

Experimental Investigation of Intake Condition and Group-Hole Nozzle Effects on Fuel Economy and Combustion Noise for Stoichiometric Diesel Combustion in an HSDI Diesel Engine

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ABSTRACT

The goal of this research is to investigate the physical parameters of stoichiometric operation of a diesel engine under a light load operating condition (6~7 bar IMEP). This paper focuses on improving the fuel efficiency of stoichiometric operation, for which a fuel consumption penalty relative to standard diesel combustion was found to be 7% from a previous study. The objective is to keep NO_x and soot emissions at reasonable levels such that a 3-way catalyst and DPF can be used in an aftertreatment combination to meet 2010 emissions regulation.

The effects of intake conditions and the use of group-hole injector nozzles (GHN) on fuel consumption of stoichiometric diesel operation were investigated. Throttled intake conditions exhibited about a 30% fuel penalty compared to the best fuel economy case of high boost/EGR intake conditions. The higher CO emissions of throttled intake cases lead to the poor fuel economy. In comparison with the use of single-hole nozzles (SHN) GHN lowered the fuel consumption by 3% under throttled intake conditions, whereas it exhibited no significant improvement in ISFC in the high boost/EGR cases. In addition, while investigating the best fuel economy location from SOI sweeps, a sweet spot [1] was observed in exhaust CO and O₂ measurements.

INTRODUCTION

This study addresses the question whether a three-way catalyst (TWC) can be used on a diesel engine for NO_x reduction. As well established in the literature a TWC operates most efficiently while an engine runs at stoichiometric conditions [2]. However, the combustion efficiency can decrease when a diesel engine runs under this condition.

Lee et al. [3] reported combustion efficiencies as high as 94% with stoichiometric diesel operation with 1.3 bar intake pressure and approximately 55% EGR for 18mg/cyc fueling rate, as shown in Figure 1.

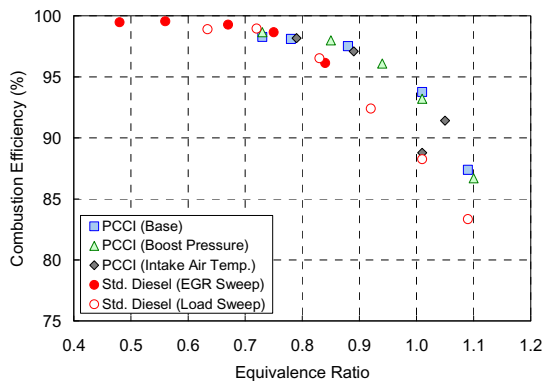


Figure 1 Combustion efficiencies with respect to equivalence ratio from Lee et al. [3]

Figure 2 shows that the carbon monoxide emission (CO) increases from 15 g/kWhr to as high as 110 g/kWhr, when the equivalence ratio is raised from 0.7 to 1.0 by increasing EGR with premixed charge compression ignition (PCCI) operation [4]. Lee et al. [4] estimated that a combined total of 6% of the fuel energy is stored in hydrocarbons (HC) and CO at stoichiometric conditions, as shown in Figure 3. Considering hydrogen production is approximately one third of CO emissions at stoichiometric operation [2], the total fuel energy stored in CO, HC, and H₂ in the exhaust can be estimated as 7%. Operating conditions of the four cases in the test are listed in Table 1.

Table 1 Operating conditions of four cases in Figure 2 and Figure 3 from Lee et al. [4]

	BASE	CASE_1	CASE_2	CASE_3
Engine Speed	2000 rpm			
Exhaust Pressure	Intake Pressure + 18 kPa			
Injection Pressure	150 MPa			
Intake Air Temp.	90 °C			40 °C
Intake	130 kPa	140 kPa	130 kPa	
SOI (° ATDC)	-35		-15	-35
EGR	55%			
Fuel Mass	8 ~ 19 mg/cycle			

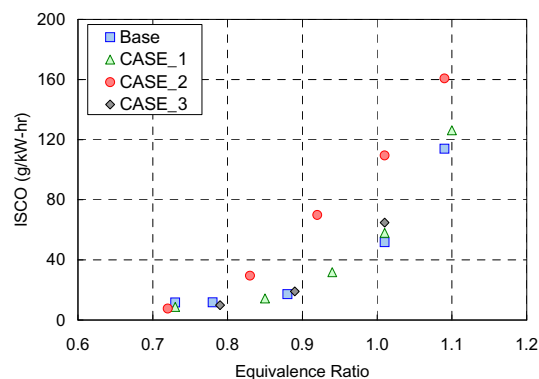


Figure 2 Carbon monoxide emission as a function of equivalence ratio from Lee et al. [4]. The operating conditions are given in Table 1.

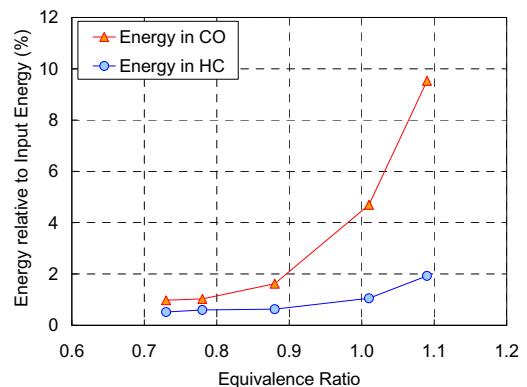


Figure 3 Energy exhausted through HC and CO emissions from Lee et al. [4] for 1.3 bar boost with 55% EGR for stoichiometric PCCI combustion (baseline case, Table 1)

Lee et al. [5] also reported that the spray included angle and injection timing affect the CO emissions substantially in the PCCI combustion regime. Among 6 different spray included angles a 130° included angle exhibited a relatively wide start-of-injection (SOI) range of low CO emissions in PCCI cases, as shown in Figure 4.

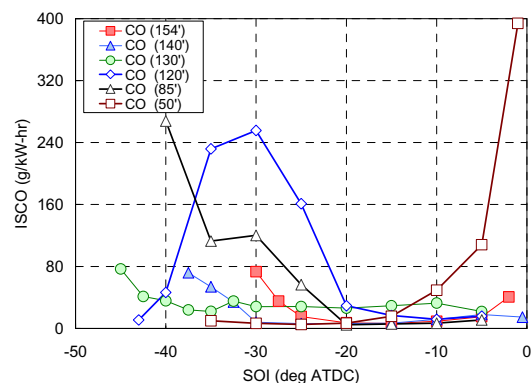


Figure 4 CO emissions with respect to spray included angles and injection timings from Lee et al. [5] for stoichiometric PCCI combustion (baseline case, Table 1)

Chase et al. [6] reported that the combustion efficiency decreased to 95% and fuel consumption increased by 15% under stoichiometric conditions with throttled intake and no EGR. Jacobs et al. [7] compared lean and rich premixed compression ignition (PCI) modes with a lean conventional combustion mode. Rich PCI was reached by reducing the air-fuel ratio (AF ratio), until NO_x and PM virtually disappeared, which happened to be near stoichiometric AF ratio. Jacobs et al. [7] concluded that the rich PCI mode eliminated 99% NO_x and 98% PM while BSFC was increased by 33% compared to the lean conventional mode. The operating conditions and results of their tests are listed in Table 2. These studies indicate that improved utilization of the fuel is needed under stoichiometric operating conditions, and therefore methods to improve mixing are of interest.

Table 2 Operating conditions and results from Jacobs et al. [7]

	Lean Conventional	Lean PCI	Rich PCI
Speed (RPM)	1500	1500	1500
BMEP (bar)	3.96	3.75	3.66
Rail Pressure (bar)	300	1000	1000
EGR Rate (%)	32	45	50
Inj Timing (°BTDC)	5	15	25
EI-NOx (g/kg-fuel)	4.28	0.31	0.05
EI-PM (g/kg-fuel)	0.34	0.07	0.01
Noise (dB)	88.2	88.7	88.8
BSFC (g/kW-hr)	235	246	307
CO (%)	--	--	5.56
AF Ratio	27.5	16.4	12.5

Group-hole nozzles (GHN) have the potential to enhance fuel and air mixing, and are explored in the present study for more efficient oxygen usage at stoichiometric conditions. A representative feature of GHN is that two or more smaller nozzle holes are closely positioned to replace one hole. GHN have been developed to allow operation with small nozzle orifices without sacrificing the spray spatial distribution [8]. A GHN can be compared with 2 different baseline nozzles to investigate its effects. One nozzle, namely the single-hole nozzle (SHN), has the conventional single hole whose diameter is such that the total cross sectional area is the same as that of each GHN. The other nozzle, the micro-hole nozzle (MHN), has a single hole whose diameter is the same as that of each GHN. SHN and MHN have half the total number of nozzle holes.

GHN influences the mixture preparation as follows. Firstly, the collision of the two identical jets in the overlap region is assumed to produce droplet agglomerates, which travel further due to their combined mass and momentum. This effect causes the spray tip to penetrate further and induces more ambient gas to be entrained. The total momentum of the fuel injected from a GHN must be the same as that of the SHN, assuming both GHN and SHN have the same fuel mass and velocity. Thus, in theory the GHN spray tip penetration would be as long as that of SHN. A second change is that the droplets outside of the jet overlap region have the same SMD as that of MHN. Thus, the amount of fuel vapor in the GHN spray at a certain time is expected to be higher than that of SHN, but less than that of MHN due to the collided droplets in the jet overlap region. The last change to be expected from GHN is in the spray spatial distribution. Two sprays are deflected from each other due to collision. Thus, the deflected spray from the bottom hole penetrates deeper towards the center of the piston bowl.

Gao et al. [8, 9] compared various types of GHN with MHN and SHN in spray visualization experiments. They found that the GHN had longer spray tip penetration than the MHN, but shorter than the SHN, and GHN had faster fuel vaporization than SHN. In addition to deeper spray penetration, they found that the “outward bulge” observed in the jet collision region in a bottom view of

the spray widens the spray further, which could give further advantage of better mixture. They concluded that the nozzle-hole diameters, distance, and angle between the two jets can affect the location and size of the jet overlap region. A bigger jet-overlap region can increase spray tip penetration effects, but slow down the fuel vaporization.

Dohle et al. [10] tested various group-hole nozzle configurations on a 400-cc single cylinder engine with a 1600-bar common rail system. For investigation of the effects of the angle between the two jets, they tested converging, parallel, and diverging jet nozzles. They found that converging and diverging jet nozzles exhibit the longest and shortest spray penetration, respectively. The converging jet nozzle also gave the highest power at 4000 rev/min, the highest NOx at all loads at 2000 rev/min, and the highest HC at low load at 1500 rev/min among the 3 nozzles.

Park et al. [11, 12] conducted simulations to investigate the effects of GHN with the KIVA code. The engine geometries and injection variables were based on the experimental facilities introduced in the present paper. They found that GHN advantages are more obvious under richer conditions with earlier start-of-injection (SOI) timing.

EXPERIMENTAL SETUP

Table 3 Engine specifications

Engine Type	DOHC DI Diesel
Bore × Stroke	82.0 × 90.4 mm
Compression Ratio	15.5
Displacement	477 cm ³
Piston Geometry	Open-Crater-bowl
Intake Ports	1 helical, 1 tangential
Swirl Ratio	2~5.5 (adjustable)
Bowl Diameter	49.4mm
IVO	344 ATDC
IVC	-132 ATDC
EVO	112 ATDC
EVC	388 BTDC

The experimental engine for this study is a single-cylinder direct-injection diesel engine based on a GM 4-cylinder DOHC 1.9-liter diesel engine. The engine has two intake valves with helical and tangential ports, and two exhaust valves. The swirl ratio is adjustable between 2.2 and 5.6 using two butterfly valves in the intake runner between the intake surge tank and the engine head. The engine head is mounted on an 82-mm bore cylinder block, which is installed on top of a Ricardo Hydra single cylinder engine block. The custom made 1.3-mm engine head gasket is a three-layer stainless steel gasket, and the outer layers are thin Viton-coated for good sealing. The piston has an open-crater type bowl, and the geometry was selected using a genetic algorithm (GA) CFD analysis for a 130° included angle injector nozzle. As shown in Figure 4, Lee et al. [5] reported that a 130° included angle nozzle provides optimal results for conventional diesel combustion. The

piston is shown in Figure 5 and the bowl volume is 24.7 cm³. The piston top surface is flat without any valve cut-outs. A detailed description of the bowl design and the GA optimization process can be found in [13].

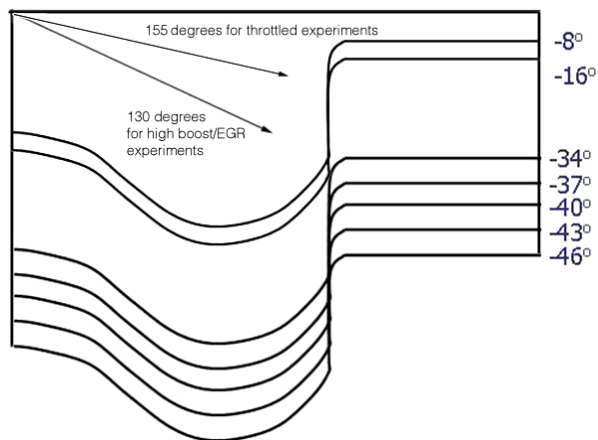


Figure 5 Vertical cross-sectional view of piston positions at SOI timings and spray targeting with 130° and 155° included angle nozzles

A schematic diagram of the engine layout is given in Figure 6. Special care was taken to utilize a 4-cylinder engine head on the single-cylinder engine block. Two factors considered in the cylinder selection were the

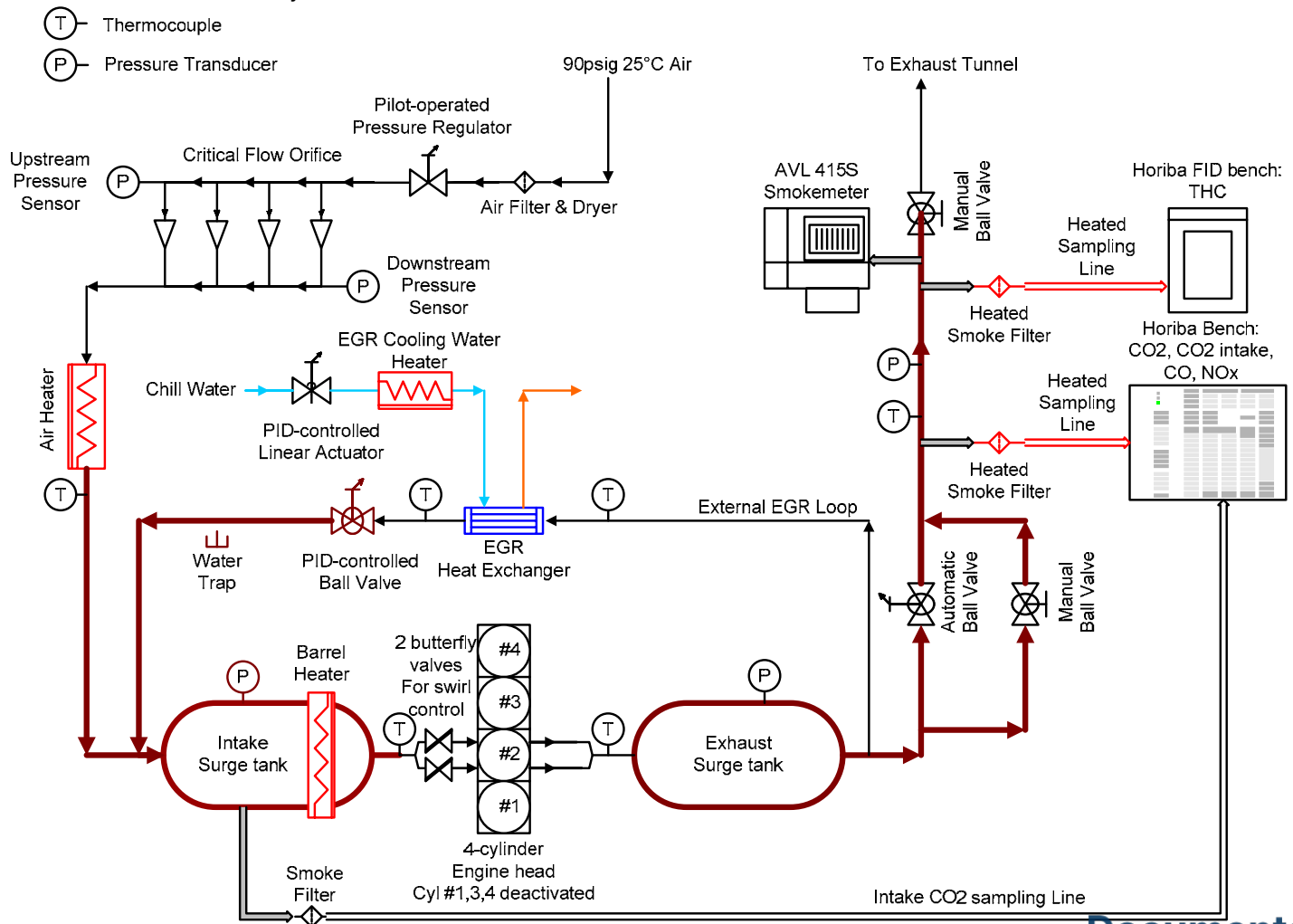


Figure 6 Schematic of the engine test facility for a single cylinder direct injection diesel

engine head balance and a minimized crankshaft extension for the crank pulley mounting. Cylinder number two was chosen because it offered a more balanced engine head layout than use of the first cylinder, although the second cylinder required an approximately 2 inch shaft extension. A regular steel 6-bolt tightened flange coupler was employed in order to connect the space between the crank pulley and crankshaft end, which was created by the first cylinder that protruded from the engine front.

Intake and exhaust surge tanks were implemented in order to minimize wave phenomena in the intake, exhaust, and external EGR lines. The surge tank volume was approximately 100 times the engine displacement volume. A series of Critical Flow Orifices (CFO) and electric heaters were employed to simulate the intake air supply in typical turbocharged vehicles. An electrically controlled pressure regulator upstream of the CFO provided compressed air at a pressure range between 0 and 6.2 bars to the CFO. The pressure regulator was operated by PID control based on intake pressure.

A 1.5-kW electric heater was located between the CFO and intake surge tank to provide heated fresh air. In addition, a barrel heater was strapped on the intake surge tank to compensate for heat losses through the

intake pipe between the electric heater and the surge tank. The two heaters and fiberglass/aluminum foil insulation on the entire intake system could heat the intake air up to 140°C within an hour while operating the engine at 2000 rev/min.

In order to control the exhaust pressure two ball valves, one electrically controlled, and the other manually operated, were located in parallel downstream of the exhaust surge tank. A sampling pressure control valve was installed downstream of the back pressure control valve and it facilitated exhaust gas sampling for the two Horiba emissions benches. Pressurizing the exhaust gas at the emission sampling points also helps prevent dilution by room air.

Typically, recirculated exhaust gas can contain substantial amounts of water and soot. A PID-controlled gate valve and a water heater were employed upstream of the EGR cooling water line to supply EGR cooling water of around 45°C at appropriate flow rates. The EGR system consists of a PID-controlled gate valve and water heater for EGR cooling water warm-up, an EGR heat exchanger for EGR cooling, and a PID-controlled ball valve for the EGR rate control.

The engine valve train cover accommodates a plunger-solenoid operated intake valve closing timing controller. This feature requires oil suction pumps to force engine oil to return from the engine to an oil reservoir. While one pump supplies oil to the engine head and crankcase for the camshaft bearing, crankshaft bearing, and piston

cooling oil jet, two separate pumps were used for oil return from the engine head and crankcase to an external reservoir. The oil and coolant were preheated to 80°C prior to engine start.

All temperatures and pressures, such as oil and coolant, intake and exhaust gas, whether monitored or PID controlled, were recorded in the DAQ computer. Exhaust temperature was measured at the exhaust port between the engine head and exhaust surge tank. A Horiba 20-series emission bench was used for CO₂, CO₂ intake, CO, and NO_x measurements. A Horiba 200-series bench with a FID analyzer was employed for C₃ based THC measurement. The bench maintains a sampling line and smoke filter heated to 188°C [14]. An AVL 415S smokemeter was used to measure soot.

A 3600-pulse per revolution encoder coupled with the crankshaft provided 0.1 crank angle (CA) resolution for CA-based data analysis, such as cylinder pressure and heat release analysis. A piezoelectric Kistler 6125B pressure transducer was installed in the glow plug hole in the engine head. The pressure trace was averaged over 300 cycles as recommended in [15]. A single-zone apparent heat release rate, which includes effects of wall heat transfer, was estimated based on the averaged pressure trace. The ratio of specific heats, γ , is a critical factor in the heat release analysis. In this study the temperature dependent γ equation of Gatowski et al. [16] was employed.

$$\gamma = 1.392 - 8.13 \times 10^{-5} \times T(K) \quad (1)$$

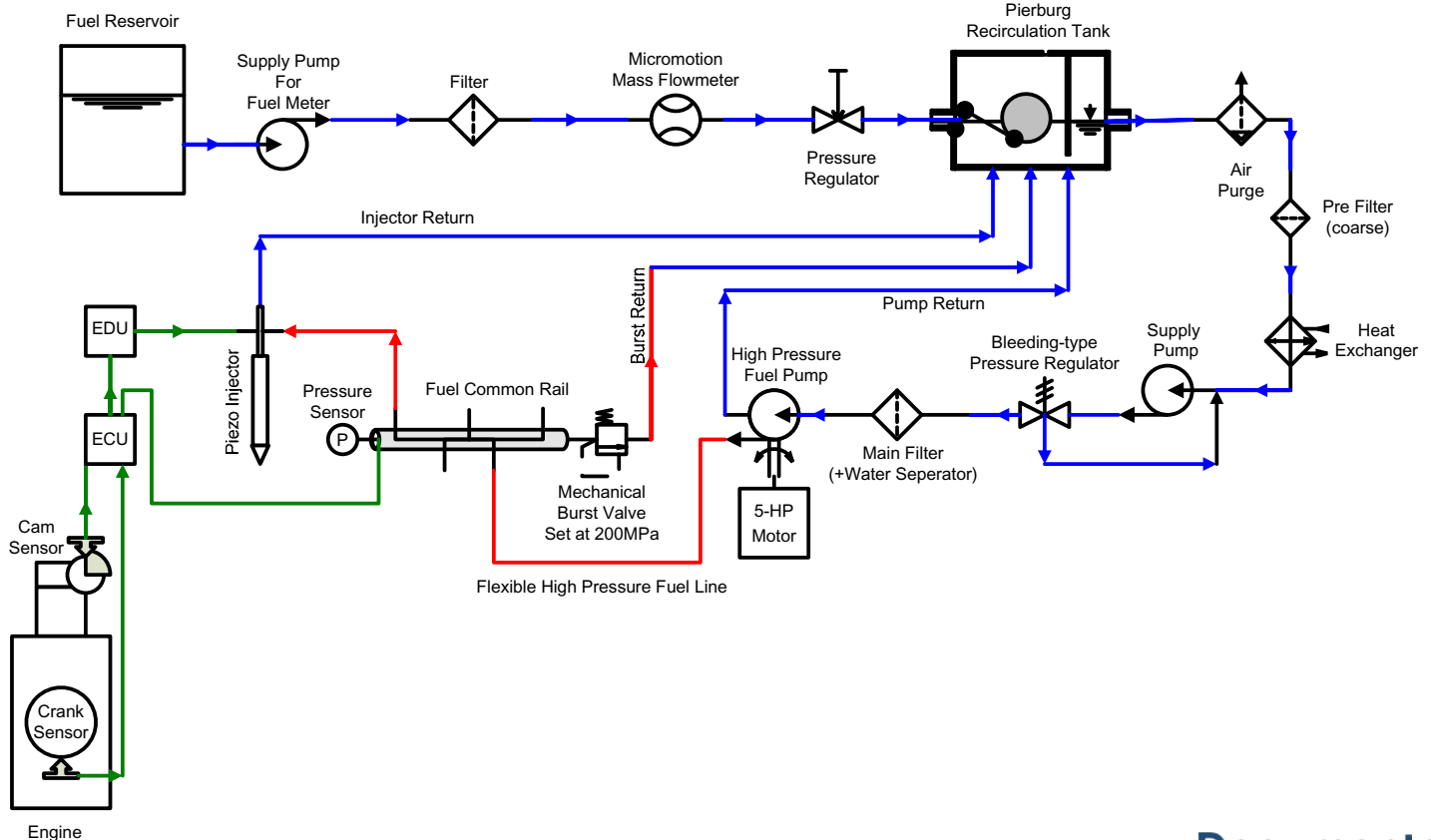


Figure 7 Schematic of injection and fuel flowrate measurement systems