

Influence of Injector Location on Part-Load Performance Characteristics of Natural Gas Direct-Injection in a Spark Ignition Engine

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ABSTRACT

Interest in natural gas as an alternative fuel source to petroleum fuels for light-duty vehicle applications has increased due to its domestic availability and stable price compared to gasoline. With its higher hydrogen-to-carbon ratio, natural gas has the potential to reduce engine out carbon dioxide emissions, which has shown to be a strong greenhouse gas contributor. For part-load conditions, the lower flame speeds of natural gas can lead to an increased duration in the inflammation process with traditional port-injection. Direct-injection of natural gas can increase in-cylinder turbulence and has the potential to reduce problems typically associated with port-injection of natural gas, such as lower flame speeds and poor dilution tolerance.

A study was designed and executed to investigate the effects of direct-injection of natural gas at part-load conditions. Steady-state tests were performed on a single-cylinder research engine representative of current gasoline direct-injection engines. Tests were performed with direct-injection in the central and side location. The start of injection was varied under stoichiometric conditions in order to study the effects on the mixture formation process. In addition, exhaust gas recirculation was introduced at select conditions in order to investigate the dilution tolerance. Relevant combustion metrics were then analyzed for each scenario.

Experimental results suggest that regardless of the injector location, varying the start of injection has a strong impact on the mixture formation process. Delaying the start of injection from 300 to 120°CA BTDC can reduce the early flame development process by nearly 15°CA. While injecting into the cylinder after the intake valves have closed has shown to produce the fastest combustion process, this does not necessarily lead to the highest efficiency, due to increases in pumping and wall heat losses. When comparing the two injection configurations, the side location shows the best performance in terms of combustion metrics and efficiencies. For both systems, part-load dilution tolerance is affected by the injection timing, due to the induced turbulence from the gaseous injection event. CFD simulation results have shown that there is a fundamental difference in how the two injection locations affect the mixture formation process. Delayed injection timing increases the turbulence level in the cylinder at the time of the spark, but reduces the available time for proper mixing. Side injection delivers a gaseous jet that interacts more effectively with the intake induced flow field, and this improves the engine performance in terms of efficiency.

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INTRODUCTION

The use of natural gas (NG) as a transportation fuel can be traced back to World War I and World War II when it was used out of necessity due to gasoline shortages [1]. Due to increasing availability and production of NG in the U.S., interest in NG has once again increased. Future production forecasts show NG production to continue to increase [2]. In addition, the cost of NG on a gasoline gallon equivalent (GGE) basis has remained stable over the last decade, as shown in Figure 1, compared to the much more volatile gasoline, diesel, and E85.

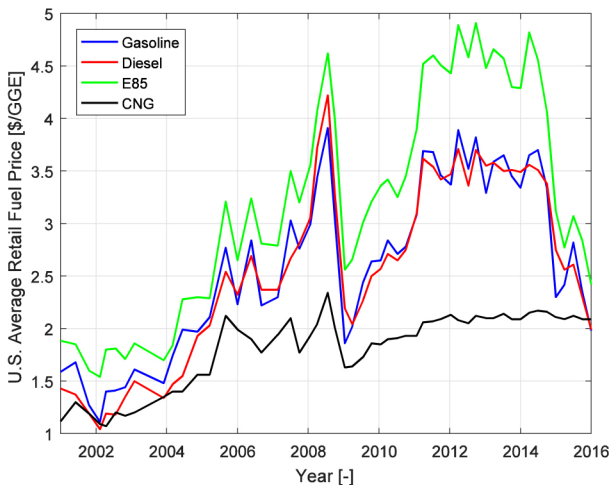


Figure 1. Marketed U.S. Natural Gas Production [3]

NG retrofit kits have gained popularity due to the domestic availability of gas, as well as the ability for NG consumption to reduce CO₂ emissions [4,5,6]. A common method is to introduce NG into the engine by replacing or adding a secondary port-fuel NG injector [7,8]. However, due to the lower flame speeds of natural gas, the length of the combustion process can increase significantly and at the same time decrease dilution tolerance when using port-injection of NG [6].

The availability of production level NG vehicles has also increased in recent years. Anderson et al. performed vehicle level tests on a chassis dynamometer over several drive cycles with a production NG vehicle and its gasoline counterpart [9]. Despite having a higher compression ratio, the NG vehicle yielded 3-9% lower fuel economy than the gasoline comparator. In addition to a 21% loss in peak power, the external-EGR loop was removed from the factory due to the lower dilution tolerance of NG. It was concluded that manufacturing an engine specifically for natural gas operation, with features such as NG direct-injection (DI) and increased charge motion, could meet or exceed efficiencies of current state-of-the-art gasoline engines.

While downsized boosted gasoline direct-injection (GDI) engines have become increasingly popular, there is also increased research involving NG DI. Husted et al has shown that under low engine speeds, NG port-fuel injection can lead to a loss in low end torque, up to 30% relative to GDI operation [10]. With NG DI, up to two-thirds of the lost engine torque can be made up for by maintaining combustion phasing for peak efficiency. In addition, where GDI

operation may become knock limited, NG DI allows for optimal combustion phasing while minimizing the chance of engine knock due to the higher knock resistance of NG.

Previous works of the authors have investigated the effects of gasoline and NG port-fuel injection and central NG DI at a part-load and full load condition [6]. Regardless of the injection system, similar part-load efficiencies were realized for gasoline and NG. Yet, while the effects of injection timing were minimal for the port-fuel injection systems, varying the injection timing for NG DI played a crucial role in influencing the early flame development process, as well as impacting the efficiency. In addition, NG DI allowed for a 10% load improvement at full load conditions relative to gasoline and NG port-fuel injection strategies. NG also allowed for higher efficiencies at full load relative to gasoline due to its higher knock resistance.

The purpose of this study is to investigate the performance, efficiency, and emissions characteristics of NG DI. In addition, the interactions of two different injector locations and their influence on the combustion process will be analyzed. Steady-state tests were conducted at part-load to study the effects of a gaseous injection event on the combustion process, as well as exhaust gas recirculation (EGR) dilution tolerance. Computational fluid dynamics (CFD) results will also be analyzed to further help explain the experimental results.

EXPERIMENTAL SETUP

Steady-state tests were conducted on a single-cylinder research engine, equipped for both centrally and side mounted NG DI. This engine configuration is representative of modern GDI engines, with specifications shown in Table 1. The cylinder head featured a 40° pent roof combustion chamber with a centrally mounted spark plug.

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.

A cooled high pressure EGR loop was installed in the test cell, using an automotive style EGR valve to control external EGR flow and temperature. Engine coolant was circulated through the EGR cooler, held at 85°C. The cooled EGR was introduced before the intake surge tank to ensure homogenous distribution in the intake charge. CO concentrations in ambient air were sampled twice daily to ensure representative EGR rates. EGR rates were calculated using Equating

1, sampling CO₂ emissions from the intake and exhaust stream. Raw emissions were sampled using a Pierburg AMA2000 emissions bench.

$$EGR = \frac{CO_{2intake} - CO_{2ambient}}{CO_{2exhaust} - CO_{2ambient}} \quad (1)$$

A conventional transistorized coil ignition (TCI) system was used for all tests performed. This coil is representative of coils currently implemented on GDI engines in the market, with a nominal energy level of 75 mJ.

Unique to this study is usage of a fourth generation natural gas DI injector, supplied by Delphi [11]. This injector features an outward-opening valve, with maximum allowable injection pressure of up to 16 bar absolute. In order to maintain critical flow, the minimum injection pressure is 8 bar absolute. This injector allows for injection events to occur after intake valves close, which has shown to improve low speed high load performance over gaseous port-fuel injection strategies [4]. Figure 2 shows a schematic of the combustion chamber with central and side mounted DI injectors. As can be seen, the side injector injects along the direction of the intake charge motion, while the central injector injects directly towards the piston. Literature has shown that it is possible for injection timings between 320-260°CA BTDC can lead to back-flow of NG into the intake manifold [12].

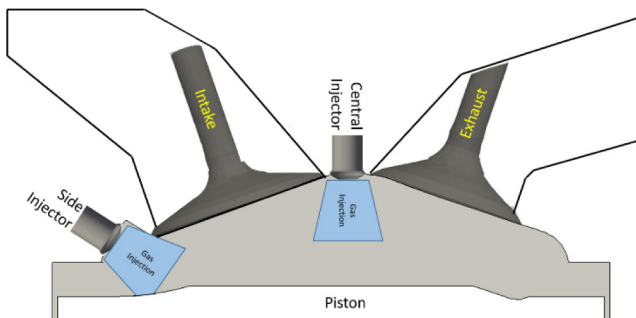


Figure 2. Combustion Chamber Schematic featuring Central and Side NG DI Injectors

Figure 3 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 ATDC is a closed valve event for DI. Injecting while the intake valves are open can assist in reducing pumping losses, especially at part-load conditions. As previously mentioned, some injection timings may lead to back-flow of natural gas into the intake manifold. As shown in Figure 3, for this given test setup, an injection timing occurring between 360 to 140°CA BTDC could lead to back-flow. If this occurs, the natural gas in the intake manifold would be reintroduced during the next combustion event.

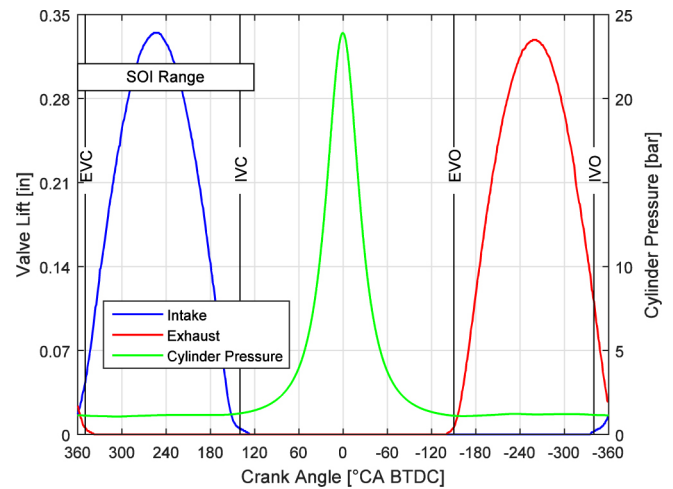


Figure 3. Valve lift profiles as a function of crank angle

The composition of natural gas will vary based off of the regional location as well as time of the year [13]. It was therefore decided to perform tests with custom blends of compressed natural gas (CNG) from a regional distributor rather than pipeline natural gas, in order to keep composition of the gas constant. Table 2 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 MON~120 and generally are not suited for fuels with very high knock resistance [14,15]. The methane number is a function of the hydrogen-to-carbon ratio, with pure methane having a reference number of 100 [16]. Heavier hydrocarbons such as ethane and propane will lower the value of the methane number.

Table 2. Gaseous Fuel Specifications

CH ₄ , C ₂ H ₆ , CO ₂ , N ₂ [mol%]	94, 3, 2, 1
MN [-]	90.7
LHV [MJ/kg]	46.93
AFR _{STOICH} [-]	16.2

EXPERIMENTAL APPROACH

Steady-state engine testing was performed at a single speed and load condition, summarized in Table 3. 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. Closed-loop control was used through the use of a Motec M800 engine control unit (ECU). Start of injection (SOI) for DI operation was limited due to increasing cylinder pressure during the early compression stroke and a need to maintain sufficient time between the end of injection (EOI) and spark timing. SOI timing with DI during the gas exchange phase was limited to 360°CA BTDC to avoid excessive short circuiting of natural gas through the open exhaust valves. In addition, injection pressure was held constant at 16 bar absolute for all conditions. Maximum brake torque (MBT) timing was maintained for all test conditions; spark timing was adjusted for each condition to target a mass fraction burned (MFB_{50%}) location of 8°CA ATDC.