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FINAL REPORT

Characterization of Single-Cylinder Small-Bore 4-Stroke CIDI Engine Combustion

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Executive Summary

This final report covers the research conducted at the Center for Automotive Research (CAR) at Wayne State University (WSU) to characterize the combustion process in an advanced Compression Ignition Engine (CIDI). The program was sponsored by U. S. Department of Energy, OTT, OFCVT, SUPERCAR and the FREEDUM CAR programs, managed by Gurpreet Singh, technically directed by Dr. Dennis Siebers at Sandia National Lab. (SNL) with Dr. Paul Miles at SNL as the principal investigator. The DOE funding was for the period from 12/1/1999 to November 11/30/2004. Currently, the research on the CIDI is in progress under the technical and financial support of U. S. Army Research Office, Research Triangle Park, N.C. and National Automotive Center (NAC) Automotive Research Center (ARC).

The characterization of combustion in the CIDI program had three interacting research projects running in parallel. One project was conducted on an optically accessible engine at SNL with Dr. Paul Miles as the P.I. The second project was conducted at WSU on an engine of the same design and dimensions as the optical engine at SNL, referred to as the metal engine, with Drs. Naeim A. Henein and Ming-Chia Lai as Co P.I.s. The third project was conducted at University of Wisconsin for the CFD KIVA simulation of the combustion process in the CIDI engine.

The goal of the program is to gain a better understanding of the spray behavior under high injection pressures in small-bore, high compression ratio, high-speed, direct-injection diesel engines equipped with advanced fuel injection system. The final results demonstrate the capability of the engine in reducing the engine-out emissions and improve the trade-off between nitrogen oxides (NOx), particulate matter, other emissions and fuel economy.

The results of the research conducted at WSU, SNL and UW have been presented and discussed at CRADA meetings held every six months at SNL, Livermore, CA and at USCAR, South field, MI. The results of the research conducted at WSU have been presented in many technical meetings and published in fifteen papers at Society of Automotive Engineers (SAE) and American Society of Mechanical Engineers (ASME).

This report describes the result of the investigations conducted at WSU CAR on the CICD metal engine. The experiments covered a wide range of injection pressures, injection timings, pilot, main and late injection, EGR rates in the conventional and low combustion temperature regimes to the misfiring limit and swirl ratios up to 7.2. Trade-off maps between the NOx and smoke, measured in Bosch Smoke units (BSU) are developed in 2-D showing the iso-bar and iso-EGR lines. Also, 3-D trade-off maps are developed to show the synergistic effects of injection pressure, and swirl ratio on NOx, BSU, CO and HC emissions. The iso-EGR surfaces developed in the 3-D maps demonstrated the strong impact of increasing EGR on reducing NOx and the injection pressure on reducing BSU and their relative effects compared to the effect of swirl ratio.

Furthermore, the investigation covered the effect of injection pressure, EGR, injection timing on the low temperature combustion regime (LTC) where the increase in EGR rate causes a reduction in both the smoke and NOx. A detailed analysis is made on the problems associated

with the LTC regime developed by Nissan in 2001known as Modulated Combustion (MK). The problems of the MK regime are related to combustion instability significant increase in HC, CO and specific fuel consumption.

This investigation resulted in the development of strategies referred to as **(OPERAS)**, to **O**ptimize injection **P**ressure, **E**GR, injection timing Retard, injection timing **A**dvance and **S**wirl ratio, Comparison is made between the effects of OPERAS and MK regime on the trade-off between NOx and BSU, HC, CO, fuel economy and combustion instability.

A summary of the research conducted at WSU is given in this report and the major accomplishments are stated. The details of the research are published in fifteen technical papers included at the end of this report.

During last year, a new Advanced Low Temperature Combustion Regime (ALTC) regime is developed to improve combustion instability and reduce the penalty in HC, CO and specific fuel consumption. (ALTC) strategy has been validated by experiments on the CIDI metal engine. The results will be presented at 2007 SAE Congress meeting.

In addition to the publications included in this report, more details of the research findings are available at WSU CAR and can be obtained by contacting N. A. Henein, WSU College of Engineering, Detroit, MI 48202, Tel: (313) 577- 3887, Fax: (313) 577-8789, email: henein@eng.wayne.edu.

I. INTRODUCTION

Direct injection diesel engines power most of the heavy-duty vehicles. Due to their superior fuel economy, high power density and low carbon dioxide emissions, turbocharged, small bore, high speed, direct injection diesel engines are being considered to power light duty vehicles. Such vehicles have to meet stringent emission standards. However, it is difficult to meet these standards by modifying the in-cylinder thermodynamic and combustion processes to reduce engine-out emissions. After-treatment devices will be needed to achieve even lower emission targets required in the production engines to account for the anticipated deterioration after long periods of operation in the field. To reduce the size, mass and cost of the after-treatment devices, there is a need to reduce engine-out emissions and optimize both the engine and the after-treatment devices as one integrated system. For example, the trade-off between engine-out NOx and PM, suggests that one of these species can be minimized in the engine, with a penalty in the other, which can be addressed efficiently in the after-treatment devices.

Controlling engine-out emissions can be achieved by optimizing many engine design and operating parameters. The design parameters include, but are not limited to, the type of injection system: (CRS) Common Rail System, (HEUI) Hydraulically Actuated and Electronically controlled Unit Injector, or (EUI) Electronic Unit Injector; engine compression ratio, combustion chamber design (bowl design, reentrance geometry, squish area and intake and exhaust ports design. With four-valve engines, the swirl ratio depends on the design of both the tangential and helical ports and their relative locations. For any specific engine design, the operating variables need also to be optimized. These include injection pressure, injection rate, injection duration and timing (pilot, main, and post injection), EGR ratio, and swirl ratio.

This report introduces a new phenomenological model for the fuel distribution and combustion, and emissions formation in the small bore, high speed, direct injection diesel engine. This will be followed by an analysis of the effect of each of injection pressure, EGR, injection advance and retard and swirl ratio on engine-out emissions and fuel economy. A discussion will be given on the 2-D and 3-D trade of maps. Finally a discussion will be made on the low temperature combustion regimes, its major problems and proposed solutions.

II. EXPERIMENTAL SET-UP

The engine used in this investigation is a single-cylinder, diameter = 79.5 mm, stroke, 85 mm, compression ratio = 20, 4-valve, direct-injection, four-stroke-cycle, water-cooled, diesel engine equipped with a common rail fuel injection system. The engine is supercharged with heated shop air, and the exhaust backpressure is adjusted to simulate actual turbo-charged diesel engine conditions. The engine has four valves. Closing the intake tangential port to different degrees by using a gate valve, varied the swirl ratio. Position 90° is for the valve in the full open position. The data is recorded in terms of valve position, rather than the swirl ratio, in order to be able to reproduce the same swirl ratio and valve opening in different runs. Engine specifications are given in Appendix and the details of the experimental setup are given in the attached publications.

Tests matrix

Nozzles : 430 VCO, 390 Mini-Sac, 320 Mini-Sac

Test Points:	RPM	IMEP(Bar)	MAP(Bar)
KP1	900	1.2	1.0
KP2	1500	3.0	1.2
KP3	2000	5.0	1.4

Injection Pressures: 400, 600, 800, 1000, 1200 Bar

Injection Modes: Main, Pilot-Main, Main-Post

EGR ratios: 0%, 25%, 35%, 40%, 45%, 50%, 55, 60%, 62% and 64%

Swirl Ratios: 1.5 – 7.2

Two fuels were used in the experiments reported in this document: AMOCO diesel fuel (DF2) and low sulfur fuel. The fuel specifications are given in Appendix B. The particulate matter was measured by using a mini dilution tunnel, filtration and a microbalance. Also soot was measured by using Bosch Smoke meter and the results are in Bosch Smoke Units (BSU).

III. MAJOR ACCOMPLISHMENTS

III. 1. DEVELOPMENT OF A NEW PHENOMENOLOGICAL MODEL FOR COMBUSTION AND EMISSION FORMATION IN SMALL BORE DIESEL ENGINES

Combustion and emission formation in diesel engines has been the subject of several investigations. Such investigations include imaging in optically accessible engines and mathematical modeling. More recently, a model has been developed at Sandia National Laboratory, for combustion and emission formation in heavy-duty diesel engines. The model is based on images taken in an optically accessible single-cylinder diesel engine, and laser-based measurements of the concentration of different autoignition and combustion intermediate species. The model explained the role of the fuel distribution in the spray and the local equivalence ratio in autoignition, combustion and the formation of NOx and soot.

In small-bore diesel engines, the conditions are different from those in heavy-duty diesel engines from many aspects. The first is the shorter spray penetration before the fuel reaches the wall in the small bore engine. Here, the fuel impingement and the liquid film formation on the wall have a significant effect on the combustion process. The second is the relatively strong swirling motion needed in the small bore engine to mix the fuel and air. The third difference is the smaller piston bowl diameter as a fraction of the cylinder diameter in the small bore engine and the resulting squish component, which might be absent in the large bore engine.

In the study of engine-out emissions in small-bore direct injection diesel engines, particularly those using high-pressure injection systems, it is important to follow the spray as if moves into the combustion chamber and determine the time it reaches the wall, the time the flame is established in the chamber, and whether the fuel is injected in the flame or reaches the wall as a vapor or liquid. Further more, it is important to find out the effect of the different operating parameters on the fuel distribution in the combustion chamber and its subsequent effect on combustion and engine-out emissions.

III.1. (A) Fuel Distribution In The Combustion Chamber Of Small-Bore Diesel Engines

Figure 1 shows a line sketch of the combustion chamber of the CIDI engine used in this investigation. Before describing the fuel distribution in the combustion chamber, reference is made to the images taken by Paul Miles at Sandia National Lab. on an optically accessible version of the engine used in the current investigation. These images taken every crank angle degree showed clearly that a sizeable flame was developed in the combustion chamber at the peak of the apparent rate of heat release (ARHR). Based on these observations and others in the literature we can divide the fuel delivered in the combustion chamber into two major parts, each of which undergoes different processes and contributes in the engine-out emissions in a different way.

The fuel distribution in the combustion chamber can be obtained from the rate of fuel delivery and rate of heat release shown in figure 2. For this figure, the engine test conditions are: nozzle: 320 mini sac, fuel injection pressure = 600 bar, IMEP = 5 bar, engine speed = 2000 rpm, swirl ratio= 1.55, EGR = 0, and LPP (Location of the peak cylinder pressure): (6° - 7°) aTDC. The rate of fuel delivery was computed from the needle lift and the pressure drop across the nozzle holes. The rate of heat release (RHR) from the combustion reactions is computed from the cylinder pressure, accounting for the heat transfer and blow by losses.

The first part is the fuel that is injected prior to the time when the flame is established in the combustion chamber, [F_{pf}]. Here, the lighter components, [F_{pfv}], evaporate, mix with the air, autoignite and burn in the premixed combustion mode. The heavier components deposit on the wall [F_{pfw}], and burn in the mixing-controlled, or diffusion-controlled combustion mode.

The difference between [F_{pf}] and [F_{pfw}] represents the fuel in the combustion chamber that does not reach the wall and is referred to [F_{trans}].

The second part is the fresh fuel injected in the flame [F_{if}] that is exposed to fairly high gas temperatures. Here the fuel may decompose forming soot, or in the presence of air, burns in the mixing-controlled combustion mode or the diffusion-controlled combustion mode.

The third part is the fuel delivered near the very end of the injection process, as the needle is about to close [F_t]. This is a very small amount that is poorly atomized and may decompose, or burn in a diffusion-controlled combustion mode.

The fuel distribution in the combustion chamber, under the operating conditions of Figure 2 can be described as follows:

(a) Fuel injected prior to the flame, evaporated and burned [F_{pfv}]

The fuel delivery starts at 352.0°. As the spray travels into the combustion chamber, the light components evaporate and mix with the surrounding air and deflect along the wall. This is the early stage of combustion where autoignition nuclei are formed at different sites in the spray envelope and produce flamelets that burn the surrounding combustible mixture. Figure 2 shows the premixed combustion took 2.4 CAD (0.264 ms) to reach an energy release rate of 53.57 J/CAD, due to the combustion of 35.14% of the fuel. As discussed earlier, the flame is established in the combustion chamber at 360.5°.

The total period of the premixed combustion is about 4.3 CAD (0.47 ms). The local quality of the fuel vapor-air mixture determines the emissions formed during this early stage of combustion. The premixed combustion fraction will have a local air/fuel ratio that ranges from zero to infinity. The mole fractions of the different gaseous emission species at different excess air factors would be close to those in the spark ignition engines, as illustrated in Figure 3. Previous investigations on diesel combustion, without EGR, showed that the dominant emission in the premixed charge is NO_x. The mass of the emissions formed at the end of this stage depends on the mass of the charge that burns in the premixed mode.

(b) Fuel injected prior to the flame and deposited on the walls [F_{pfw}]

Previous investigations on heavy duty diesel engines showed that during the early stages of diesel combustion, the core of the spray remains in the liquid phase and its length from the nozzle is known as the liquid length In the current small bore engine, under similar operating conditions, the liquid core was found to reach the wall. The spread of the liquid film on the wall depends mainly on the speed of the spray, which in turn depends on the injection pressure.

In order to determine [F_{pfw}] in this investigation, the spray penetration during the injection period is calculated from the pressure drop across the nozzle holes and the flow area, calculated at different values of the needle lift. Figure (2) shows that the first element of the spray arrives at the wall at 356.5°. The heavier parts of the fuel continue to be deposit on the wall up to 360.6°, before the flame is established in the combustion chamber. It is noticed that the period of time the spray takes to arrive at the wall varies during the injection process, due to the changes in the injection pressure, cylinder gas density and needle opening area.

The rate of evaporation of the fuel deposited on the wall depends on the relative velocity between the gas phase and the liquid fuel film, the gas temperature as well as the wall temperature. The gas motion in the combustion chamber depends on the fuel spray velocity and the interaction between the swirl and squish components.

The wall temperature plays an important role in the rate of liquid fuel evaporation. At light loads, the wall temperatures are relatively low, allowing the heavier fuel components to remain on the wall and evaporate late in the expansion stroke. Upon evaporation, the heavier molecules may evaporate and appear as exhaust hydrocarbons, or they may be partially oxidized and produce aldehydes if the gas temperature is low, or they may incompletely burn at higher temperatures and produce CO and soot.

2. Fresh Fuel injected into the flame [F_{if}]

The fresh fuel injected after the peak of energy release rate evaporates and burns at a fast rate as it mixes with the hot combustion products in the presence of the flame. This is the part of the fuel that has the least chance to penetrate into the combustion chamber and be exposed to the high velocity swirling air near the periphery of the piston bowl.

The emission species formed in this mode of combustion varies with the load. At light loads there is a chance that the evaporated fuel will be completely oxidized and thus contribute to NO_x formation. However, at higher loads, the chances for the complete oxidation of this part of the fuel decrease, and the result is the formation of incomplete combustion products including carbon.

<u>3. Fuel delivered near the very end of the injection process</u> [F_T]

During the closing of the injector needle, the flow area decreases, and the flow coefficient drops to a point where the fuel is not well atomized. This part of the fuel will be referred to as the spray tail. Microscopic images taken very close to the nozzle hole, showed the fuel flowing out of the nozzle hole and forms ligaments. This anatomized fuel has a small chance to evaporate, mix with the cylinder gases and completely burn. [F_T] is believed to be a source of unburned hydrocarbons and soot emissions.

III.1.B. Engine-Out Emissions In Small Bore High Speed Direct Injection Diesel Engines

The following analysis is based on the above discussions for the fuel distribution in the combustion chamber, and is aimed at identifying the different parameters that affect the formation of the different emission species.

NO_x Emissions

Many expressions have been developed to calculate NO_x formation during engine combustion, based on both the formation and decomposition reactions in the extended Zeldovitch mechanism. Assumptions, usually made to arrive at an expression for NO concentration, include the equilibrium concentration or the steady state approximation for the nitrogen and/or oxygen atoms. In this paper, in order to simplify the discussions, NO decomposition reactions will not be considered and the steady state approximation for N will be assumed. Under these conditions, the NO formed over the period of time from the start of combustion to the opening of the exhaust valve can be given by:

$$NO = \int_{SOI}^{EVO} \int_{0}^{V} A \omega K_{O_2} \exp\left(\frac{-E}{\bar{R}} \frac{}{T} \right) [F]^a [O_2]^b \, dV dt \tag{1}$$

where
A, a and b: Constants
ω: Engine instantaneous angular velocity
K_{O2}: Equilibrium constant for O_2 which is a function of temperature.
E: Activation energy

T: Local gas temperature
[F] : Fuel vapor concentration
[O_2]: Oxygen concentration.
V: Volume of the reacting gases
t: time

Equation (1) shows that engine-out NO_x concentration is a function of oxygen concentration, fuel vapor concentration and residence time, in addition to the combustion products temperature. All these parameters change with the engine load, injection pressure P, EGR, retarding or advancing injection timing, and swirl.

Soot Emissions

Engine-out soot emission is the result of its formation and oxidation.

$$\frac{d[S]}{dt} = \frac{d[S_f]}{dt} - \frac{d[S_o]}{dt} \qquad (2)$$

Equation (2) is integrated over the whole period from the start of combustion to the time the exhaust valve opens.

The following are the sources of soot formation, based on the fuel distribution in the combustion chamber, described earlier:

1.Liquid fuel injected in the flame [F_{if}].
2.Fuel injected, evaporated and premixed with air and formed a rich mixture.
3.Fuel injected and premixed with air diluted with EGR.
4. Fuel film on the wall [F_{pfw}].

The rate of soot formation can be given by (26)

$$\frac{d[S_f]}{dt} = K_f [F_v] P^{0.5} \exp\{-\frac{E_f}{\bar{\bar{R}}T}\} \qquad (3)$$

E_f = 12,500 cal/mole

where [F_v] is the vapor concentration

The rate of soot oxidation can be given by (26)

$$\frac{d[S_o]}{dt} = K_o [S] X_{O_2} P^{1.8} \exp\{-\frac{E_o}{\bar{R}T}\} \qquad (4)$$

E_o= 14,000 cal /mole

where [S] is the soot concentration

The extent of soot oxidation depends on the following principal controlling parameters : oxygen concentration, charge temperature, mixing rate and a certain period of time. During the expansion stroke, the gas temperature drops and there is a window during which the mass average gas temperature is high enough to render effective oxidation reactions. Many operating variable affect the residence time within this window.

Note, the activation energy for the oxidation reactions is higher than that for the formation reactions. Accordingly, in the presence of enough oxygen, the oxidation reactions are favored as the temperature increases.

Hydrocarbon Emissions

The hydrocarbons in diesel exhaust are originated from the following :

1. The very lean and rich flameout regions of the premixed charge in the spray.
2. The liquid fuel film deposited on the wall.
3. Spray tail, where the fuel in not well atomized.

Similar to soot, HC can be oxidized later in the expansion stroke, if they are mixed with the oxidants, at a temperature high enough to render effective oxidation reactions.

CO Emissions

CO is related to the incomplete oxidation of the hydrocarbons. This occurs wherever there is a rich mixture. Also, CO can be formed from the incomplete oxidation of the HC late in the expansion stroke.

III.2. CORRELATION BETWEEN NOX and BSU AT DIFFERENT EGR RATES

Figures (4, 5 and 6) show the following general characteristics of engine-out emissions in the form of trade-off maps for NOx and BSU at different injection pressures and EGR rates:

(1) The constant pressure lines, referred to as iso-bar lines, indicate that the increase in EGR is effective in reducing NOx emissions, at almost negligible penalty in soot emissions at low EGR rates. However, as EGR rates increase the penalty in soot formation increases at a much higher rate.

(2) The constant EGR lines, referred to as iso-EGR lines, show that the increase in injection pressure at low EGR rates reduces soot, but causes a penalty in NOx. However, the penalty in NOx decreases at the higher EGR rates.

(3) The increase in injection pressure is very effective in reducing soot emissions at all EGR rates.

III.3. EFFECT OF INJECTION PRESSURE, EGR RATE AND SWIRL RATIO ON ENGINE-OUT EMISSIONS

The following results are for the engine running at KP3, IMEP = 5 bar, engine speed =2000 rpm, cylinder gas peak pressure location (PPL) = 6° - 7° aTDC, EGR = 0% , 25%, 35% and 45%, swirl ratio = 1.44 – 3.5.

III.3.A. Effect on NOx

The combined effects of injection pressure, EGR rate and swirl ratio on NOx are shown in 3-D maps given in Figure (7). The iso-EGR surfaces shown in the map indicate the following characteristics:

(1) The increase in swirl ratio at zero EGR has a minor effect on NOx up to a point after which NOx increases with the increase in swirl ratio.
(2) At 25% EGR the effect of the increase in swirl ratio is the same as that at 0% EGR.
(3) At higher EGR rates (35% and 45%) the increase in swirl ratio has a minor effect on NOx.

It is important to notice the relative effects of the increase in EGR and swirl ratio on NOx. Figure (7) shows clearly that EGR rate is the major parameter that can reduce NOx at any injection pressure and swirl ratio.

III.3.B. Effect on BSU

The combined effects of injection pressure, EGR rate and swirl ratio on BSU are shown in 3-D maps given in Figure (8). The iso-EGR surfaces shown in the map indicate the following characteristics:

(1) The effect of the increase in swirl ratio on reducing BSU is significant at the low injection pressures.
(2) This effect becomes less significant at higher EGR rates.
(3) The lowest value of BSU is achieved at the highest injection pressure and close to the highest swirl ratio.

III.3.C. Effect on CO

The combined effects of injection pressure, EGR rate and swirl ratio on CO are shown in 3-D maps as shown in Figure (9). The iso-EGR surfaces shown in the map indicate the following characteristics:

(1) CO increases at high swirl ratios under all injection pressures at zero EGR. At higher EGR rates, the increase in swirl reduces CO to a point after which any further increase in swirl produces higher CO.
(2) CO decreases at higher injection pressures at the low swirl ratios at zero EGR. At high EGR rates, CO drops with the increase in pressure at low swirl ratios is to a point after which CO increases again.
(3) At high injection pressures and swirl ratios CO increases

Figure (9) shows that there is an optimum injection pressure and swirl ratio that give the lowest CO at different EGR rates. The reasoning for such behavior is currently under investigation.

III.3.D. Effect on HC

Figure (10) shows the combined effects of injection pressure and swirl ratio at different EGR rates on HC emissions. The iso-EGR surfaces in the map indicate the following characteristics:

(1) HC emissions increase at a fairly low rate with the increase in EGR from 0% to 35%. But, an addition of 10% EGR rate causes a significant increase in HC emissions.
(2) Similar to CO, there is a combination of injection pressures and swirl ratios that produce the lowest HC emissions.

The reasoning behind the changes in HC with injection pressure and swirl ratio is currently under investigation.

III.4. CORRELATION BETWEEN NOX AND BSU AT DIFFERENT INJECTION TIMINGS

Figure (11) shows the cylinder pressure and rate of heat release at different injection timings. Later injection timings reduce the cylinder gas pressure and the rates of heat release. The effect of retarding injection timing on the trade-off between NOx and soot emissions is illustrated in Fig. (12). It is noticed that at a constant EGR rate, retarding the timing reduces both NOx and soot emissions all the way till misfiring occurs. Also, at any injection timing, increasing EGR rate reduces NOx at a slow rate up to a point after which any further increase in EGR causes a significant increase in soot emissions all the way till misfiring occurs.

III. 5. DIESEL COMBUSTION REGIMES DIFFERENTIATED BY EGR RATES

Different combustion regimes have been under investigation to reduce engine out emissions without, or with minimum, penalty in engine performance and fuel economy. These regimes include the smokeless lean combustion (MK concept), the low temperature smokeless rich combustion and the UNIBUS system, and low-temperature, non-sooting combustion. This report covers the conventional and the low temperature lean combustion regime (LTC). Figure (13) shows the effect of increasing EGR rate on moving from the conventional diesel combustion regime to the low temperature combustion regime and the misfiring zone. The effect of increasing EGR rate on ISNOx, BSU, ISHC and ISCO in these regimes is as follows:

(a) Conventional Diesel Combustion:

In the conventional diesel combustion NOx emissions decreased continuously with the increase in EGR, while BSU increased at a slow rate up to an EGR rate of 55%. A further increase in EGR to 60% caused a significant increase in BUS. Moderate changes are observed in ISFC as EGR increased to 50%, after which ISFC and CO increased sharply.

(b) Low Temperature Combustion (LTC):

LTC is shown between 60% EGR and 64% EGR, where BSU dropped sharply, while the already low NOx dropped to low values. ISFC and CO increased at an accelerating rate caused by incomplete combustion.

(c) Unstable Operation and Misfiring Zone:

An increase in EGR from 64% to 66% resulted in unstable engine operation, large cycle-to-cycle variations and occasional misfiring.

Figure (14) shows the trade off between NOx and BSU in the conventional diesel combustion and low temperature combustion regimes.

Figure (15) shows the trade-off between the ISFC and NOx in the conventional diesel combustion and MK LTC regime. It is clear that the increase in EGR from 60% to 64% resulted in a penalty of 11% in specific fuel consumption.

Figure (16) shows the trade-off between NOx and CO in the conventional diesel combustion and and MK LTC regime. It is clear that CO increased at higher rates as EGR rates higher than 56%.

III.6. COMPARISON BETWEEN THE EFFECTS OF INCREASING INJECTION PRESSURE AND THE SWIRL RATION ON THE TRADE-OFF BETWEEN NOX AND BSU

Figure (17) summarizes the effects of injection pressure and swirl ratios on the conventional diesel and LTC regimes. In the conventional diesel combustion increasing the injection pressure or the swirl ratio reduces the BSU with a penalty in NOx. The penalty decreases and diminishes at EGRs above 45%. This trend is reversed in the low temperature combustion regime, where the increase in injection pressure or swirl ratio reduces both the BSU and NOx. Figure (17) shows that increasing the injection pressure by a factor of 2, from 600 bar to 1200 bar, reduced BSU by 80%. Meanwhile, increasing the swirl ratio from 1.44 to 4.94 reduced BSU by 66%.

III.7. EVALUATION OF THE LOW TEMPERATURE COMBUSTION REGIME (LTC) MODULATED KINETICS (MK) STRATEGY

In the LTC regime MK strategy combustion takes place at low combustion temperatures to reduce the soot and NOx formation. To reduce the combustion temperature excessive amounts of EGR are used, the injection timing is retarded and high swirl ratios are applied. The effect of retarding the injection timing on the trade-off between NOx and BSU is shown in Figure (18). Figure (18) shows that retarding the injection timing and applying high EGR rates brings engine operation close to the misfiring limit (ML).

In summary, the operation of the engine in the LTC regime MK strategy increases specific fuel consumption and HC emissions and cause a significant increase in CO emissions.and brings the engine operation close to the misfiring limit where the operation is unstable.

III.8. STRATEGIES TO MINIMIZE ENGINE-OUT EMISSIONS

Figure (19) shows the trade-off between ISNOx and BSU at IMEP = 3.0 bar, engine speed = 1500 rpm, swirl ratios varying from 1.44 to 7.12, EGR rates = 48% t0 64%. At these high EGR rates the increase in swirl ratio causes a drop in both NOx and BSU. The drop in NOx becomes insignificant at the highest EGR rates in the LTC regime. The trade-off map in the small dark area of the figure near zero NOx and zero BSU is enlarged and given in Fig.(20). Figure (20) shows the different OPERAS that can be applied to reach almost a smokeless and NOxless engine operation.

IV. CONCLUSIONS

These conclusions are based on an experimental investigation on a single cylinder research diesel engine, equipped with a common rail injection system. Tests were conducted under simulated turbocharged conditions at three key points that represent idling, city driving and high way driving. The investigation covered a wide range of injection pressures from 600 bar to 1200 bar, EGR rates from 0% to 64% up to the misfiring limit, injection timings up to the misfiring limit and swirl ratios from 1.44 to 7.12. The fuels used include commercial DF2 fuel and low sulfur California diesel fuel.

1. A new phenomenological model is developed to account for the behavior of the spray in the small bore high speed direct injection diesel engine and considers fuel evaporation, fuel impingement on the wall, fuel injected in the flame and the tail of the spray. This model is different than models published in the literature for heavy duty, large bore, and quiescent chamber direct injection diesel engines where the fuel does not reach the wall and the swirl motion is not accounted for.
2. The new model can explain the effect of the injection pressure, EGR rate, injection timing and swirl ratio on the premixed combustion fraction, and the mixing-controlled and diffusion-controlled combustion fractions and their effect on the engine-out emissions.
3. New 2-D trade-off maps are developed in this investigation between NOx and BSU with the iso-EGR lines and iso-bar lines that show clearly the characteristics of the small-bore CIDI engine in the conventional diesel combustion and LTC regimes.
4. The 2-D trade-off maps showed the effects of the injection pressure and EGR on the engine-out emissions:
    - a. Increasing injection pressure at low rates of EGR reduces BSU with a penalty in NOx.
    - b. The penalty in NOx caused by the increase in injection pressure, decreases with the increase in EGR and becomes negligible at the high EGR rates.
    - c. Increasing EGR rate up to 25% is effective in reducing NOx, with a small penalty in BSU
5. New 3-D figures are developed to show the effects of injection pressure and swirl ratio on each of NOX, BSU, HC and CO. The iso-EGR surfaces developed in the figures show the relative effects of the injection pressure and swirl ratio on the engine-out emissions:
    - a. There is an optimum swirl ratio at which each of the engine-out emissions reaches a minimum. The optimum swirl ratio varies with EGR and injection pressure.
    - b. Minimum smoke is reached at the high swirl ratios and injection pressure at all the EGR rates.
    - c. Minimum NOx is reached at the highest EGR ratio.
    - d. Increasing injection pressure is more effective than increasing the swirl ratio on reducing the smoke emissions.
6. The major difference between the conventional and LTC regimes is in the % EGR applied to the fresh charge. Increasing EGR reduces NOx continuously in both regimes. However, increasing EGR increases engine-out soot emissions in the conventional diesel combustion, to a point where it peaks. Any further increase in EGR brings the engine in the LTC regime where any increase in EGR reduces soot to a level still higher than that at 0% EGR. This has been observed at all the injection pressures used in this investigation.

7. In the LTC regime, combustion is very sensitive to small variations in EGR, which is a few percentages from the misfiring EGR limit.
8. Soot, at the high EGR rates in the two regimes, can be reduced by applying high injection pressures and a moderate increase in the swirl ratio. There is an optimum swirl ratio beyond which any increase would result in a penalty in BSU.
9. There is a penalty in fuel economy and a fairly high increase in CO at the higher EGR rates particularly in the LTC regime. The penalties reported in this investigation do not reflect the additional energy required to produce high injection pressures, the increase in the cooling losses and drop in the volumetric efficiency at the high swirl ratios.
10. **O**ptimizing injection pressure **P**, EGR **E**, **R**etarding injection timing, **A**dvancing injection timing and **S**wirl ration **(OPERAS)** can be achieved by using the 2-D trade-off maps and the 3-D figures developed in this investigation. The maps and 3-D figures can help in developing engine control strategies to meet different emission goals.

V. RECOMMENDATIONS

1. Conduct investigations at advanced injection timing in the low temperature combustion regime to improve fuel economy, reduce the HC and CO emissions and improve combustion instability.
(Note: This has been completed last year and the results will be published at SAE 2007 convention) .
2. Conduct investigations to find out if the characteristics found for the CIDI metal engine are generic to small bore HSDI engines. This requires the development of a data bank for all the available test results conducted under DOE and other governmental institutions to maximize the benefits to industry and eliminate the waste of data obtained over many years.
(Note: This is progress at WSU)
3. The experimental investigation on the metal engine dealt with a limited number of operating variables. Other operating conditions and loads need to be investigated.
4. Combustion control strategies should consider the engine and after treatment devices as one system. The engine should deliver the most suitable feed gas composition for the operation of the after treatment devices.

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VII. A. APPEDIX I

FUEL PROPERTIES:

DIESEL 2007 EMISSION CERT FUEL
LOT 3FPUL701

TESTS	RESULTS	SPECIFICATIONS	METHOD
Specific Gravity, 60/60	0.8406	.84 - .865	ASTM D-4052
API Gravity	36.83	32-37	ASTM D-1298
Sulfur, ppm	8.1	7-15	ASTM D-5453
Flash Point, ºF, PM	170.5	130 Min	ASTM D-93
Pour Point, ºF, PM	-15	Report	ASTM D-97
Cloud Point, ºF	0	Report	ASTM D-2500
Viscosity, cs 40 ºC	2.2	2.0 Min	ASTM D-445
Carbon, wt%	86.72	Report	Phillips
Hydrogen, wt%	13.28	Report	Phillips
Net Heat of Combustion btu/lb	18478	Report	ASTM D- 3338
Particulate Matter (mg/l)	11.6	15 max	ASTM D-2276
Cetane Index	46.0	40-50	ASTM D-976
Cetane Number	47.1	40-50	ASTM D-613
HYDROCARBON TYPE, VOL%			**ASTM D-1319**
Aromatics	27.5	27 – 32	
Olefins	1.7	Report	
Saturates	70.8		
SFC Hydrocarbons, wt%	31.3	Report	ASTM D-5186
Polynuclear Aromatics, wt%	9.44	Report	ASTM D-5186

VII. B. APPENDIX II

ENGINE SPECIFICATIONS

- Single Cylinder CLR Crankcase
- Cylinder Head and Piston designed by Ricardo
- 4 valve
- Bore: 79.5 mm
- Stroke: 85 mm
- Compression Ratio: 20:1
- Connecting Rod: 179mm
- Common Rail Injection
- Rated Speed (max) (rev/min)= 4000
- Valve System: 2IN/2Exh
- Combustion Chamber: Bowl-in-piston sign Flat head, Central located Nozzle
- Cooling System: External Water Pump
- Lubricating System: External Oil Pump

- Injection System: Bosch Common Rail(max pressure 1350bar)
- Injection Controller: Fiat CPU controller
- Swirl Ratio- 1.44 to 8.94 (Ricardo)

VIII. LIST OF TECHNICAL PUBLICATIONS

- **1.** N. A. Henein, A. Bhattacharyya, J. Schipper, A. Kastury and Walter Bryzik "Effect of Injection Pressure and Swirl Motion on Diesel Engine-out Emissions in Conventional and Advanced Combustion Regimes" SAE 2006-01-0076
- **2.** N. A. Henein, I.P. Singh, L. Zhong, Y. Poonawala, J. Singh and Walter Bryzik " A Phenomenological Model for Combustion and Emissions in Small Bore, High Speed, Direct Injection Diesel Engines" ICES2005-1024
- **3.** N. A. Henein, A. Bhattacharyya, J. Schipper and Walter Bryzik **"** Combustion and Emission Characteristics of a Small Bore HSDI Diesel Engine in the Conventional and LTC Combustion Regimes", SAE 2005-24-045
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- **5.** Zhong, N.A. Henein and W. Bryzik "A new Predictive ID model for advanced high speed direct injection diesel engines" IMECE 2004-892
- **6.** I.P. Singh, L. Zhong, Ming-Chai Lai, N. A. Henein, and Walter Bryzik "Effect of nozzle hole geometry on a HSDI diesel engine- out emissions", SAE 2003-01-0704
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- **8.** N.A. Henein, I.P. Singh, L. Zhong, M.C. Lai and W. Bryzik "New Integrated "O.P.E.R.A.S." Strategies for Low Emissions in HSDI Diesel Engines ", SAE 2003-01-0261
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- **11.** N.A. Henein, M.C. Lai, I.P. Singh, L. Zhong and J. Han, **"**Characteristics of a common-rail diesel injection system under pilot and post-injection modes", SAE 2002-01-0218
- **12.** Pai-Hsiu Lu, Joong-Sub Han, Ming-Chia Lai, Naein A. Henein and W. Bryzik "Combustion visualization of DI diesel spray combustion inside a small-bore cylinder under different EGR and swirl ratios", SAE 2001-01-2005
- **13.** K. J. Richards, M. N. Subramaniam, R. D. Reitz, M.-C. Lai, N. A. Henein and P. C. Miles, **"**Modeling the effects of EGR and injection pressure on emissions in a high- speed direct-injection diesel engine", SAE 2001-01-1004
- **14.** Naeim A. Henein, Ming-Chia Lai, Inderpal Singh, Dahai Wang and Liang Liu**, "**Emissions trade-off and combustion characteristics of a high-speed direct injection diesel engine", SAE 2001-01-0197
- **15.** Joong-Sub Han, T.C. Wang, X. B Xie, M.C. Lai, N.A. Henein, D.L. Harrington, J. Pinson and P. Miles, "Dynamics of multiple-injection fuel sprays in a small-bore HSDI diesel engine", SAE 2000-01-1256

IX. TECHNICAL PUBLICATIONS