

Combustion Characteristics of a 3000 Bar Diesel Fuel System on a Single Cylinder Research Engine

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ABSTRACT

Modern diesel systems have come to rely on fuel systems with the capacity for high injection pressures. The benefits of such high pressures include improved tolerance for EGR, reduced emissions and improved performance. Current production fuel systems have typical capacities to 2500 bar, when a decade ago 1800 bar was a typical limit. Following the trend, this paper investigates the effect of rail pressures up to 3000 bar on a 1.5L single cylinder research engine. The injector nozzles tested include two variations in flow rate, the number of holes, and spray cone angle. In addition to fuel rail pressure, the effects of intake swirl, excess-air ratio, EGR, and injection timing are evaluated at speed and load points representative of A100, B100, and C100 test conditions of the U.S. EPA on-highway 13 Mode test cycle.

CITATION: Palanisamy, M., Lorch, J., Truemner, R., and Baldwin, B., "Combustion Characteristics of a 3000 Bar Diesel Fuel System on a Single Cylinder Research Engine," *SAE Int. J. Commer. Veh.* 8(2):2015, doi:10.4271/2015-01-2798.

INTRODUCTION

Increasingly stringent emission legislation, while reducing GHG emissions, is the primary driver behind new technologies and advancing the capabilities of existing systems in the commercial vehicle market segment. The emission standards for the US2010 on-road and the Final Tier 4 off-road commercial engine power ratings are summarized in Table 1. In addition to these pollutant standards, greenhouse gas (GHG) standards are placing further pressure on both the vehicle and the engine.

Table 1. Emission Standards* Relevant to the Commercial Engine Displacement Investigated

Emission	Unit	US2010 On-Road	Tier 4 (2011 full compliance)
CO	g/kW-h (g/hp-hr)	20.79 (15.5)	3.5 (2.6)
NMHC	g/kW-h (g/hp-hr)	0.188 (0.14)	0.19 (0.14)
NOx	g/kW-h (g/hp-hr)	0.27 (0.2)	0.4 (0.3)
PM	g/kW-h (g/hp-hr)	0.0134 (0.01)	0.02 (0.015)

Manufacturers are to meet these standards while still providing value to the end customer. Part of this challenge is finding the best tradeoff between engine-based technologies, including the fuel injection equipment (FIE), and after-treatment technologies [1].

The generation of emissions in diesel engines is a process dominated by the in-cylinder mixing of the fuel spray with the air charge. This process is heavily influenced by engine operating parameters including the fuel injection pressure and cylinder charge density. Increases in fuel injection pressure are one way to improve the performance of the mixing process [2]. Another method to reduce emissions and fuel consumption is to apply variable charge motion [3].

Figure 1 indicates typical EGR levels to meet the 2010 on-road NOx emission limit. With the high levels of EGR, soot production increases because of the reduced oxygen percentage and lower temperatures in the cylinder. Increased EGR tolerance and reduced soot emissions are the main drivers for higher injection pressure.

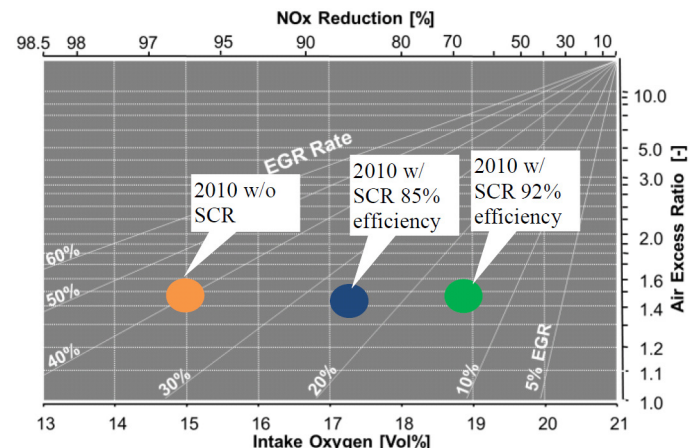


Figure 1. EGR Requirements for 2010 Emission Levels

The FIE must be capable of rapid response for multiple injections, and close control of the fuel rates over a wide range of injection pressures [4]. In low load engine operation, lower pressures are required to minimize the parasitic loss incurred by the fuel pump. At higher loads, this loss may be acceptable as a result of improved combustion performance and/or reduced emissions.

Spray chamber investigations have been conducted [5,6] to study the effects of spray impingement and hole size at elevated pressures. Engine testing by Herrmann et al. [7] have included advanced injectors, pumps and controls on current multi-cylinder engines and determined the benefit of the higher pressure. The benefit of this approach is that the additional pumping power requirements are included. However, the potential impact of optimizing the combustion chamber and air handling equipment has not been taken into account.

The current investigation began in 2010 as a cooperation between AVL, Michigan Technological University, and Delphi Diesel Systems. The goal of the project was to make a detailed investigation of the impact of ultra-high injection pressures on diesel combustion. The first step was to characterize the flow in a combustion chamber with representative, multi-hole injectors. The first tests characterized non-vaporizing and vaporizing sprays [8]. The spray characteristics were then correlated to existing theoretical and empirical models [9].

This step in the investigation is the evaluation of the FIE in a single cylinder research engine. This paper documents the work completed to date. The paper includes descriptions of the engine and test cell, and the injector and nozzle configurations being evaluated. The results from the test data are presented and discussed.

Future work will focus on integrating the results of the spray chamber work and the single cylinder research engine work through computational fluid dynamics. The integration will focus on applying the detailed spray physics, identified in the chamber tests, to the engine environment. Once the simulations capture the engine performance, further investigations of the spray characteristics and combustion chamber will be performed.

EXPERIMENTAL SETUP

Test Cell Specifications

The engine was installed in AVL PEI's Ann Arbor facility, the engine being mounted on an AVL 579 isolating base and connected to an AVL AFA-E 250 dynamometer. Figure 2 is a schematic diagram showing the test cell layout.

The intake manifold is instrumented with a Resistive Temperature Device (RTD) and pressure transducer, channels T_IM and P_IM respectively.

Intake air is provided by the facility compressors which deliver air to the test cell at approximately 650 kPa. This pressure is reduced by a regulator followed by a throttle valve, both of which are closed-loop controlled by the AVL PUMA control system to maintain the desired intake pressure. The intake air passes through a 2" Coriolis flow meter, through the electric heater and into a 227 liter buffer tank, which is connected to an intake plenum through 3.5" tube. Air then passes through the intake plenum and into the combustion chamber.

Exhaust gas exits through a single pipe manifold, into a plenum, then into a 60 liter exhaust buffer tank, through a backpressure valve and into the exhaust stack. Sample gas for smoke measurement is drawn prior to the buffer tank, while EGR gas is drawn from the buffer tank itself. The emissions sampling point is downstream of the backpressure valve. The backpressure valve is electrically controlled and adjusted by the operator to achieve the desired backpressure. EGR rate is controlled by varying the EGR valve and the backpressure valve positions.

High speed indicating data was acquired by IndiCom. The system used an AVL GU22CK cylinder pressure transducer, Kulite LP11DA pressure transducer mounted in the intake manifold and a water-cooled AVL GU21C pressure transducer mounted in the exhaust manifold.

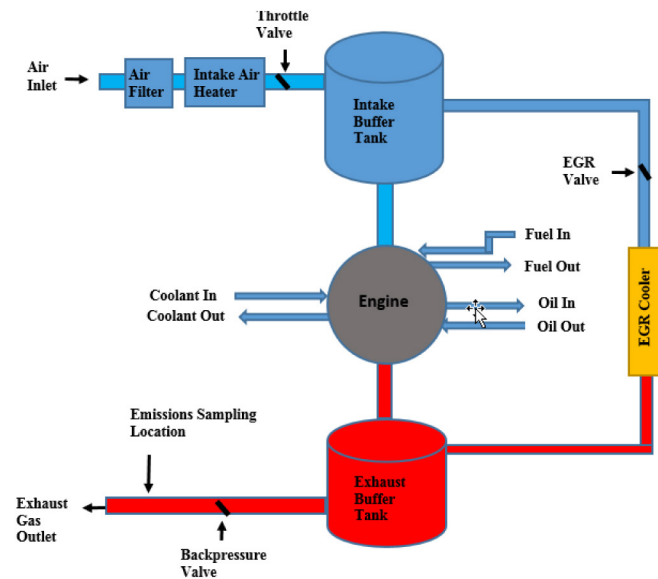


Figure 2. Test Cell Configuration

Base Engine Specifications

This project used an AVL 530 single cylinder research engine (SCRE) base fitted with an AVL-designed cylinder head, cylinder block and piston typical of modern on-road engines. Table 2 summarizes the engine specifications.

The cylinder head has four valves, two intake and two exhaust. One intake port is of a quiescent design while the other is of a swirl design. Flaps at the entry to each intake port allow the flow through

each port to be varied. The intake swirl level can be varied from 0 to 3.5. The flap positions required for a given swirl number are determined by reference to flow curves generated from a flow bench using the paddle wheel. This configuration means that synergies between the FIE and variable charge motion can be investigated. Additional flow losses are observed based on the flap settings. For this study, the boost pressure was adjusted to maintain airflow. As a constant pressure difference was maintained between the intake and exhaust manifolds, the increased flow losses were not significant to this study.

Table 2. Engine Specifications

Properties	Value	Unit
Engine	AVL 530	-
Engine Type	4-Stroke	-
Combustion System	Diesel	-
Boosting System	Externally boosted	-
Fuel Injection System	Delphi CR 3000 bar, solenoid	-
Valve Configuration	SOHC 4-valve	-
Engine Configuration	SCE	-
Displacement	1.5	L
Bore	122	mm
Stroke	128	mm
Compression Ratio	16	-
Connecting Rod Length	239	mm
Piston Pin Offset	0	mm
Rated Power	60	kW
Rated Speed	2600	Rpm
Peak Torque	300	Nm
Peak Torque Speed	1600	Rpm
Fuel	ULSD pump fuel	-

The engine is controlled through a Mototron® modular control system. This synchronizes to the engine via cam and crank sensors and controls injection timing, quantity and pressure. An inductive current clamp on the injector wiring is used to determine injection timing through IndiCom by the test cell operator, who can then make adjustments through the AVL PUMA interface. Injection quantity is adjusted by the operator varying the injection duration until the desired fuel quantity measured by the fuel balance is reached. Injection pressure is maintained through closed-loop control by the Mototron® controller. The boosted intake air system and EGR circuit, both of which are part of the test cell hardware, are also independently controlled.

The fuel injection system for this project was supplied by Delphi as part of their collaborative input. The system is largely production based and are standard except for the nozzles fitted to the injectors.

The high pressure pump is driven from the end of one of the engine balancer shafts. Fuel is fed from the pump to a vertically-orientated fuel rail which in turn feeds into a horizontally mounted rail which then supplies the injector. The two rails were implemented due to the height of the engine and to avoid long pipe lengths and small rail

volumes leading to dynamic pressure wave effects. A pressure sensor is fitted in the lower rail and a pressure control valve is fitted in the upper rail. Figure 3 shows the layout of the fuel injection system.

The accessories (oil pump, coolant pump) are run independent of the engine and the frictional losses are not included in the brake specific values. Hence the indicating data from the single cylinder engine will be comparable to a multi cylinder engine but not the brake specific values.

Several injectors and nozzle configurations were procured to evaluate the impact of the injector characteristics on the combustion and emissions from the engine. The specification of the injectors evaluated for this report are provided in Table 3.

Table 3. Injector Characteristics

Injector	Hole #	Spray Cone Angle (°)	Flow (cc/30 sec)
1	8	147	516
2	8	155	645
3	8	155	525

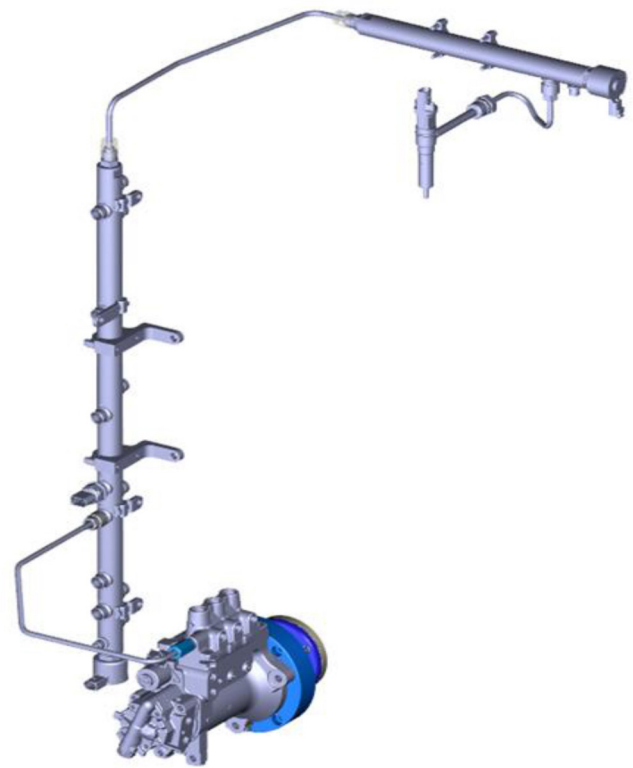


Figure 3. Layout of the Fuel Injection Equipment

Injector hole diameters were specified in such a way that the duration is about 35 Crank Angle Degrees (CAD) at desired rail pressures. Injectors 1 and 3 were sized at 3000 bar and injector 2 was sized at 2000 bar. Figure 4 and Figure 5 indicate the spray patterns at TDC and 20 degrees after TDC respectively.

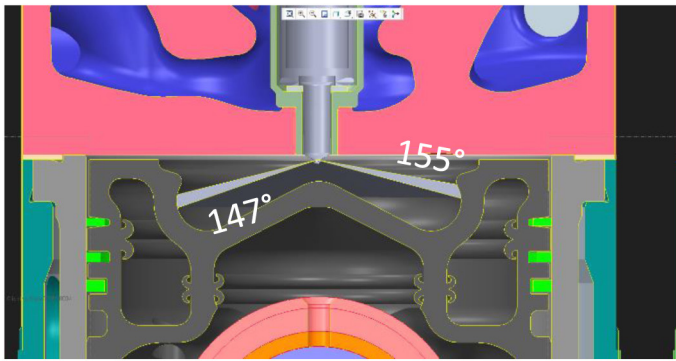


Figure 4. Spray Pattern Comparison at TDC

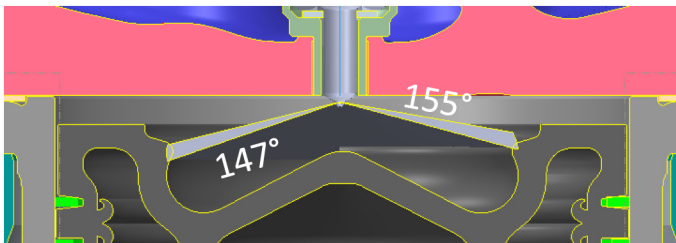


Figure 5. Spray Pattern Comparison at 20 deg after TDC

SWIRL SWEEP

Swirl increases the rate of air entrainment of the fuel spray. The result is a decrease in local fuel rich zones which can be responsible for excessive smoke in diesel engines. Increase in swirl also increases the temperature within the combustion chamber as a result of faster rate of mixing and fuel burn [10].

$$\text{Fuel Specific NOx} \left(\frac{\text{g}}{\text{kg}} \right) = \frac{\text{NOx Mass Flow}}{\text{Fuel Mass Flow}}$$

The main objective of the swirl sweep tests was to determine the impact of swirl level on the emissions and fuel consumption tradeoffs at rail pressures of 2000 bar and 3000 bar for the three different nozzle configurations. The swirl sweeps were conducted at 20% EGR and at constant air to fuel ratio (i.e boost pressure was slightly increased with an increase in swirl in order to compensate for the flow losses at the intake port, which in turn resulted in constant intake air mass flow). Fuel flow was held constant at 14 kg/hr, corresponding to a C100 operating condition. Injection timing was adjusted for each injector to achieve a peak firing pressure of 190 bar and then held constant for subsequent points. Differential pressure between intake and exhaust was held constant for all the three injectors in order to keep the pumping losses constant, irrespective of the variation in boost pressure.

Figure 6 and Figure 7 indicate the variation in NOx, smoke and fuel consumption tradeoffs for three injector configurations with varying swirl for two different rail pressures.

Table 4. Swirl Sweep - Test Plan

Swirl Sweep - Test Plan	
Operating Point	C100 (2200 rpm / 14 kg/hr fuel flow)
DeltaP	25kPa (P_ Exhaust - P_ Intake)
Boost Pressure	Varies
Rail Pressure	2000 bar / 3000 bar
EGR Rate	20%
Fuel Flow	14 kg/hr
Air flow	270 kg/hr
Injection Timing	Adjusted to achieve Peak Firing Pressure of 190bar for each injector at the start & held constant for subsequent points.

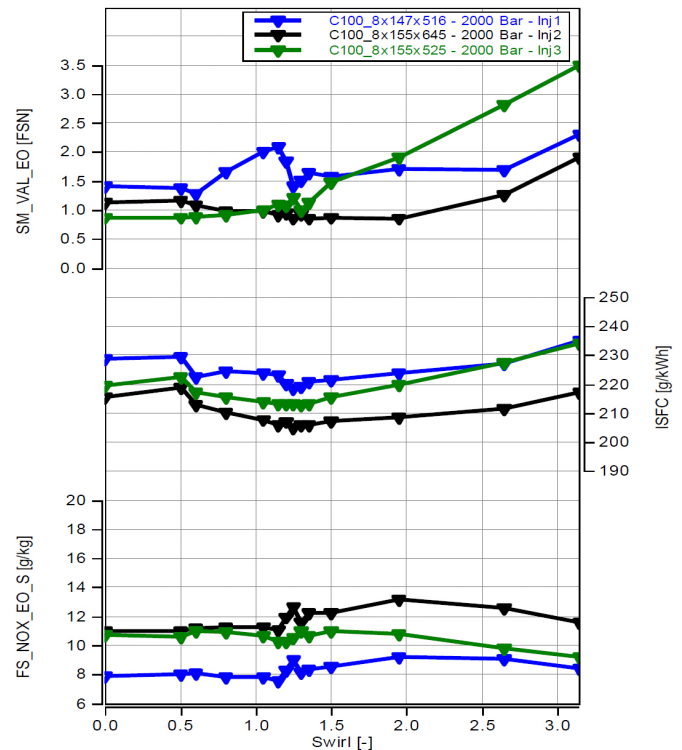


Figure 6. Effect of Swirl on Emissions & Fuel Consumption Tradeoffs - 2000 bar Rail Pressure

Effect of Spray Angle

In reviewing the results for the impact of swirl, or pressure, the spray angle influences must be discussed. Injector 1 has a steeper spray angle (147 degrees) than injectors 2 or 3 (155 degrees). Figure 6 indicates that for a similar flow rate (injectors 1, 3), the steeper spray angle results in slower combustion. This is evidenced by the higher smoke emissions and fuel consumption, with lower NOx emissions. Injectors 2 and 3, with the same spray angle, show similar trends to each other. At the higher rail pressure, Figure 7, the steeper spray angle shows less sensitivity to swirl (on comparing the smoke emissions). Figure 8 shows a comparison of the cylinder pressure traces and Rate of Heat Release (ROHR) for the swirl variations of injector 1. The ROHR is very consistent with a noticeable increase in duration only at the highest intake swirl level tested, 3.15.