

Multiple Injection Strategy in a Direct-Injection Natural Gas Engine with Entrained Diesel

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ABSTRACT

A new fuel injector prototype for heavy-duty engines has been developed to use direct-injection natural gas with small amounts of entrained diesel as an ignition promoter. This “co-injection” is quite different from other dual-fuel engine systems, where diesel and gas are introduced separately.

Reliable compression-ignition can be attained, but two injections per engine cycle are needed to minimize engine knock. In the present paper the interactions between diesel injection mass, combustion timing, engine load, and engine speed are investigated experimentally in a heavy-duty single-cylinder engine.

For the tests with this injector, ignition delay ranged from 1.2 – 4.0 ms (of which injector delay accounts for ~0.9 ms). Shorter ignition delays occurred at higher diesel injection masses and advanced combustion timing. At ignition delays shorter than 2.0 ms, knock intensity decreased with increasing ignition delay. Ignition delays longer than 2.0 ms indicated that the pilot gas injection did not ignite before large amounts of fuel were injected with the second injection; intense knock could occur in these cases.

At low load, carbon monoxide (CO) and total hydrocarbons (THC) emissions were anti-correlated with the diesel injection mass. Oxides of nitrogen (NOx) emissions depended strongly on timing, but not on diesel fuelling. At higher load, emissions of CO and THC were very low and showed no clear correlation with diesel fuelling. At advanced timing where the pilot gas injection was not observed to ignite (such as for the low-diesel runs) NOx emissions were higher.

INTRODUCTION

The use of natural gas vehicles (NGVs) around the world is growing rapidly. In 2001 there were about 1.7 million NGVs around the world. By 2007 the number had grown to about 7 million vehicles [1]. In heavy-duty applications, liquefied natural gas can provide useful vehicle range, but spark-ignited NG engines do not provide the power or efficiency of conventional diesel engines. Westport Innovations' High Pressure Direct Injection (HPDI) system can match the performance of conventional diesel engines and reduce emissions of NOx, particulate matter (PM) and greenhouse gases.

HPDI uses a dual-needle, dual-actuator fuel injector to inject both diesel and gas into the combustion chamber near the end of the compression stroke. Pilot diesel and natural gas are injected through separate sets of holes. With this system, small quantities of diesel auto-ignite while the main injection of natural gas is underway. The present generation of HPDI injectors have been developed over more than 10 years, and the emission and combustion characteristics of “conventional” HPDI are reasonably well understood [2-6].

The “co-injector” [7] has been developed as a simpler way of implementing the HPDI strategy, using only a single needle and a single actuator. It is hoped that the co-injector will result in a simplified injector with lower cost. The research prototype described in this paper was built by modifying an existing HPDI injector and therefore uses two needles. In co-injection, a small quantity of diesel fuel enters the injector above the needle seat and is carried into the combustion chamber when the gas injection starts. This gas-blast atomization results in very small diesel droplets, despite the fact that the fuel

supply pressure is only 200-260 bar (far lower than a modern common-rail diesel injection pressure), similar to conventional HPDI.

Gas blast atomization is common in other areas of engineering, and has some automotive antecedents. Air-assist atomization was used by Rudolf Diesel in the first compression ignition engines [8], and remained in use on larger engines into the 1940's [9]. The Orbital injector system uses a type of air-assist atomization for spark-ignited direct-injection gasoline fuelled engines [10]. In these systems, gas is used only to help disperse the liquid, not as the primary fuel.

In co-injection the primary fuel is gaseous, and the liquid is used only to tailor the ignition characteristics of the fuel mixture. Patents by Yang at Ford [11] and Denso [12] describe co-injectors, but provide no guidance regarding the correct diesel/gas ratios, the major characteristics of combustion, or the emissions that might result from co-injection. In fact, we are not aware of any work in the open literature on these key questions other than [13, 14].

It is important to note that HPDI and HPDI co-injection are different from conventional diesel/natural-gas engines where natural gas is fumigated into the intake charge air. For example, the ignition delay is larger in fumigated pilot-ignited engines [15, 16]. For HPDI, ignition delay is similar to pure diesel-fuelled engines [5, 6]. In HPDI, since both the natural gas and diesel are injected near top-dead center, knock is usually minimal and does not limit the compression ratio. In fumigated pilot-ignited natural-gas engines, knock depends strongly on fuel composition and compression ratio [16]. HPDI co-injection is expected to be similar to HPDI operation.

In this paper we experimentally look at the differences in combustion characteristics and gaseous emissions over a wide range of operating conditions for the co-injector. Tests were carried out in the Single Cylinder Research Engine (SCRE) facility at the University of British Columbia, Canada.

APPARATUS DESCRIPTION

CO-INJECTOR

The HPDI™ system consists of two concentric injectors with diesel introduced in the inner injection system and natural gas in the outer injection system, as shown in Figure 1. The injector geometry of the co-injector is based on the original HPDI injector geometry with the following modifications. The diesel needle was shortened and a "diesel plug" was inserted below it. Diesel is injected through 7 new holes into the gas plenum, which is above the gas-needle seat.

It is important to note that due to actuator and fuel supply constraints, the diesel must be injected into the plenum over a short duration (2-4 milliseconds), with a driving pressure ("diesel-gas bias" i.e. differential pressure of diesel over gas) of 0.7 to 2 MPa. Although this is far lower than conventional diesel injection pressures, it results in sufficient jet velocities to partially atomize the diesel and distribute it inside the injector. Therefore, the diesel is not injected *before* the gas, but rather *throughout* the pilot injection and during part of the second ("main") injection. This has important consequences for the combustion, as discussed later. A sleeve was installed on the outside of the gas needle in an attempt to reduce the distribution of diesel within the injector and increase the efficiency with which the gas flow would entrain it.

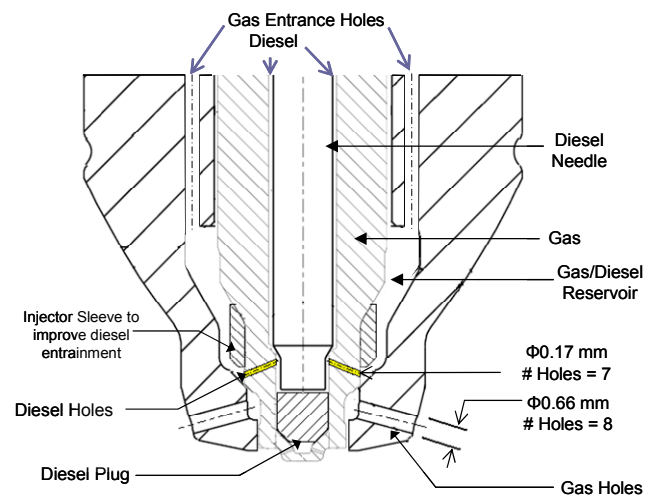


Figure 1: Schematic of HPDI nozzle modifications to make co-injector.

For normal operation, three injections are needed for this co-injector: the diesel pre-injection, the pilot injection (gas and diesel) and the main injection (mostly gas, Figure 2). During the pre-injection, the diesel needle lifts and diesel is injected into the common gas/diesel reservoir. For future co-injectors, the diesel pre-injection will be removed and diesel will continuously "leak" into the gas/diesel reservoir. During the pilot injection, the mixture of diesel and gas is injected into the combustion chamber. Based on preliminary spray visualization [17] and the low liquid/gas ratio of this 2-phase injection process, we believe that the diesel is very finely atomized. Finally, the gas needle lifts again and the main charge of natural gas is injected with a leftover quantity of diesel in the gas plenum. The main injection usually occurs between -10° to 3° after top dead centre (ATDC).

Figure 2: Schematic of commanded injections with injection parameters. RIT = Relative injection timing.

ENGINE

Tests were conducted in a single-cylinder research engine ("SCRE", Figure 3). The SCRE was derived from a 6-cylinder, 400 HP Cummins ISX engine modified to operate on a single cylinder (2.5 L displacement, 137 mm bore, 169 mm stroke, 261.5 mm connecting rod). The engine has a compression ratio of 16.7:1.

The SCRE is fully instrumented for emissions, cylinder pressures, and other operating conditions. The mass flow rate of the compressor-supplied combustion air is measured by an in-house designed and calibrated venturi and controlled through a pressure regulator. Engine speed was controlled by an eddy-current water-cooled dynamometer. At low loads a 30 kW electric drive motor assisted in overcoming the frictional losses of the non-firing cylinders.

FUEL SUPPLY AND MEASUREMENT

A screw-type compressor supplies high-pressure natural gas to the engine. Typical natural gas in Vancouver BC contains approximately 96% methane with 2% higher hydrocarbons and 2% $N_2 + CO_2$. Gas pressure is held at a pressure slightly below the diesel pressure using a dome-loaded regulator. A coriolis mass-flow meter is used on the gas supply. The diesel was road-grade low-sulphur (<500 ppm) which met Canadian General Standards Board (CAN/CGSB-3.520) specification. Diesel flow is supplied by a conventional Bosch diesel fuel pump; pressure is set with a back-pressure regulator. Most of the diesel flow is used for injector actuation and returned at low pressure to the small diesel supply tank. This tank sits on an electronic balance, from which the net mass of diesel injected into the engine can be inferred. This provides accurate measurements of the diesel mass injected in each cycle, but *cannot* be used to determine the distribution of diesel mass between two injections in the same cycle. Thus, for an experiment with a diesel pre-injection of 12 mg/inj, the actual mass of diesel in the pilot injection might be only 7 mg/inj. The distribution of diesel between the two pilot and main injections can be roughly inferred from examination of the integrated heat release (IHR) curves for single and double injection tests [13]; it changes with test conditions.

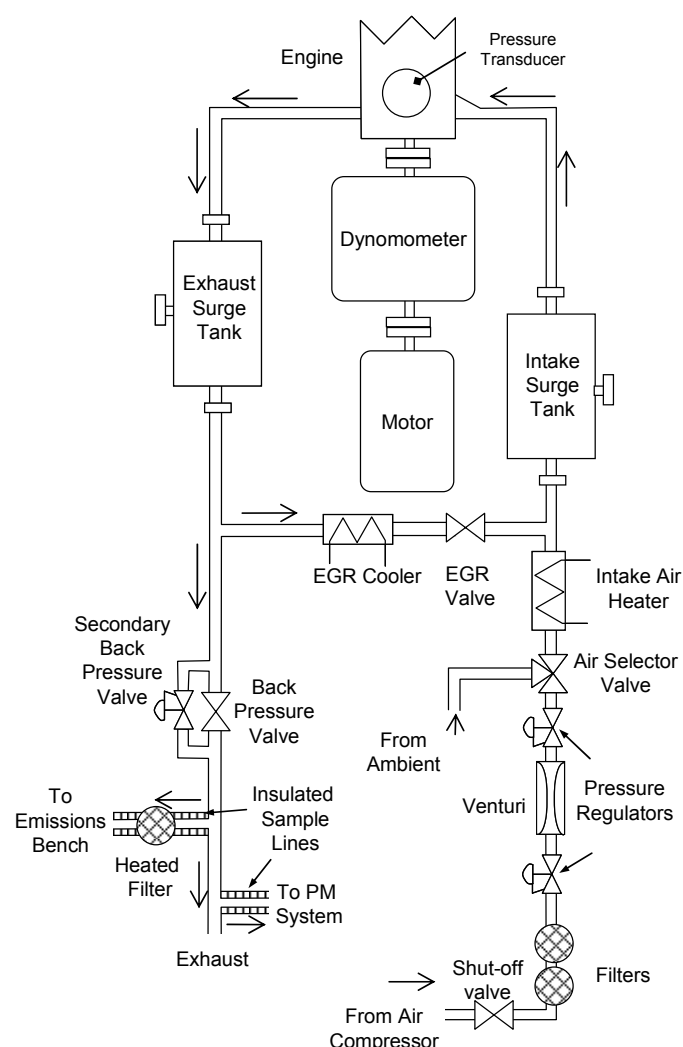


Figure 3: Schematic of test engine. EGR not used in the present study.

EXHAUST EMISSIONS MEASUREMENT

Exhaust emissions were determined with an AVL CEB-NA emissions bench. This is a newer system than used in [2]; comparison with the older system is described in [13]. Particulate matter was not measured in this test series. Earlier work with the co-injector showed that PM emissions were extremely low for non-EGR conditions; in future work with EGR, particles will be measured.

IN-CYLINDER PRESSURE MEASUREMENT

Cylinder pressure was measured with a flush-mounted water-cooled piezoelectric transducer [2]. The indicated pressure was recorded every $\frac{1}{2}$ degree crank angle for 45 cycles in order to calculate in real time metrics such as the heat release rate (HRR), the IHR, the gross indicated mean effective pressure (GIMEP), the ignition delay, and the knock intensity.

The HRR is computed from the cylinder pressure from the following relation:

$$\text{HRR} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta}$$

where θ is the crank angle and γ is the specific heat ratio (assumed constant). The IHR was computed by integrating the HRR for the entire combustion process and was used to adjust the combustion timing as desired. Similarly, the total fuelling rate was adjusted to produce the GIMEP required by the test matrix. GIMEP is the pressure averaged over the compression and power strokes (hence, neglecting pumping losses). It is a consistent indicator of engine torque, but power computed from GIMEP (used to normalize emissions and fuel consumption) is therefore higher than the engine brake horsepower. The UBC SCRE, with 5 non-firing cylinders, has much higher friction than typical engines of the same power, so it would be misleading to report brake-specific properties. The standard deviation of GIMEP, normalized by GIMEP, is the coefficient of variation (COV) used to characterize cyclic variability.

Ignition delay was calculated as the duration between the start of injection and the start of combustion. Consistent with Stone [18], the start of combustion was defined as the first non-zero heat release rate (HRR). Practically, this was determined by extrapolating from the crank angles with $\text{HRR}=30$ and $\text{HRR}=20$ $\text{kJ/m}^3/\text{deg}$ to find the crank angle at a HRR of 0 $\text{kJ/m}^3/\text{deg}$.

The total ignition delay consists of three parts: the injector mechanical delay, the physical delay, and the chemical delay. The injector delay is the interval between the injection command signal and the needle opening. The injector delay has been found to be between ~ 0.9 ms for the co-injector (compared to 0.7 ms for the HPDI reference injector), relatively independent of cylinder pressure [17]. In subsequent figures showing ignition delay, the injector delay has been removed to more accurately show the physical and chemical delay. The physical delay is the time it takes for the fuel to heat up, evaporate, and mix to an ignitable mixture, with chemical delay being the time it would take for an ignitable mixture to ignite. The chemical and physical delay overlap since chemical reactions can be taking place as the fuel is mixing.

KNOCK INTENSITY

We found operating conditions where significant amounts of high-frequency pressure fluctuations were observed on the indicated pressure trace (Figure 4).

The high-frequency pressure fluctuations (an indication of knock) are due to the gas in the cylinder resonating at the first transverse mode acoustical frequency [19]. This may be due to a significant amount of fuel burning during the rapid premixed burning phase of combustion [20].

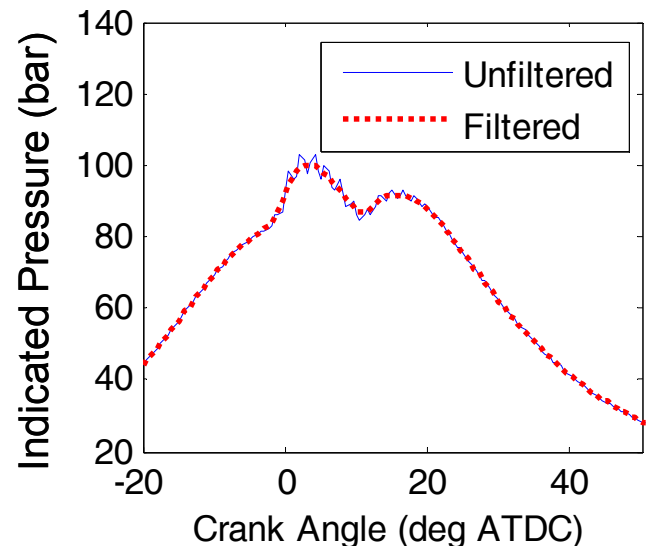


Figure 4: Filtered and original unfiltered cylinder pressure traces for a test exhibiting engine knock.

In order to quantify knock intensity, the first transverse mode acoustical frequency is calculated as

$$f_{1st\ mode} = \frac{ac}{\pi D},$$

where “a” is a geometric constant (taken as 1.84 for this case), “c” is the speed of sound of the gas, and “D” is the cylinder bore diameter [21]. For the knocking case of Figure 4, significant energy is visible in the range of the expected frequency, as shown when the indicated pressure curve is transformed into the frequency domain (Figure 5). Rather than using analog hardware filters directly on the pressure measurement, we have instead post-processed unfiltered measurements. The first mode vibrational frequency was calculated to be about 3.7 kHz.

The transformed pressures can be filtered around 3.7 kHz, transformed back to time domain, and subtracted from the original cylinder pressure. For each cycle, the maximum difference between filtered and unfiltered pressure is computed, and then these maxima are averaged over 45 cycles (Figure 6, where knock intensity for this case is approximately 4 bar) [19].