin to miss the combustion bowl at timings advanced more than 35° BTDC.

Figure 1 illustrates the percent mass burn rate for the SOP sweep with 115-degree nozzles. The percent mass burn rate is calculated from gross heat release rate, normalized by total gross heat release. As SOP advances, the amount of pre-TDC combustion decreases as shown in Figure 1, indicating a higher concentration of premixed burn after TDC. This coincides with previous observations that ignition delays increase as injection timings advance [27]. At the earliest timing of 47° BTDC, there is hardly any pre-TDC combustion. While combustion at this earliest timing is likely premixed (fuel injection is completed before burning), the high rate of heat release produces combustion temperatures high enough to yield excessive NO_x of about 4 g/kg-fuel.

Figure 2 illustrates the fuel-specific NO_x and PM levels. While PM levels are acceptable, NO_x levels are unacceptably high. This indicates that a higher EGR rate is likely needed to lower combustion temperatures.

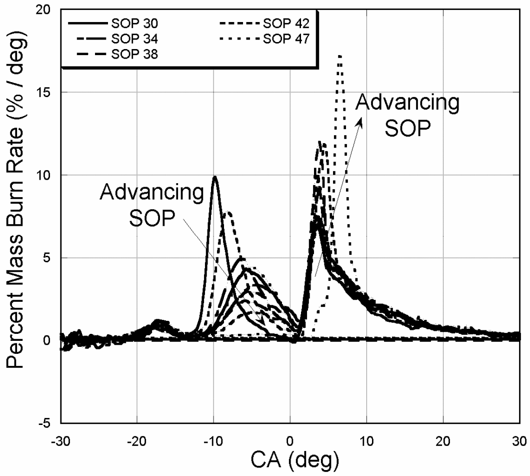

Figure 1: Percent Mass Burn Rate for 115-degree nozzles, as start of pilot (SOP) varies. EGR rate is roughly 45%, rail pressure is fixed at 1000 bar, and main injection timing is fixed at 2° BTDC.

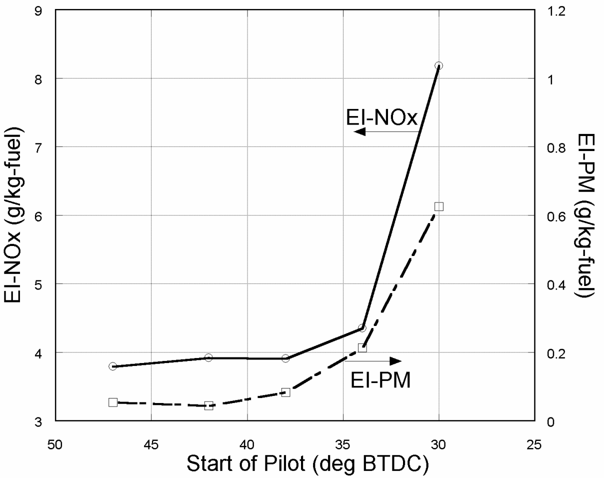

Figure 2: Specific NO_x and PM for 115-degree nozzles, as start of pilot varies. EGR rate is roughly 45%, rail pressure is fixed at 1000 bar, and main injection timing is fixed at 2° BTDC.

More importantly however, notice that both NO_x and PM decrease as SOP advances. Clearly, this injection strategy offers the potential to escape from the perennial PM-NO_x trade-off of conventional combustion modes. This strongly suggests that, with this injection strategy, PPCI combustion has been realized.

80-Degree Spray Cone Angle Nozzle

Working to satisfy the objective of this study - advance injection timing as much as possible to yield low NO_x and PM - the research proceeded to test the 80-degree spray cone angle nozzles. The tighter cone angles allowed for injections to be advanced considerably, while containing the spray within the combustion bowl (up to 50° BTDC). Thus, pilot injection timing varied from 62°

BTDC to 43° BTDC. For the following set of results, EGR rate equaled 48%, rail pressure remained fixed at 1000 bar, and the main injection timing remained fixed at 2° BTDC.

Figure 3 illustrates that the fraction of energy released in pre-TDC combustion decreases as timing is advanced; this is similar to the observation made for the 115-degree nozzle. More importantly, the peak rate of heat release decreases, and the burn rate slows down, thus extending combustion duration. Specific NO_x and PM emissions, shown in Figure 4, demonstrate the same trends as the 115-degree nozzle. In particular, both are simultaneously decreasing as SOP advances to its maximum extent. The simultaneous decrease in NO_x and PM indicates an increasing attainment of premixed burn. The overall decrease in NO_x and the overall increase in PM, relative to the 115-degree spray angle, are likely due to the slower burn rate resulting from an increased level of EGR (from 45% to 48%).

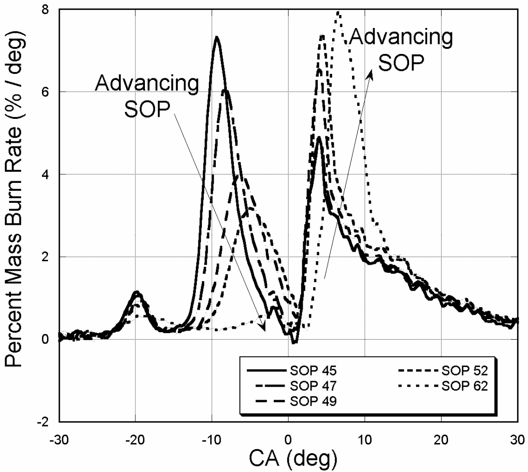

Figure 3: Percent Mass Burn Rate for 80-degree nozzles, as start of pilot (SOP) varies. EGR rate is roughly 48%, rail pressure is fixed at 1000 bar, and main injection timing is fixed at 2° BTDC.

Carbon monoxide and combustion efficiency trends, shown in Figure 5, provide insight into the combustion temperatures existing in the cylinder. The relatively low CO and relatively high combustion efficiency indicate combustion temperatures high enough to form excessive NO_x and soot. Furthermore, CO and combustion efficiency gauge the quality of combustion; fuel that misses the bowl results in higher CO and HC, thus lower combustion efficiency. Previously mentioned, the fuel spray misses the piston bowl at timings greater than 50° BTDC. As shown in the figure, EI-CO rises dramatically and combustion efficiency drops dramatically as the fuel starts to miss the bowl.

The 80-degree spray angle demonstrated the possible benefit of advancing fuel injection and creating a PPCI type combustion pattern. In spite of the fuel spray missing the bowl at advanced timings, PM and NO_x levels decreased. However, as a result of missing the bowl, products of incomplete combustion increased. Thus, exploring an even narrower spray angle, to prevent missing the bowl at advanced timings, may allow for improved PPCI combustion.

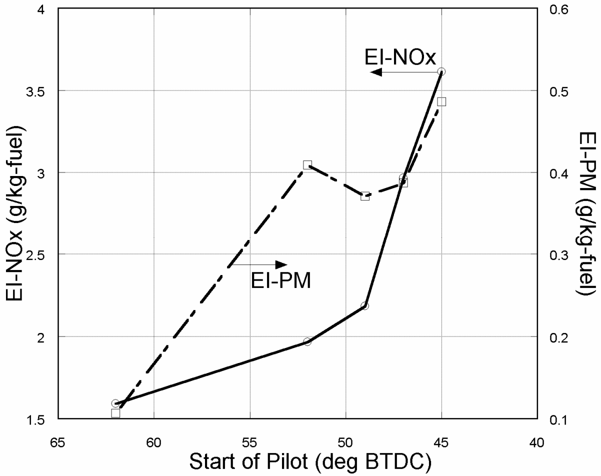

Figure 4: Specific NO_x and PM for 80-degree nozzles, as start of pilot varies. EGR rate is roughly 48%, rail pressure is fixed at 1000 bar, and main injection timing is fixed at 2° BTDC.

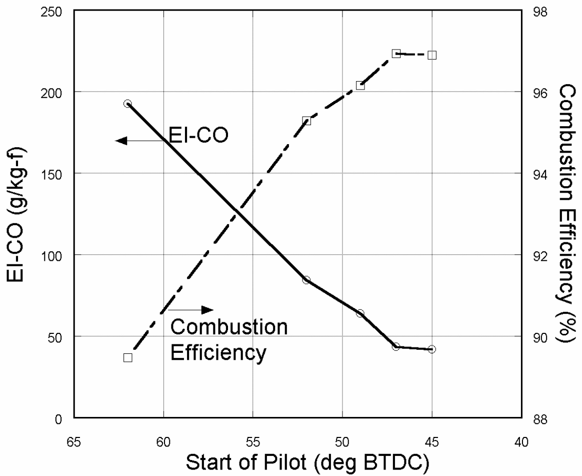

Figure 5: Specific CO and Combustion Efficiency for 80-degree nozzles, as start of pilot varies. EGR rate is roughly 48%, rail pressure is fixed at 1000 bar, and main injection timing is fixed at 2° BTDC.

60-Degree Spray Cone Angle Nozzle

The last set of nozzles explored had 60-degree spray angles, allowing for injection timings up to 65° BTDC before the fuel spray starts to miss the bowl. Similar to the 80-degree nozzles, pilot injection timing was swept from 62° BTDC to 43° BTDC. For all tests, EGR rate equaled 38%, rail pressure remained fixed at 1000 bar, and main injection timing remained fixed at 2° BTDC.

Unlike the other two sets of nozzles, Figure 6 reveals that pre-TDC combustion does not disappear at advanced timings. This feature appears to impact the

NO$_x$ and PM emissions, shown in Figure 7, where both increase beyond a timing of 47° BTDC. Perhaps a larger fraction of the second injection burns diffusively since the pre-TDC combustion remains, even at the earliest timings.

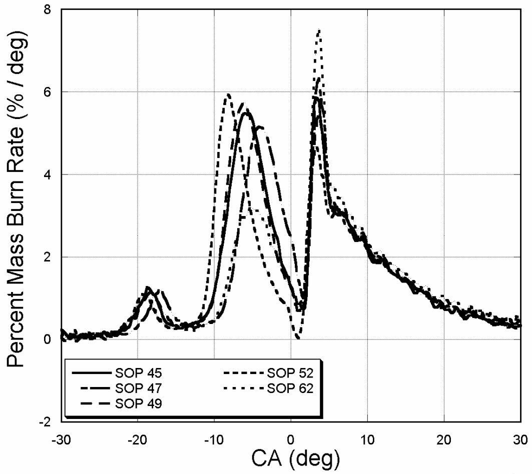

Figure 6: Percent Mass Burn Rate for 60-degree nozzles, as start of pilot (SOP) varies. EGR rate is roughly 38%, rail pressure is fixed at 1000 bar, and main injection timing is fixed at 2° BTDC.

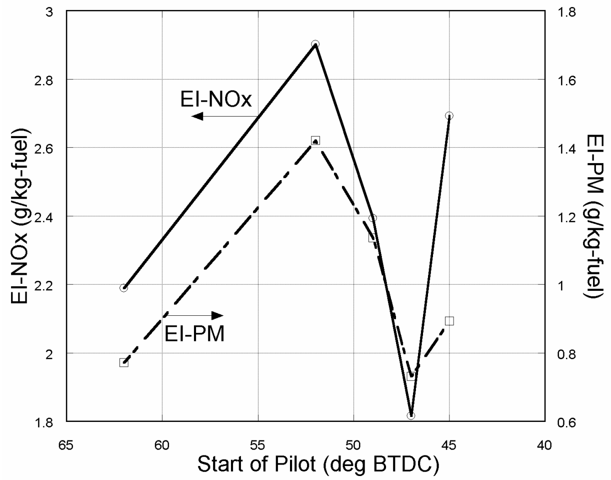

Figure 7: Specific NO$_x$ and PM for 60-degree nozzles, as start of pilot varies. EGR rate is roughly 38%, rail pressure is fixed at 1000 bar, and main injection timing is fixed at 2° BTDC.

As expected, CO and combustion efficiency have much less sensitivity to injection timing for the 60-degree nozzles, compared to the 80-degree nozzles, as shown in Figure 8. In spite of this, EI-CO is higher and combustion efficiency is lower at the NO$_x$ and PM-optimized timing of 47° BTDC for the 60-degree nozzles. Furthermore, the PM levels for the 60-degree nozzles are higher than the 80-degree nozzles. These two observations initially indicate the 60-degree nozzles hold no promise for achieving PPCI-mode emission levels.

However, the NO$_x$ levels are slightly lower for the 60-degree nozzles, relative to the 80-degree nozzles, in spite of a lower EGR rate. This observation gives hope for the 60-degree nozzles, provided that PM emissions can be improved. In the next section, CFD analysis is conducted to study the spray characteristics resulting from the two nozzle geometries and to improve upon the PM emissions from the 60-degree nozzles.

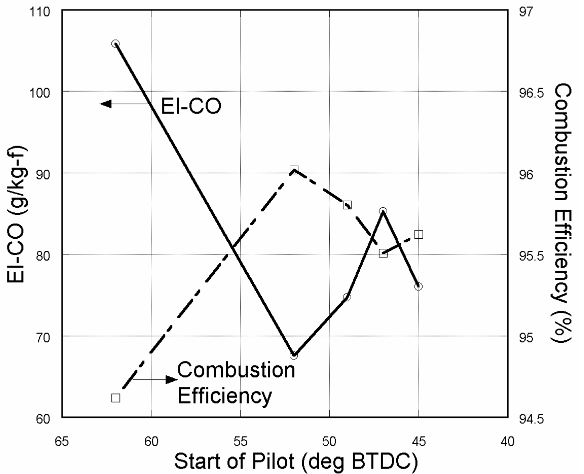

Figure 8: Specific CO and Combustion Efficiency for 60-degree nozzles, as start of pilot varies. EGR rate is roughly 38%, rail pressure is fixed at 1000 bar, and main injection timing is fixed at 2° BTDC.

CFD INSIGHT INTO 60-DEGREE NOZZLES

A Case for 60-Degree Nozzles?

As identified previously, the 60-degree nozzle offers some hope to achieving PPCI low-PM combustion since it allows improved spray targeting into the piston bowl at early timings. Despite the higher PM levels compared to the 80-degree nozzles, the NO$_x$ values were roughly the same even though EGR rate was lower for the 60-degree nozzles. Further development of the 60-degree nozzle may offer more benefit than the 80-degree nozzle, if the higher PM level issue can be resolved. To help evaluate this issue, CFD was used to compare the 60- and 80-degree nozzles.

An injection timing of 47° BTDC yielded the lowest level of NO$_x$ and PM for the 60-degree nozzle, thus warranting its use for CFD analysis. CFD images show the fuel vapor distribution 1°, 2° and 10° after SOP for both the 60- and 80-degree nozzles (both with same SOP of 47° BTDC). Based on qualitative observations, both the 60-degree nozzle, shown in Figure 9 through Figure 11, and the 80-degree nozzle, shown in Figure 12 through Figure 14, have the same vapor distribution throughout the injection process. However, one could argue that the 80-degree nozzle experiences higher wall interaction in the bowl region (see Figure 11 compared to Figure 14). This observation suggests that the 80-degree nozzle produces more PM from the first injection.

Similarly, fuel swirls outward from the piston bowl for the 60-degree nozzle (see Figure 11), while it swirls inward for the 80-degree nozzle (see Figure 14). Since swirl intensity generally increases as distance from the bowl centerline increases, the fuel core near the outside of the bowl likely mixes better than the fuel core near the inside of the bowl. Again, this suggests the first injection of the 80-degree nozzle produces more PM than the first injection of the 60-degree nozzle.

In spite of these observations, data shown in Figure 4 and Figure 7 clearly show that the 60-degree nozzle produces more PM than the 80-degree nozzle. The second fuel injection explains this discrepancy. The fuel vapor distributions at TDC for the 60-degree and 80-degree spray angles are shown in Figure 15 and Figure 16, respectively. Notice for both cone angles that the fuel impinges upon the inside bowl region, referred to as the piston pedestal. However, the 60-degree spray angle impinges on the pedestal to a greater extent.

This increased impingement likely deteriorates fluid momentum, and the level of mixing of the second injection event. Since the heat release from the first injection event raises cylinder temperatures, this second injection perhaps burns more diffusively for the 60-degree nozzle than for the 80-degree nozzle. Although combustion has not been simulated in the present CFD analysis, the momentum of the fuel stream toward the piston pedestal should not change dramatically as a result of combustion from the first injection.

Finally, the CFD analysis illustrates the 60-degree nozzle's wall film issues with overly advanced injection timings. Figure 17 shows the wall film thickness on the inside bowl region at a crank angle location of 35° BTDC when pilot injection occurred at 62° BTDC. Showing a marked difference is Figure 18, which illustrates a significantly reduced wall film thickness when pilot injection occurred later at 47° BTDC.

The increased wall wetting for the earlier timing may be explained by differences in cylinder conditions during injection. Since cylinder pressure is lower at 62° BTDC, the fuel has more momentum to travel further into the cylinder. Similarly, cylinder temperature is lower at 62° BTDC reducing the rate of evaporation. Both of these effects increase spray penetration, and thus the spray-wall interaction, at the earlier injection timings. The combustion efficiencies for both the 80-degree nozzle (Figure 5) and the 60-degree nozzle (Figure 8) demonstrate a behavioral shift for timings advanced of 47° BTDC. In spite of fuel spray still remaining in the bowl for the 60-degree nozzle, the decrease in combustion efficiency indicates that overly advanced timings may not be desirable.

The issue of wall wetting is of concern to many PCI and PPCI researchers. Iwabuchi, et al. as well as Wahlin and Cronhjort suggest the use of injector nozzles with impinging spray jets to lower spray penetration, and thus reduce the occurrence of wall wetting [28-29]. However, spray penetration also decreases with smaller injector holes. Better fuel atomization and lower fluid momentum help to reduce the fuel spray penetration, albeit requiring a longer pulse width to attain the same fuel quantity. The longer pulse reduces the amount of time between end of injection and start of combustion, however it also enhances mixing.

Therefore, based on the CFD analysis of the 60-degree nozzle, the exploratory study now shifts to focusing on a 60-degree nozzle with a lower flow rate. As described earlier, using the 60-degree nozzle offers a NO_x advantage over the 80-degree nozzle. Using the lower flow rate 60-degree nozzle improves fuel vapor distribution, thus lowering PM and assisting with attainment of the study's objective.

EFFECT OF NOZZLE FLOW RATE

In order to improve the PM levels with the 60-degree nozzles, injector tips with lower flow rates (320 cc/30s @ 100 bar) were compared against the previously studied tips (390 cc/30s @ 100 bar). To realize lower flow rates, nozzle tips with smaller holes were used. The latter improve spray breakup and atomization, and decrease fuel penetration, thus decreasing spray-wall interaction. Since the number of holes remained the same, the injection pulse width had to be lengthened to inject the same amount of fuel at the same injection pressure.

For the following data, the EGR rate equals 45% for both sets of nozzles. Rail pressure remains set to 1000 bar, and main injection timing remains set to 2° BTDC. The pilot injection timing is set to the previously determined optimal timing of 47° BTDC.

The combination of improved fuel vaporization and decreased spray penetration produces an increased level of premixed burning for the lower flow rate nozzles, as shown in Figure 19. Reductions in NO_x of 40% and PM of 45% result as a greater fraction of the fuel burns premixed, again demonstrating the ability of PPCI combustion to defy the traditional PM-NO_x tradeoff. The lower flow rate nozzles improve fuel atomization, thus promoting additional premixed burning. Similarly, the lower flow rate nozzles decrease fuel penetration, and reduce the impact of the second injection on the piston pedestal. As had been argued from the CFD analysis, this second injection hitting the piston pedestal likely resulted in diffusive burning as the fuel desorbed from the piston surface.

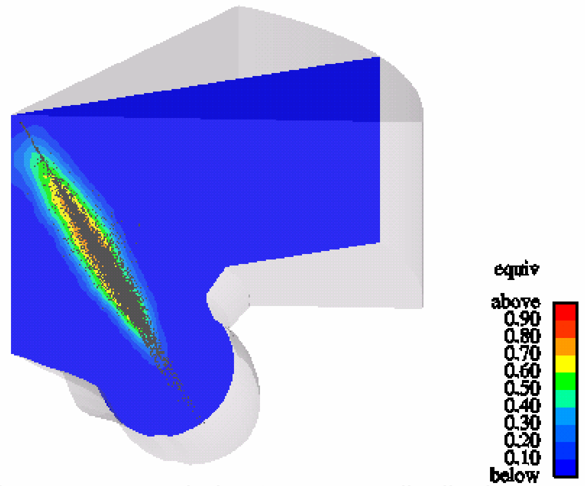

Figure 9: In-cylinder fuel vapor distribution for the 60-degree nozzle at 46° BTDC, when fuel was injected at 47° BTDC. EGR rate is roughly 38%, rail pressure is 1000 bar.

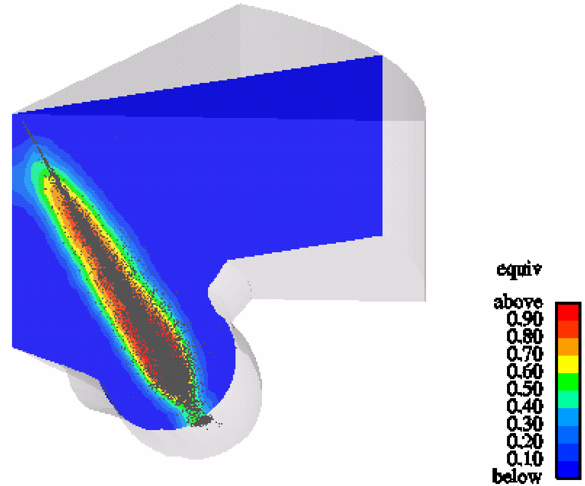

Figure 10: In-cylinder fuel vapor distribution for the 60-degree nozzle at 45° BTDC, when fuel was injected at 47° BTDC. EGR rate is roughly 38%, rail pressure is 1000 bar.

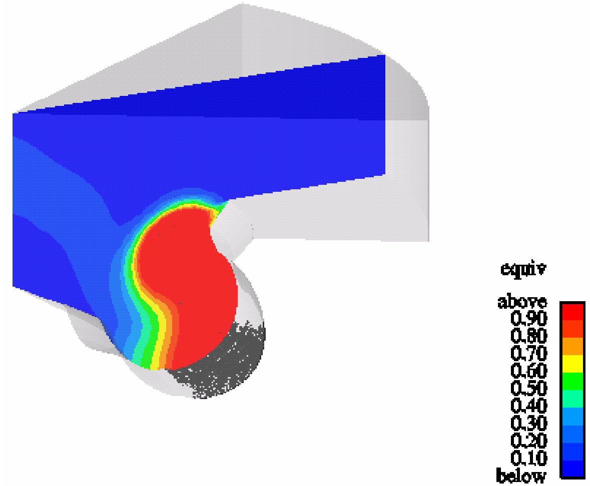

Figure 11: In-cylinder fuel vapor distribution for the 60-degree nozzle at 35° BTDC, when fuel was injected at 47° BTDC. EGR rate is roughly 38%, rail pressure is 1000 bar.

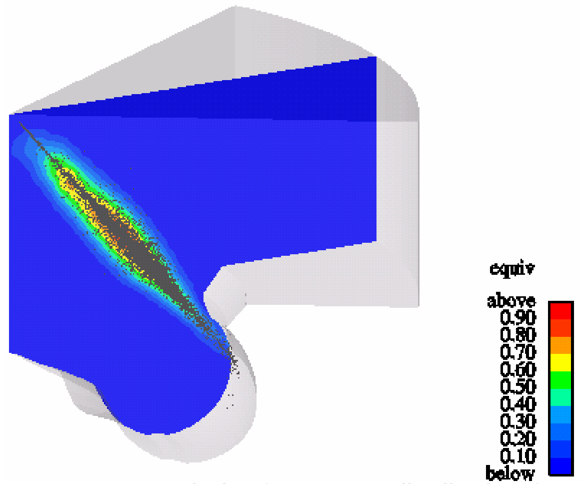

Figure 12: In-cylinder fuel vapor distribution for the 80-degree nozzle at 46° BTDC, when fuel was injected at 47° BTDC. EGR rate is roughly 48%, rail pressure is 1000 bar.

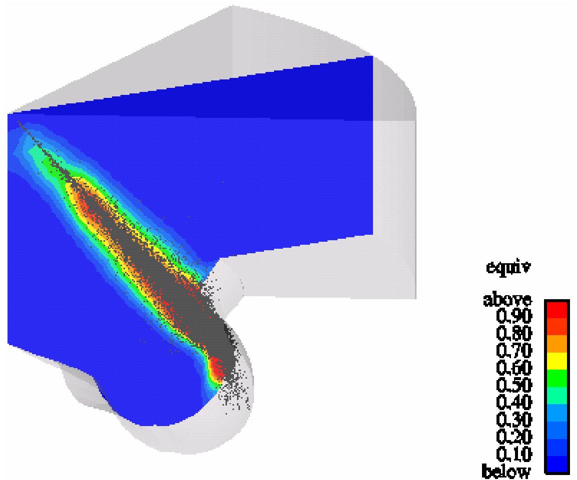

Figure 13: In-cylinder fuel vapor distribution for the 80-degree nozzle at 45° BTDC, when fuel was injected at 47° BTDC. EGR rate is roughly 48%, rail pressure is 1000 bar.

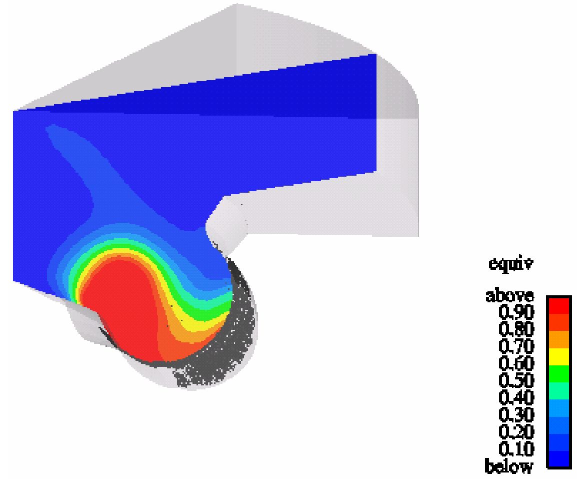

Figure 14: In-cylinder fuel vapor distribution for the 80-degree nozzle at 35° BTDC, when fuel was injected at 47° BTDC. EGR rate is roughly 48%, rail pressure is 1000 bar.

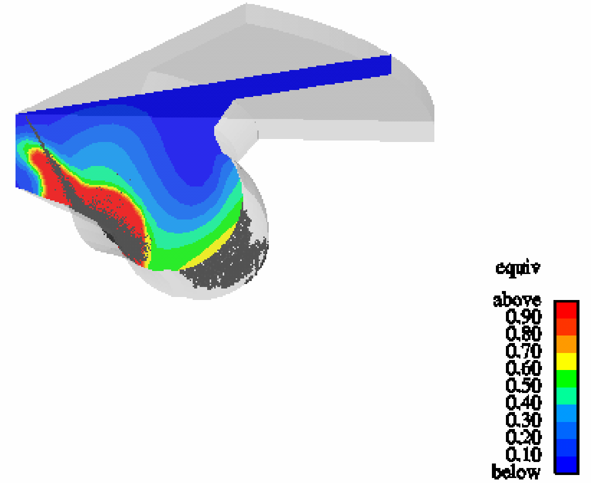

Figure 15: In-cylinder fuel vapor distribution for the 60-degree nozzle at TDC, when fuel is injected at 2° BTDC. EGR rate is roughly 38%, rail pressure is 1000 bar.

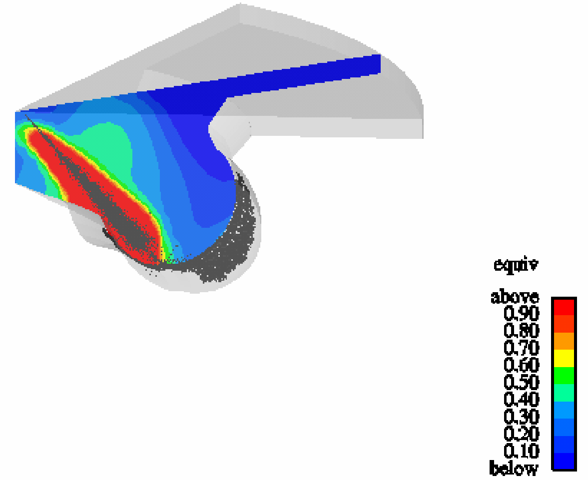

Figure 16: In-cylinder fuel vapor distribution for the 80-degree nozzle at TDC, when fuel is injected at 2° BTDC. EGR rate is roughly 48%, rail pressure is 1000 bar.

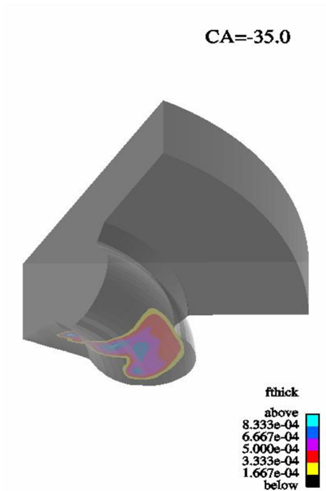

Figure 17: Fuel wall film thickness at 35° BTDC when fuel is injected at 62° BTDC for the 60-degree nozzle. EGR Rate is roughly 38%, rail pressure is 1000 bar.

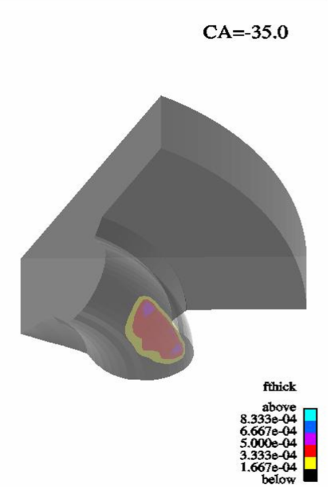

Figure 18: Fuel wall film thickness at 35° BTDC when fuel is injected at 47° BTDC for the 60-degree nozzle. EGR rate is roughly 38%, rail pressure is 1000 bar.

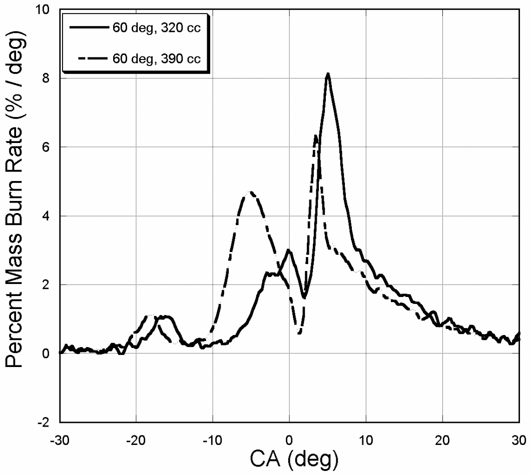

Figure 19: Percent Mass Burn Rate for 60-degree nozzles, with 320cc and 390cc flow rates. EGR rate is roughly 45%, rail pressure is fixed at 1000 bar, pilot injection timing is fixed at 47° BTDC and main injection timing is fixed at 2° BTDC.

THE PROMISE OF PPCI COMBUSTION

Now that a nozzle geometry has shown considerable promise for achieving PPCI combustion in a diesel engine, EGR rate is swept to increase ignition delay further and attain a low PM / low-NO_x combustion strategy. For all testing, pilot injection timing remains fixed to 47° BTDC and main injection timing remains fixed to 2° BTDC. Rail pressure is set at 1000 bar while EGR rate is swept from 31% to 59%.

Figure 20 illustrates the effect of EGR on burn rate for the 320cc 60-degree spray cone angle nozzle. As EGR rate increases the percentage of combustion occurring after TDC also increases. Furthermore, combustion rates slow down and combustion duration is prolonged, as more fuel energy burns in a slow-rate premixed fashion. Figure 21 demonstrates that NO_x values naturally decrease as EGR rate increases. However, after an EGR rate of 45%, Figure 21 similarly shows decreasing PM levels derailing the conventional PM-NO_x tradeoff penalty. This observation indicates that combustion has become predominantly premixed at EGR rates above 45%. Further increasing EGR rate to 55% achieves NO_x and PM levels that meet the combustion development targets for this research study.

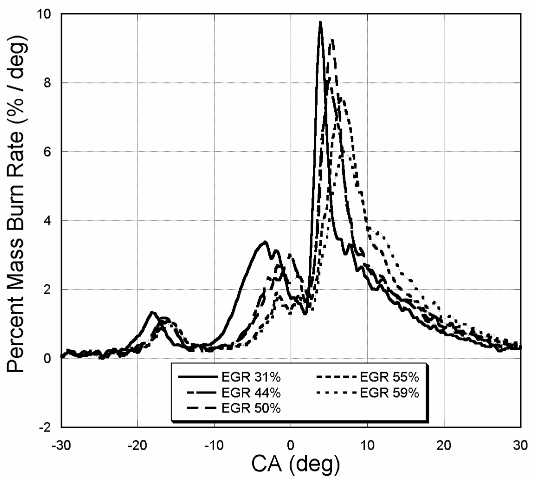

Figure 20: Percent Mass Burn Rate for the 320 cc 60-degree nozzle. Start of pilot was fixed at 47° BTDC, main injection timing was fixed at 2° BTDC, rail pressure was set to 1000 bar, and EGR Rate varied from 31% to 59%.

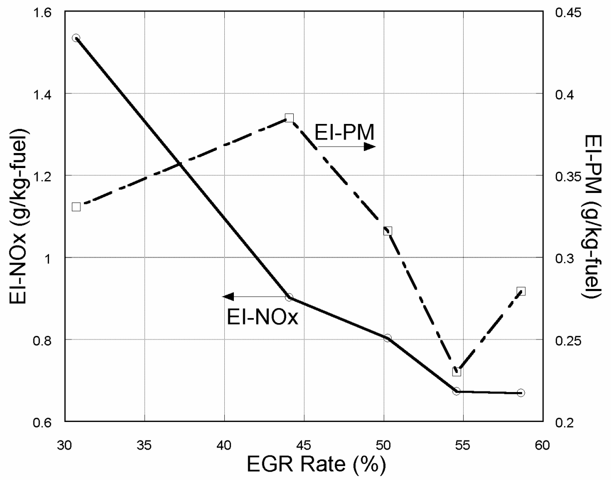

Figure 21: Specific NO_x and PM for the 320 cc 60-degree nozzle. Start of pilot was fixed at 47° BTDC, main injection timing was fixed at 2° BTDC, rail pressure was set to 1000 bar, and EGR Rate varied from 31% to 59%.

COMPARISON OF OPTIMAL COMBUSTION STRATEGY

The objective of this research study has been satisfied, as it has demonstrated that narrow spray cone angle nozzles can be effectively employed in conjunction with very advanced injection timings in order to develop PPCI combustion. The benefit of this novel combustion strategy is the simultaneous reduction of NO_x and PM. A comparison to the base engine's calibration provides a gauge to determine the novel strategy's effectiveness. The decreases in NO_x and PM emissions are clearly very attractive. Compared to the prototype engine's base calibration, an 82% reduction in NO_x has been realized. This dramatic decrease provides promise towards possibly meeting future emission regulations with purely in-cylinder methods, without relying on after-treatment NO_x catalysts. At the same time, a 39% decrease in PM can be achieved. While impressive, this decrease is not likely to meet future strict regulations on PM. Continued combustion development will certainly lower this value; however growing demand for smokeless diesel engines will probably necessitate the implementation of particulate traps.

Nevertheless, the dramatic reductions in emissions associated with the proposed PPCI strategy come at the expense of a modest fuel consumption penalty, as illustrated in Figure 22. The penalty is attributed to some combustion occurring before TDC (see Figure 20), which is not an optimal period for fuel burning. Relative to the prototype engine's base calibration, the penalty amounts to around 4.5%, which could possibly be recovered by improved EGR delivery and VGT turbocharger matching.

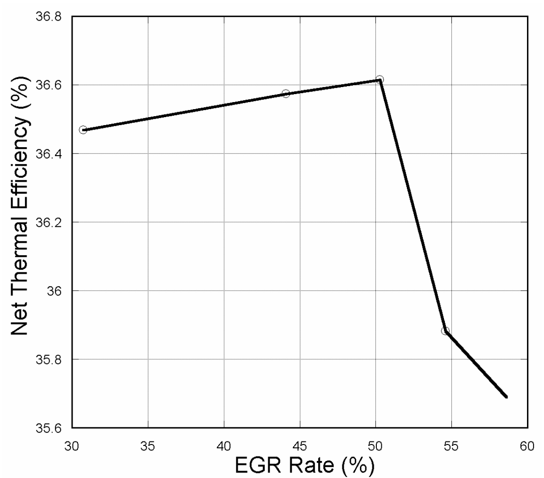

Figure 22: Net indicated thermal efficiency for the 320cc 60-degree nozzle. Start of pilot was fixed at 47° BTDC, main injection timing was fixed at 2° BTDC, rail pressure was set to 1000 bar, and EGR Rate varied from 31% to 59%.

Of course, the use of very narrow spray angles causes concern for the attainment of satisfactory performance over the engine's entire operating map. The narrow cone angles may prove particularly effective at light load PPCI conditions, but may inhibit performance at high load, conventional combustion conditions. While the extension of the PPCI regime to higher load engine conditions presents challenges, the possibility of using narrow spray cone angle nozzles to advance injection timings much earlier than traditional timings, and thus achieve PPCI combustion, offers a viable developmental strategy.

CONCLUSIONS

The present study has evaluated the potential for using narrow spray cone angle injector nozzles to avoid out-of-bowl injection with possible cylinder wall wetting issues at advanced injection timings for diesel PPCI combustion. Investigation of three different nozzles of varying spray cone angle, i.e. 115-, 80-, and 60-degrees, demonstrated differing levels of success in achieving PPCI combustion. For each nozzle, the timing of pilot injection (where 50% of the fuel was injected) was varied, but the main injection timing was kept fixed at 2° BTDC. The following conclusions can be drawn from each configuration:

- The 115-degree nozzle was studied up to injection timings of 47° BTDC. As timing advances, the amount of premixed burn increases. However, defeat of the traditional PM-NO_x tradeoff occurs as both PM and NO_x decrease when timing advances. In spite of this victory, both NO_x and PM levels exceed development goals, thus suggesting that a narrower cone angle is required.

- The 80- and 60-degree nozzles were studied up to injection timings of 62° BTDC. Again, NO_x and PM levels simultaneously decrease, demonstrating for all three nozzles PPCI-type trends, as injection timing varies. Worsening combustion efficiency for timings advanced of 47° BTDC with the 60-degree nozzles indicate that overly-advanced timings may not be desirable, in spite of the fuel spray remaining in the bowl.

- The 60-degree nozzle demonstrates higher levels of PM; however, it produces roughly the same NO_x emission as the 80-degree nozzle, despite a lower EGR rate. This latter observation supports the further development of the 60-degree nozzle.

CFD analysis identified possible causes for the increased PM levels with the 60-degree nozzle. This insight motivated the study of a lower flow rate, 60-degree nozzle. The lower flow rate nozzle, by way of smaller nozzle holes, assists spray atomization and vaporization creating more premixed burn and thus reducing the PM level. The CFD analysis also revealed considerable spray impingement on the bowl and pedestal surface, also likely elevating PM levels. The lower flow rate nozzle also decreases spray penetration, thus decreasing the occurrence of fuel-wall interaction on the bowl.

The lower flow rate 60-degree nozzle, along with optimized EGR rate, is capable of achieving the study's objective: to obtain low-NO_x and -PM levels at the engine's operating condition (1400 RPM and 3.5 bar BMEP). The resulting calibration lowered NO_x by 82% and PM by 39% at the expense of a modest increase (4.5%) in fuel consumption.

ACKNOWLEDGEMENTS

The authors of this research study would like to acknowledge the technical and financial support of the General Motors (GM) - University of Michigan (UM) Collaborative Research Laboratory (CRL) in Engine Systems Research. The GM-UM CRL provides the focal point for joint research between GM's Research and Development and UM, establishing the framework to link UM faculty and student expertise with long-term GM needs. In particular, the authors are grateful to Dr. Roger Krieger for his technical input and insights into improved combustion development. In addition to the collaborative research with GM, Isuzu's Advanced Engineering Center in Japan is thanked for technical and hardware support. Finally, the Robert Bosch Company is thanked for their support in developing fuel injection nozzles.

REFERENCES

1 Zhao, F., Asmus, T., Assanis, D., Dec, J., Eng, J., and Najt, P., *Homogeneous Charge Compression Ignition (HCCI) Engines; Key Research and Development Issues*, SAE Publication PT-94.

2 Siewert, R., "Staged Internal Combustion Engines and Methods of Operation," US Patent 3 924 576, December 9, 1975.

3 Siewert, R., and Turns, S., "The Staged Combustion Compound Engine (SCCE): Exhaust Emissions and Fuel Economy Potential," *SAE Transactions*, pp. 2371-2420, 1975.

4 Onishi, S., Jo, S., Shoda, K., Jo, P., and Kato, S., "Active Thermo-Atmosphere Combustion (ATAC) – A New Combustion Process for Internal Combustion Engines," SAE Paper 790501, 1979.

5 Thring, R., "Homogeneous-Charge Compression Ignition (HCCI) Engine," SAE Paper 892068, 1989.

6 Ryan III, T., and Callahan, T., "Homogeneous Charge Compression Ignition of Diesel Fuel," SAE Paper 961160, 1996.

7 Epping, K., Aceves, S., Bechtold, R., and Dec, J., "The Potential of HCCI Combustion for High Efficiency and Low Emissions", SAE Paper 2002-01-1923, 2002.

8 Takeda, Y., Keiichi, N., and Keiichi, N., "Emission Characteristics of Premixed Lean Diesel

Combustion with Extremely Early Staged Fuel Injection," SAE Paper 961163, 1996.

9 Yanagihara, H., Sato, Y., and Mizuta, J., "A Study of DI Diesel Combustion under Uniform Higher-Dispersed Mixture Formation," *JSAE Review*, Volume 18, pp. 247-254, 1997.

10 Walter, B., and Gatellier, B., "Development of the High Power NADI Concept Using Dual Mode Diesel Combustion to Achieve Zero NO_x and Particulate Emissions," SAE Paper 2002-01-1744, 2002.

11 Kimura, S., Aoki, O., Kitahara, Y., and Aiyoshizawa, E., "Ultra-Clean Combustion Technology Combining a Low-Temperature and Premixed Combustion Concept for Meeting Future Emission Standards," SAE Paper 2001-01-0200, 2001.

12 Yokota, H., Kudo, Y., Nakajima, H., Kakegawa, T., and Suzuki, T., "A New Concept for Low Emission Diesel Combustion," SAE Paper 970891, 1997.

13 Hasegawa, R., and Yanagihara, H., "HCCI Combustion in DI Diesel Engine," SAE Paper 2003-01-0745. 2003.

14 Akagawa, H., Miyamoto, T., Harada, A., Sasaki, S., Shimazaki, N., Hashizume, T., and Tsujimura, K., "Approaches to Solve Problems of the Premixed Lean Diesel Combustion," SAE Paper 1999-01-0183, 1999.

15 Okude, K., Mori, K., Shiino, S., and Moriya, T., "Premixed Compression Ignition (PCI) Combustion for Simultaneous Reduction of NO_x and Soot in Diesel Engine," SAE Paper 2004-01-1907, 2004.

16 Shimazaki, N., Tsurushima, T., and Nishimura, T., "Dual Mode Combustion Concept with Premixed Diesel Combustion by Direct Injection near Top Dead Center," SAE Paper 2003-01-0742, 2003.

17 Docquier, N., "Influence of Fresh Charge Preparation and Composition on Auto-Ignition Delays and Combustion Development in an Optical HCCI Direct Injection Diesel Engine," SAE Paper 2003-01-3174, 2003.

18 "Opel Touts Euro-4 Compliance with Astra 1.7 CDTI 'EcoTec' Diesel Engine", *Hart's European Fuel News*, Volume 7, Number 1, January 8, 2003.

19 Dodd, A. and Holubecki, Z. The Measurement of Diesel Exhaust Smoke. MIRA Report. Number 1965/10. 1965.

20 Amsden, A., O'Rourke, P., and Butler, T., "KIVA II – A Computer Program for Chemically Reactive Flows with Sprays," Los Alamos National Laboratory, LA-11560-MS, 1989.

21 Amsden, A., "KIVA-3V: A KIVA Program with Block-Structured Mesh for Complex Geometries," Los Alamos National Laboratory, LA-12503-MS, 1993.

22 Amsden, A., "KIVA-3V: A Block-Structured KIVA Program for Engines with Vertical or Canted Valves," Los Alamos National Laboratory, LA-13313-MS, July 1997.

23 Reitz, R., and Diwakar, R., "Structure of High-Pressure Fuel Sprays," SAE Technical Paper 870598, 1987.

24 Reitz, R., "Modeling Atomization Processes in High-Pressure Vaporizing Sprays," *Atomisation and Spray Technology*, Volume 3, Pgs. 309-337, 1987.

25 Grover, R., Assanis, D., "A Spray Wall Impingement Model Based Upon Conservation Principles," *The Fifth International Symposium on Diagnostics and Modeling of Combustion in Internal Combustion Engines (COMODIA 2001)*, July 1-4, 2001.

26 Hong, S., Wooldridge, M., Im, H., Assanis, D., Kurtz, E., "Modeling of Diesel Combustion, Soot and NO Emissions Based on a Modified Eddy Dissipation Concept," Submitted to *Combustion Science and Technology* December 6, 2004.

27 Lyn, W., "Study of Burning Rate and Nature of Combustion in Diesel Engines," *Proceedings of the Combustion Institute*, Ninth Symposium (International) on Combustion, Academic Press, New York, pp. 1069-1082, 1963.

28 Iwabuchi, Y., Kawai, K., Shoji, T., and Takeda, Y., "Trial of New Concept Diesel Combustion System – Premixed Compression-Ignited Combustion," SAE Paper 1999-01-0185, 1999.

29 Wahlin, F., and Cronhjort, A., "Fuel Sprays for Premixed Compression Ignited Combustion – Characteristics of Impinging Sprays," SAE Paper 2004-01-1776, 2004.