

Direct Injection of Natural Gas at up to 600 Bar in a Pilot-Ignited Heavy-Duty Engine

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ABSTRACT

Retaining the diesel combustion process but burning primarily natural gas offers diesel-like efficiencies from a natural-gas fuelled heavy-duty engine. This combustion event is limited by the injection pressure of the fuel, as this dictates the rate of mixing and hence of combustion. Typical late-cycle direct injection applications are limited to approximately 300 bar fuel pressure. The current work reports on tests for the first time at natural gas injection pressures up to 600 bar. The results show that significant efficiency and particulate matter reductions can be achieved at high loads, especially at higher speeds where the combustion is injection rate limited at conventional pressures. Increases in combustion noise and harshness are a drawback of higher pressures, but these can be mitigated by reducing the diameter of the nozzle gas holes to control the fuel injection rate. Higher pressures lead to faster combustion, which allows earlier combustion phasing without exceeding peak cylinder pressures, improving efficiency. Steady-state cycle-composite results show that efficiency benefits on the order of 3% and PM reductions of 40-60% can be achieved through increased gas rail pressures.

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INTRODUCTION

Engine efficiency is critical in maximizing the environmental and economic benefits of natural gas fuelling of heavy-duty vehicles. Retaining a non-premixed, diesel-like combustion process offers high fuel-conversion efficiency and high compression ratio while avoiding part-load throttling losses, resulting in efficiencies similar to those of equivalent diesel-fueled engines. One such technology, commercialized by Westport Innovations Inc., is high pressure direct injection (HPDI) of natural gas [1,2]. In this system, natural gas is injected at high pressure late in the compression stroke and burns in a predominantly non-premixed combustion event very similar to that in a diesel engine. One technique for improving diesel engine efficiencies is through the use of higher injection pressures. Some of these benefits have been attributed to improved atomization of the diesel fuel spray. As a result, similar benefits would not automatically be expected from a non-premixed natural gas jet. The work presented here aimed to evaluate for the first time the potential power, efficiency and emissions benefits of increasing the injection pressures in an HPDI engine from 300 bar up to 600 bar for heavy-duty on-road applications.

HIGH PRESSURE INJECTION

The advent and widespread adoption of common rail diesel injection systems over the past 15-20 years has led to a dramatic increase in injection pressures. In the mid 1990's, pressures in common rail

systems were on the order of 1400 bar [3,4]; production systems now offer up to 2700 bar [5], with research systems over 3000 bar. The key benefits of these high pressure systems include higher power, better efficiency, reduced smoke, and better NOx - PM tradeoffs [3,6,7]. They have also enabled more advanced injection strategies, including split injection and post injection [8], which are important for noise control and thermal management. Higher pressures have been shown to generate higher spray velocities and greater penetration [9], and narrower spray angles with better atomization [10]. In parallel to these developments in the diesel engine industry, Westport has pursued the development of common-rail direct-injection of natural gas technology for heavy-duty engines. Due to the added work requirements to compress a gaseous fuel, pressures have typically been limited to approximately 300 bar. To date, there has been no evidence of whether the benefits of higher injection pressures demonstrated by diesel engines, many of which are due to the spray formation for the liquid fuel, would be seen in a direct-injection of natural gas engine.

High Pressure Direct Injection (HPDI) of Natural Gas

HPDI is a pilot-ignited non-premixed natural gas combustion strategy. It is in service in commercial heavy-truck applications in North America and Australia (Figure 1). Results show that maximum torque and efficiency are very close to those of equivalent diesel engines, while up to 95% of the fuel used (on an energy basis) is

natural gas [1,11]. Tank-to-wheel GHG emissions as much as 20% lower than an equivalent diesel engine have been demonstrated, when the HPDI system is optimized for efficiency [2].



Figure 1. KW T408 SARE with a B-double trailer configuration powered by a 15 L HPDI engine in service in Australia

HPDI Combustion

As in a conventional diesel engine, in HPDI the inducted charge consists of fresh air externally mixed with some recirculated exhaust.

The charge is then compressed; just before the end of the compression stroke, a small amount of diesel is injected. This is followed by an injection of natural gas through separate nozzle holes. The diesel pilot auto-ignites, providing multiple ignition sources that ignite the natural gas jets. Assessment of key aspects of the HPDI combustion process has been reported previously (e.g. [12]).

In an HPDI engine, end-gas autoignition (knock) is not a concern as the fuel and air are not premixed; this allows the engine to retain a relatively high compression ratio. The global air-fuel ratio remains similar to that of an equivalent diesel engine, with similar sensitivity to EGR and combustion phasing. Load control is achieved by simply reducing the amount of natural gas injected; part-load throttling is not required. Emissions of unburned methane are controlled, even at low load, by adjusting the injection process to avoid over-mixing of the natural gas. Crevice and end-gas regions see virtually no fuel, improving fuel conversion efficiency and reducing tailpipe methane emissions.

HPDI Fuel System

Liquefied natural gas (LNG) offers benefits of increased volumetric and gravimetric energy density compared to other alternatives for heavy-duty natural gas engines [13]. For HPDI, this state has the added benefit of being much less compressible than gaseous natural gas, reducing the compression work needed to achieve the desired fuel rail pressure. A schematic of the key components for an LNG-based HPDI fuelling system is shown in Figure 2. In this system, cryogenic LNG is first compressed to the desired supply pressure (up to approximately 300 bar in a standard HPDI engine), using a pump immersed in the LNG tank. The natural gas is then passed through a heat exchanger to bring it up to engine temperatures.

The HPDI system also includes a common-rail diesel fuel system. This includes a high-pressure common-rail pump, providing diesel to the engine fuel system at a pressure slightly above that of the natural

gas. The diesel rail pressure is regulated using an inlet-metering valve (IMV) on the diesel pump. The controlled diesel rail pressure is also used to regulate the natural gas fuel rail pressure.

Both high pressure gas and diesel are supplied to the on-engine fuel system. The gas pressure is regulated relative to the diesel pressure using a dome-loaded self-venting regulator. HPDI injectors are twin-fuel concentric needle injectors, with separate needles for controlling diesel and natural gas injection through different sets of holes. A schematic of the HPDI injector nozzle is shown in Figure 3. The diesel needle is located inside the gas needle. The diesel holes are located in the tip of the gas needle, which protrudes below the bottom of the injector nozzle. Gas and diesel injection are independently controlled, with separate actuators to enable changes in relative timing and injection durations.

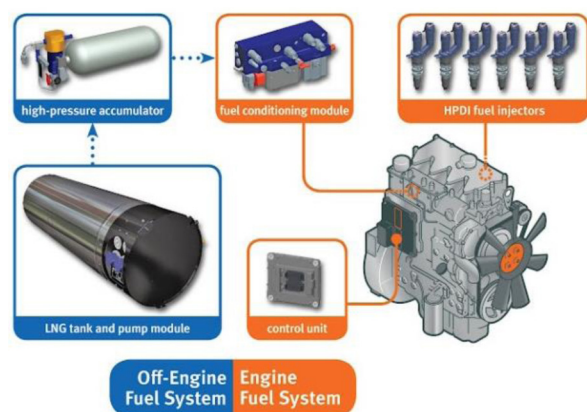


Figure 2. HPDI LNG fuel system

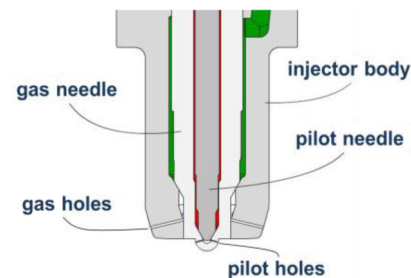


Figure 3. HPDI injector nozzle schematic

Increased Fuel System Pressures

As in common-rail diesel engines, increased diesel fuel pressure in an HPDI engine is achieved by requesting a higher pressure from the IMV on the pump. As the pressures used are low relative to modern diesel common-rail systems [4], the diesel fuel system is easily capable of meeting the demands for pressure increases up to 600 bar.

Unlike liquid diesel, natural gas is a compressible fluid; any increase in pressure results in a significant increase in density. As the pressures encountered in the fuel system of an HPDI engine (300 bar) are well above the critical pressure for methane (the principle constituent of natural gas), there is no clear distinction between liquid and vapour states; the natural gas remains compressible down to very low

temperatures. The density of natural gas (composition shown in [Appendix 1](#)), as a function of pressure is shown in [Figure 4](#) for several temperatures. This was calculated using Refprop v 9.1 [14].

RESEARCH METHODOLOGY

The evaluation of increased injection pressure for HPDI combustion reported here includes both computational and experimental assessment. The computational aspects include 1-D coupled hydraulic/electrical injector modelling to assess performance and to specify injector configuration. Combustion CFD using Westport's in-house combustion code, designed specifically to simulate HPDI combustion, was used to both guide the experimental program and to explain the results. The core of the work was experimental engine testing on a multi-cylinder research engine.

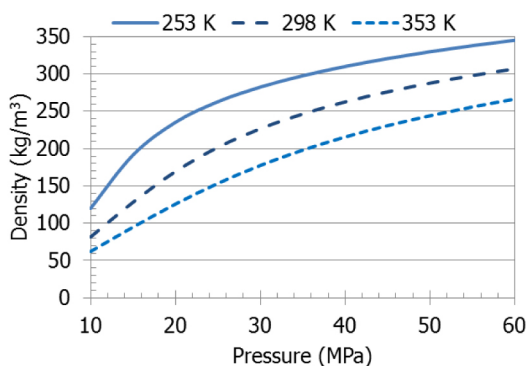


Figure 4. Natural gas density as a function of pressure, at selected temperatures

Injector 1-D Model

The effects of higher rail pressures on injector performance were evaluated using a one-dimensional model of the twin-fuel HPDI injector developed in LMS Imagine.lab Amesim version 11. Amesim provides a multi-physics environment that incorporates interactions between the mechanical and hydraulic elements of the injector. The model is capable of simulating the response of both diesel and gas fuel streams and the actuation of both diesel and gas needles. Standard Amesim components are used in most cases, but a customized element was developed to simulate squeeze damping of the injector plungers. This model used the Reynold's equation to adjust the force transfer between the plunger and the needle due to the presence of a liquid film between the two elements.

The model is used to evaluate the sensitivities of injector needle lift and flow characteristics for both diesel and gas with respect to different input parameters. The model was tuned and validated with test data collected on in-house custom test rigs under controlled pressures and temperatures. The model included the following assumptions and limitations:

1. methane was used in place of CNG;
2. match fits (where high pressure gas and diesel sections are separated by closely machined components) were assumed to be leak-tight;
3. the combustion chamber pressure was assumed to be constant;
4. the electromagnetic actuation was not modeled in detail -

look-up tables based on test rig data were used to provide the relationship of force versus displacement and current;

5. pressure pulsations in both the gas and diesel rails were ignored; and
6. the dynamic jet forces associated with the flow of fluid through the valve were assumed to be negligible.

Combustion 3-D CFD Model

A three-dimensional CFD model was used to simulate in-cylinder flows and was coupled with a chemical kinetic solver to evaluate combustion progression. The main objective of the modeling was to evaluate the local equivalence ratio and the local temperature during the injection and combustion event, and to use that information to interpret the experimental engine results. The CFD program was based on OpenFoam, with turbulent flow structures modelled using large eddy simulation (LES). The chemical mechanisms were implemented through a trajectory-generated low-dimensional manifold (TGLDM) method [15] using detailed chemical kinetic mechanisms for natural gas and heptane (a surrogate for diesel fuel). A conceptual schematic of the chemical closure methodology is shown in [Figure 5](#). Natural gas was simulated with a modified GRI mechanism (71 species, 379 reactions) [16,17], while heptane used a Lawrence Livermore National Lab (LLNL) mechanism (170 species and 1500 reactions). The interactions between turbulence and chemical reaction were addressed using the conditional source term estimation (CSE) method [18]. The CFD model has been extensively validated against in-cylinder pressure, heat-release, and NOx emissions data from single- and multi-cylinder HPDI natural gas engines; more detail on the model is reported elsewhere [11,19]. Particulate matter formation is modelled using the Hiroyasu approach [20], with parameters modified to better represent the natural gas combustion in an HPDI engine.

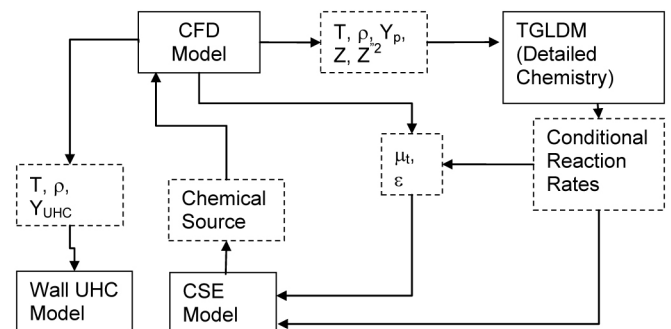


Figure 5. Structure of turbulent combustion closure model for chemical source term

For the work reported here, simulation was of a sectional model covering a 40° wedge of the combustion chamber, representing one pair of diesel and gas nozzle holes. Boundary conditions at the section sides were cyclic, i.e. the combustion event was assumed to be the same for all 9 holes. As a result, variations in flow or combustion chamber geometry, due for example to valve cut-outs, are not represented in the model. This also neglects hole-to-hole variation in diesel-gas alignment, due to the non-equal number of gas and diesel holes (see [Table 1](#)) in the actual engine. However, previous work has shown that the simplification assumption of consistent

gas-diesel alignment is representative of actual engine conditions [11,19]. The model is run over the closed compression and power strokes, from 90° crank angle (CA) before top-dead-centre (TDC) to 125° CA after TDC.

Research Engine

The work reported here was conducted on an instrumented research version of the Westport HD 15L HPDI engine. The base engine configuration and calibration was identical to the CARB/EPA 2010 certified engine [21]. The engine was installed on an engine test bed equipped with an eddy-current dynamometer. The specifics of the research engine are given in Table 1. The base engine hardware (including pistons and valve timing), air handling system, and exhaust aftertreatment are unmodified from the 2007 ISX15 engine that the HD 15L is based upon. More details on the research engine are presented elsewhere (see, for e.g. [19,22]).

Table 1. Research engine specifications

Base Engine	Cummins ISX15
Bore / Stroke / Connecting Rod Length	137 / 169 / 261 mm
Swept Vol. / Cyl.	2.49 L
Number of cylinders	6
Base CR	17:1
Air handling	VGT, cooled high pressure EGR
Aftertreatment	DOC, DPF, SCR
Fuel System	
Injector	Westport J36 HPDI
Hole number / angle	Diesel: 7 / 18°; Gas 9 / 18°
Nozzle flow (static)	Standard flow: 61 SLPM Low flow: 42 SLPM
Fuel rail pressures	Gas: up to 600 bar Diesel: Gas + 30 bar

The research engine was fully instrumented, including six flush-mounted piezoelectric pressure transducers; two (cylinders 1 and 6) were water cooled and the other four were uncooled. Only the water-cooled transducers were used for all further analysis. This data was collected at a 0.5°CA resolution over at least 80 consecutive cycles. The in-cylinder pressure from the water-cooled transducers was used to determine the apparent heat release rate (HRR) and to define the combustion phasing. This is defined as the timing at which half of the apparent heat release has occurred, as calculated from the cumulative heat release. Engine brake torque and speed were determined from sensors on the dynamometer. Intake and exhaust stream pressures and temperatures were recorded at multiple points in the system. Charge air flow was measured using a Meriam laminar flow element and the diesel flow was measured using a gravimetric fuel balance. Emissions of CO, CO₂, NO_x, THC and CH₄ were measured using a Horiba MEXA3100DEGR emissions bench; this was also used to measure intake CO₂, to quantify the EGR rate. Engine-out smoke emissions were measured using an AVL 415 smoke meter. These are reported on the basis of filter smoke number (FSN), and only represent black-carbon emissions. Previous work [11] has shown that percentage reductions in FSN agree well with the reductions determined from gravimetric filter-based measurements.

Repeatability and reproducibility was assessed by evaluating the long-term variations in measured parameters at a fixed operating condition. For this work, the variation in key parameters was evaluated for a reference condition of 2400 N.m and 1490 RPM. The results are shown in Table 2, in the form of coefficient of variation (COV), defined as the standard deviation divided by the mean. The results shown are for variability over a period of approximately three months during the testing program reported here. This value incorporates measurement errors, random variations in engine condition set-point, and long-term variations in engine performance.

Table 2. Coefficient of variation of key parameters at 1490 RPM, 2400 N.m, and base gas rail pressure

Parameter	Long-term COV
BSFC	±1.2%
FSN	±10%
CH ₄	±11%
CO	±8%
NO _x	±5%
Peak HRR	±5%
dP/dCA _{max}	±6%
Ringing intensity	±9%

Both diesel and natural gas fuels were supplied to the test cell from standard facility supplies. Ultra-low sulphur road-grade diesel that met North American fuel specifications was used as the pilot fuel. Natural gas was drawn from the distribution network in Vancouver, Canada and then compressed on-site. Over the duration of the tests, CH₄ concentration remained in the range of 95% ±2% by mol. A representative fuel composition is provided in Appendix 1.

High Pressure Fuel Supply System

A new natural gas supply system capable of supplying pressures at up to 690 bar was developed for this project. This was composed of a Hydropac industrial gas compressor to boost the pressure from the existing 320 bar test cell gas supply up to 690 bar. A GE-Rheonik Coriolis flow meter was used to measure high pressure gas flow.

The on-engine fuel system also modified for the higher pressures. The diesel supply system, a Bosch CP-3 common rail pump with IMV, was retained but plumbing was replaced with thicker wall tube. The natural gas pressure regulation, controlled relative to the diesel pressure, was provided by a custom dome-loaded self-venting regulator rated to 690 bar. This system ensured that the differential between the diesel and natural gas rail pressures remained constant throughout the tests, independent of rail pressure.

The high pressure diesel and gas were supplied to the injectors through the ISX engine's internal fuel rails drilled through the cylinder head. Hydrostatic testing verified that these rails would withstand the desired pressure with a sufficient margin for pulsating, gaseous flows. The bodies and nozzles of the HPDI injectors were not modified for high pressure operation. FEA was conducted on the injector to ensure structural integrity under both static and operating conditions. All evaluations demonstrated sufficient factors-of-safety for test cell research purposes.

Injectors

The two-fuel concentric needle injectors used in the work were based on the production version used in the HD 15L HPDI product. The modifications for high pressure testing were to the actuator settings and the gas nozzle holes.

In the HPDI injectors, diesel acts as the hydraulic fluid that actuates both the gas and diesel needles. The pressures in the control chambers above the needles are controlled by separate solenoid actuators. To account for the higher pressure of diesel supplied to these actuators, the actuator ball and seat areas, and the spring preload and stroke, were adjusted based on input from the Amesim model. To achieve progressively higher injection pressures, the preload was increased; this led to an increase in the minimum diesel pressure required to actuate the injectors. This minimum pressure could not exceed approximately 200 bar, as this was the maximum pressure that could be achieved during engine start-up. This then directly limited the maximum achievable gas rail pressure (GRP) to 600 bar.

As natural gas is compressible (see Figure 4), higher supply pressures lead to higher densities. For flow through an orifice (choked or unchoked), this leads to higher mass flow rates. Reducing the injection flow area by reducing gas hole diameter offers the potential to retain the kinetic energy and velocity benefits of higher rail pressures while avoiding excessive fuelling rates. For this work, two injector nozzle configurations were tested, one with standard flow area and a second with reduced flow, as shown in Table 1. The lower flow was achieved by reducing the diameters of the gas holes by 20%. The diesel hole size was unchanged between the two injector sets tested.

Results

The research program included 1-D injector and 3-D combustion CFD modelling and experimental engine testing. Injector performance was predicted using the 1-D model to help in defining the injector configuration and to understand injector performance.

Injector Performance Modelling

The 1-D coupled hydraulic-mechanical model was used to evaluate actuator performance, needle lift and fuel mass flow rate as a function of rail pressure for various injector actuator configurations. A final configuration was defined based on these results, providing a trade-off between minimum injector opening pressure, maximum achievable pressure, and good injector control. The results from this model are shown in Figure 6 for the standard-flow injector nozzles. The model assumes a constant downstream chamber pressure; in this case, a value of 170 bar was used to represent the average chamber pressure during the injection event. As the flow was choked (pressure ratio > 1.8) for all the conditions shown in the figure except for the baseline pressure, the downstream pressure has no significant effect on the mass flow rate. Even at maximum cylinder pressure (180 bar) the pressure ratio across the injector remained above 1.8 for gas rail pressures above 325 bar.

As Figure 6 shows, the opening and closing phasing of the actuator are not significantly influenced by pressure, but the rate of opening of the actuator is increased. This leads to earlier start of actual injection,

as shown in the mass flow rate plot. For most of the rail pressures evaluated, the initial gas injection rate (slope of the mass flow rate curve) was essentially constant. However, the higher density of the gas increases the maximum mass flow rate. The gas closing ramp / time was not affected by rail pressure.

The effects of rail pressure on the modelled performance of the diesel pilot injection are also shown in Figure 6. The higher diesel rail pressure led to a higher mass flow rate due to a larger pressure differential across the nozzle. This increase was approximately 75% between the lowest (280 bar) and highest (600 bar) rail pressures simulated. The injector also opened more rapidly with higher rail pressures. For the injector closing, the timing of the actuator was fixed, but the modelling results suggest that, above a certain pressure, the needle itself was slower to close after the commanded end of injection. This led to higher mass flow rates and shorter separation until the gas start-of-injection. This was observed for cases where the system pressure was sufficiently high that the force exerted by the spring to close the actuator needle was only slightly above the force of the high pressure diesel acting to keep the actuator open. Increasing the actuator spring preload, to increase the closing force, reduced this impact, and brought the actual diesel needle closing time back to the level seen at lower pressures. However, this was at the cost of increasing the minimum pressure required to open the actuator and hence to initiate the injection event.

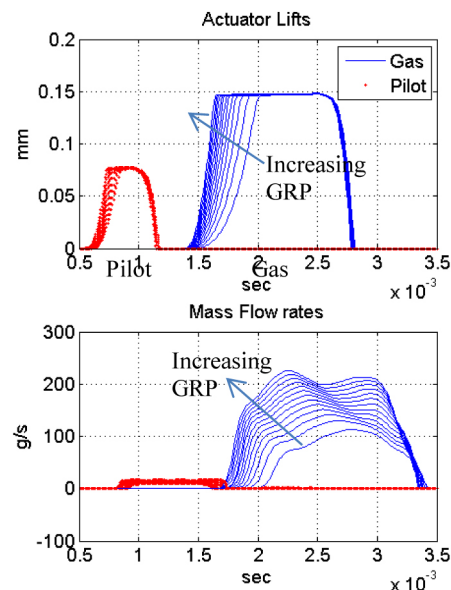


Figure 6. Amesim 1-D model results of injector performance at gas rail pressures between 280 and 600 bar. Blue curves are gas injection, red are diesel injection. Commanded actuator open times: pilot 0.5-0.9 ms, gas 1.3-2.5 ms.

The key findings from the simulation of injector performance are given below.

1. Injected gas mass increases with higher rail pressure for a given gas pulse width. This is a result of both earlier and faster needle lift, leading to a longer injection duration, and to higher density of the natural gas.