

Performance, Efficiency and Emissions Assessment of Natural Gas Direct Injection compared to Gasoline and Natural Gas Port-Fuel Injection in an Automotive Engine

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ABSTRACT

Interest in natural gas as a fuel for light-duty transportation has increased due to its domestic availability and lower cost relative to gasoline. Natural gas, comprised mainly of methane, has a higher knock resistance than gasoline making it advantageous for high load operation. However, the lower flame speeds of natural gas can cause ignitability issues at part-load operation leading to an increase in the initial flame development process. While port-fuel injection of natural gas can lead to a loss in power density due to the displacement of intake air, injecting natural gas directly into the cylinder can reduce such losses.

A study was designed and performed to evaluate the potential of natural gas for use as a light-duty fuel. Steady-state baseline tests were performed on a single-cylinder research engine equipped for port-fuel injection of gasoline and natural gas, as well as centrally mounted direct injection of natural gas.

Experimental results suggest that similar efficiencies can be achieved in part-load operation for both gasoline and natural gas. While the effects of injection timing are generally minimal for port-fuel injection, varying the injection timing for direct injection, especially after intake valve closure, can speed up the early flame development process by nearly 18°C. Results at full-load suggest that operation with natural gas regardless of fuel system allows for an efficiency increase. While port-fuel injection of natural gas leads to a power density loss, direct injection of natural gas allows for up to a 10% improvement in full-load power density over liquid and gaseous port-fuel injection for a naturally aspirated engine. In addition to increasing full-load efficiencies, natural gas operation allows for up to a 30% reduction in engine out carbon dioxide emissions at full-load.

CITATION: Sevik, J., Pamminger, M., Wallner, T., Scarcelli, R. et al., "Performance, Efficiency and Emissions Assessment of Natural Gas Direct Injection compared to Gasoline and Natural Gas Port-Fuel Injection in an Automotive Engine," *SAE Int. J. Engines* 9(2):2016, doi:10.4271/2016-01-0806.

INTRODUCTION

A renewed interest in natural gas has been sparked, due to advances in hydraulic fracturing techniques and low fuel costs compared to petroleum. Natural gas, primarily composed of methane, can be extracted from shale deposits in the ground or made from renewable sources such as biomass. Figure 1 shows the historic development of natural gas production in the U.S. since 2007. Total U.S. natural gas gross production reached 85 billion cubic feet per day in 2014, while shale gas wells became the largest source of total natural gas production in 2013. Whether the natural gas is derived from conventional wells, shale deposits or biomass, it can be sourced domestically, allowing for a decreased dependence on foreign oil.

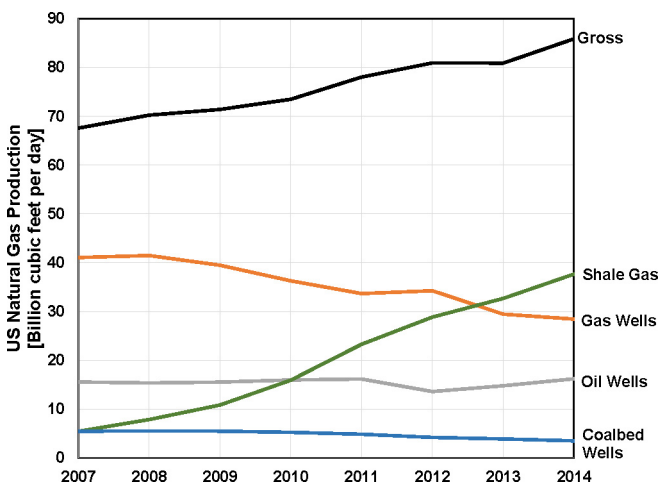


Figure 1. US Natural Gas Production [1]

In addition to the lower fuel cost, natural gas can reduce overall CO₂ emissions. Natural gas has the potential to produce 0.35kg less CO₂ per kg of fuel burned with respect to gasoline. Natural gas also has a higher energy content than gasoline. Combining the difference in hydrogen-to-carbon ratio and higher energy content can result in a 24% benefit for CO₂ reduction [2]. However, natural gas has a lower volumetric energy density, which can cause storage issues for vehicle level applications.

Recent studies have found that natural gas fueled vehicles can reduce the overall carbon footprint of the vehicle considerably. Researchers tested nine gasoline, eight natural gas, and two hybrid passenger cars on a chassis dynamometer, with 25% urban, 50% rural and 25% highway driving and found up to a 36% reduction in CO₂ tailpipe emissions for the natural gas vehicles, when compared to their gasoline counterparts. The CO₂ reduction of the natural gas vehicles can be further increased by taking into account the 23% reduction in well-to-tank CO₂ emissions. [3]

Natural gas can be particularly beneficial at full-load conditions in an automotive application, due to its higher resistance to engine knock. This would negate the need for fuel enrichment required at full-load with gasoline, as well as the ability to maintain optimal combustion phasing. Current vehicle retrofit applications still use port-fuel injection, which can lead to a power density loss at full-load due to natural gas displacing intake air [4]. For instance, the bi-fuel Chevrolet 6.0l V8 L96 engine is rated for 360HP at 5400rpm. With

port-fuel injected natural gas operation, peak power is limited to 301HP at 5000rpm [5]. One way to improve the full-load performance is through direct injection of natural gas.

Recent research has been performed on a downsized boosted gasoline direct injection (GDI) engine with port-fuel and direct injection (DI) natural gas. The most noticeable losses and improvement with natural gas were found at the low load, depending on the injection system. With port-fuel injected natural gas, there was a 30% loss in low end torque when compared to GDI operation at the same engine speed [6]. This study showed that with natural gas DI, two-thirds of the torque lost due to port-fuel injection can be made up for with combustion phasing set for peak efficiency. Retarding the combustion phasing to increase exhaust enthalpy into the turbo results in increased boost pressure, bringing the low end torque back to the same levels as GDI. Under low speed high loads, GDI operation is knock limited resulting in the 50% mass fraction burned (MFB) location needing to be delayed until 20-30°CA ATDC. With port-fuel and natural gas DI, the 50%MFB location was able to be held in the optimal range of 8-10°CA ATDC. However, some spark retard was required due to peak cylinder pressure limitations [6].

The purpose of this study is to investigate the performance, efficiency and emissions differences comparing gasoline and natural gas port-fuel injection, as well as natural gas DI. Steady-state tests were conducted at part-load as well as full-load conditions. Part-load will highlight the benefits of the tested configurations, as this is a region where the potential for efficiency improvements are greatest. Full-load tests will help to outline the improvements due to the different tested configurations, as well as different fuels.

EXPERIMENTAL MATRIX

Experimental Setup

Steady-state tests were conducted on a single-cylinder research engine, equipped for both port-fuel, and centrally mounted DI of the fuel. This engine configuration is representative of modern GDI engines, with specifications shown in Table 1.

Table 1. Engine geometry

Displaced volume	0.6264 L
Stroke	100.6 mm
Bore	89.04 mm
Compression ratio	10.5:1
Number of Valves	4
Spark Plug	NGK, 0.7mm gap
Tumble ratio	0.6

High speed crank angle resolved cylinder pressure data were recorded using an AVL GU21C transducer and AVL 365X crank angle encoder with a 0.5°CA physical resolution. Combustion air was supplied to the engine from an Atlas Copco air compressor; throttled conditions were achieved using a Parker pilot operated regulator in the intake

stream. Because combustion air was supplied from the external compressor, tests at wide open throttle (WOT) conditions were not affected by variations in ambient pressure.

Table 2 shows information for each injection system. For the steady-state tests performed, each injection system was tested independently. Unique to this study is usage of a fourth generation natural gas DI injector, supplied by Delphi [6]. This injector features an outward-opening valve, with maximum allowable injection pressure of up to 16 bar absolute. This injector allows for injection events to occur after intake valve closes, which has shown to improve low speed high load performance over gaseous port-fuel injection strategies [6].

Table 2. Fuel Injector Specifications

Type	Manufacturer	Injection Pressure	Type
PFI (liquid)	Ford	4.1 barg	Multi hole
PFI (gaseous)	Bosch	7 barg	Multi hole
DI (gaseous)	Delphi	15 barg	Outward opening

Figure 2 shows the valve lift profiles as a function of crank angle, with markers for exhaust valve closure (EVC), intake valve closure (IVC), exhaust valve open (EVO), and intake valve open (IVO). As seen, any injection timing after -140°CA is a closed valve event for DI. At the same time, a PFI injection event that ends before 340°CA is also closed valve, injecting on the backside of the intake valve before it opens. This is beneficial for E10 PFI, because early injection events allow for the charge cooling effect of the fuel to be utilized. Injecting while the intake valves are open can assist in reducing pumping losses, especially at part-load conditions.

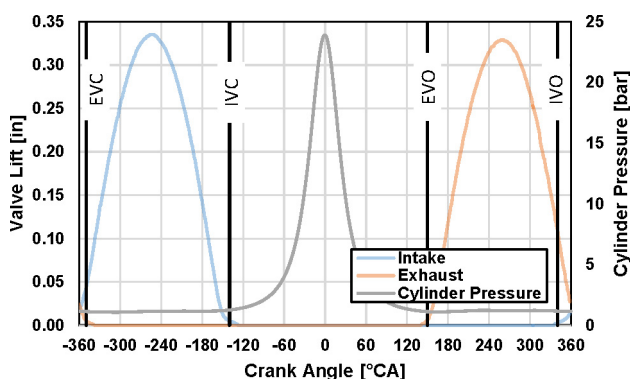


Figure 2. Valve lift profiles as a function of crank angle

Certification E10 was used for the liquid PFI tests, with specification listed in Table 3. This blend of E10 was chosen because it is representative of typical market fuels in the United States.

Table 3. E10 fuel specifications

$\text{C}_7\text{H}_{16}\text{O}$ [wt%]	82.08, 14.13, 3.79
AKI (R+M)/2 [-]	87.2
LHV [MJ/kg]	42.017
AFRSTOICH [-]	14.1

Rather than perform gaseous tests with pipeline natural gas, custom blends of compressed natural gas (CNG) were ordered from a regional distributor in order to keep composition of the gas constant. Table 4 shows the speciated composition of the CNG, as well as other relevant parameters. Methane number (MN) was chosen to evaluate the knock resistance of CNG, as opposed to using the motor octane number (MON). Test bounds setup to determine MON are designed for fuels with a maximum $\text{MON} \leq 120$ and generally are not suited for fuels with very high knock resistance [7,8]. The methane number is a function of the hydrogen-to-carbon ratio, with pure methane having a reference number of 100 [7,9]. Heavier hydrocarbons such as ethane and propane will lower the value of the methane number, effectively reducing the knock resistance. The composition of natural gas will vary based off of the regional location as well as time of the year [10].

Table 4. Gaseous Fuel Specifications

$\text{CH}_4, \text{C}_2\text{H}_6, \text{CO}_2, \text{N}_2$ [mol%]	94, 3, 2, 1
MN [-]	90.7
LHV [MJ/kg]	46.93
AFRSTOICH [-]	16.2

Experimental Approach

Steady-state engine testing was performed across a range of typical speed and load conditions summarized in Table 5. Every data point plotted in subsequent figures represent three averaged data points collected over 30 seconds at 5 Hz. Stoichiometric operation was maintained for all operating points unless otherwise noted. WOT conditions were measured holding the intake pressure constant, allowing the engine load to float. Start of injection for PFI operation was varied across a wide range spanning both open valve and closed valve timings. SOI for DI operation was limited due to increasing cylinder pressure during the early compression stroke and to maintain sufficient time between the end of injection and spark timing. SOI timing with DI during the gas exchange phase was limited to 360°CA BTDC to avoid excessive short circuiting natural gas through the open exhaust valves.

Table 5. Experimental test conditions

Engine Speed [rpm]	1500, 2000, 2500
IMEP [bar]	3.2, 5.6, 8.0, WOT
SOI DI [$^{\circ}\text{CA}$ BTDC fired]	Sweep, 120-360
SOI PFI (part-load/WOT) [$^{\circ}\text{CA}$ BTDC]	Sweep, 150-630

Combustion Metrics

Several combustion metrics are used to analyze the data. The flame development angle, shown in Equation 1, is defined as the crank angle duration from the time of the spark event, until a fraction of the fuel energy has been released; MFB10% has been chosen for this study [11].

$$\text{Flame Development Angle} = \text{Spark} - \text{MFB10\%}$$

The combustion duration is used to describe the overall burning process, defined in Equation 2 as the crank angle duration from MFB10% to MFB90% [11].

$$\text{Combustion Duration} = \text{MFB90\%} - \text{MFB10\%} \quad (2)$$

Cyclic variability was also of interest, measured using the coefficient of variance of the indicated net mean effective pressure (COV_{IMEP}), defined in Equation 3 as the standard deviation of IMEP divided by the mean IMEP. A 3% limit was imposed for stable engine operation and all high speed data were analyzed over 375 cycles.

$$\text{COV}_{\text{IMEP}} = \frac{\sigma_{\text{IMEP}}}{\mu_{\text{IMEP}}} \quad (3)$$

For all part-load tests, maximum brake torque (MBT) timing conditions were attained, allowing for optimal combustion phasing. Attempts were made to maintain MBT conditions under full-load tests, however, combustion phasing had to be delayed for E10 PFI in order to limit engine knock. Combustion phasing was held at 8°CA ATDC for CNG PFI and CNG DI.

For full-load tests, the volumetric efficiency was analyzed. The volumetric efficiency relation is shown in Equation 4 and 5 for PFI and DI, respectively. m_a and m_f are the mass of air and fuel on a cycle basis and V_d is the displacement. For mixture aspirating engines (PFI), the density of the fuel air mixture is used, denoted as $\rho_{m,i}$ in Equation 4. The density of air is used for DI, as the fuel injection process does not affect air in the intake, denoted as $\rho_{a,i}$ in Equation 5.

$$\eta_{V_{\text{PFI}}} = \frac{m_a + m_f}{\rho_{m,i} V_d} \quad (4)$$

$$\eta_{V_{\text{DI}}} = \frac{m_a}{\rho_{a,i} V_d} \quad (5)$$

Measurement Uncertainty

With any experimental measurement, there is some known error associated with the measurement devices. These uncertainties will propagate themselves through any calculation, introducing some deterministic variability in the data. The expected uncertainty of a measurement can be shown in Equation 6, where ω_R is the uncertainty in the result and ω_N is the uncertainties in each variable [12,13].

$$\omega_R = \left[\left(\frac{\partial R}{\partial x_1} \omega_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} \omega_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial x_N} \omega_N \right)^2 \right]^{\frac{1}{2}} \quad (6)$$

For the experiments performed, gaseous and liquid fuel flows are measured using a Micro Motion CMF010M fuel flow meter. The error in the fuel flow measurement is dependent on the flow rate, with lower flow rates will have a higher uncertainty while higher flow rates will have lower uncertainty [14]. There is a different level of uncertainty associated with measuring gaseous and liquid fuels. There

is also a level of uncertainty due to the cylinder pressure transducer and top dead center alignment. However, these were assumed to be constant for this scenario.

A hypothetical situation was designed in order to quantify the uncertainty in the measurement for the fuel flow meter, cylinder pressure transducer and top dead center alignment, which are used to calculate indicated efficiency. Equation 6 was used to develop a relation for the uncertainty of calculating the indicated thermal efficiency. A constant speed and efficiency was chosen for multiple load points; from there, a fuel flow rate could be calculated. Nominal heating values were used for each test fuel, shown in Table 3 and Table 4.

Figure 3 shows the propagation of uncertainty for the determination of indicated efficiency, as a function of engine load. As seen, the highest uncertainty occurs at lower loads due to lower fuel flow rates. The uncertainty of the gaseous flow remains constant for this scenario, while the liquid uncertainty continues to decrease. This becomes important when trying to compare two sets of data for the same calculated variable. For the instance of the 10 bar IMEP case, the uncertainty due to the measurement alone for E10 and CNG would be $\pm 0.36\%$ and $\pm 0.76\%$, respectively.

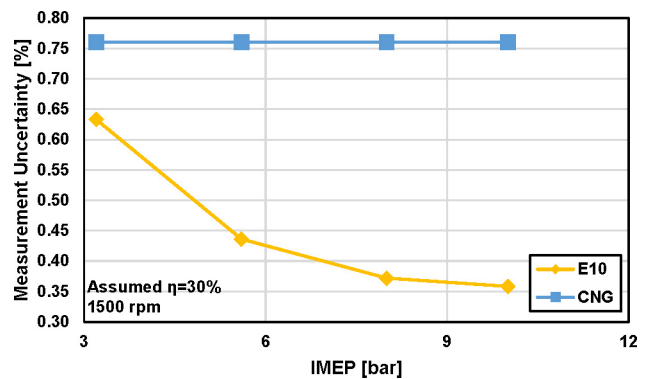


Figure 3. Propagation of uncertainty in the measurement for assumed conditions

EXPERIMENTAL RESULTS

Part-Load Performance

Figure 4 shows the net indicated thermal efficiency (ITE) and COV_{IMEP} as a function of SOI for the three tested configurations. Varying the SOI for E10 PFI has minimal impact on ITE, with a maximum of 32.5% at SOI=330°CA BTDC. There is some effect of changing the SOI for CNG PFI, resulting in efficiency changes of less than 0.5% absolute. ITE reaches a maximum of 32.5% at SOI=300°CA BTDC for CNG PFI, corresponding to a complete open valve injection event; the CNG PFI injection duration at this condition is 80°CA. CNG DI shows a greater sensitivity to SOI changes with a minimum of 30.3% at a SOI=120 and 300°CA BTDC, and a maximum of 31.4% at SOI=240 and 360°CA BTDC. Also, Figure 4 shows that stable operation with a COV_{IMEP} below 3% was maintained with E10 PFI and CNG PFI regardless of SOI. With the exception of SOI=270 and 300°CA BTDC, CNG DI had similar combustion stability levels relative to E10 and CNG PFI.