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Development of a Natural Gas Spark Ignited Direct Injection Combustion System

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Abstract

Natural gas is increasingly being utilized for vehicle applications both to reduce vehicle emissions and as an alternate energy source to gasoline and diesel fuels. The composition of natural gas can be used to reduce carbon dioxide emissions while the global distribution of natural gas allows energy independence for regions with gas rather than oil reserves. Utilisation of natural gas in spark ignited automotive applications typically results in torque and power losses of approximately fifteen percent to twenty percent. This reduced engine performance, along with the increased vehicle weight for fuel storage, are obstacles to the wider adoption of natural gas as an automotive fuel. Approximately nine percent of the performance loss is attributable to the displacement of air by the gaseous fuel with the balance due to the combustion characteristics of natural gas or a non-optimum combustion system. By adopting a direct injection system, it is possible to inject the gas into the combustion chamber predominantly after inlet valve closing in order to limit the losses due to the displacement of air.

The development of a direct injection combustion system has been undertaken to improve the engine performance of a spark ignited CNG fuelled engine. This included development of an injection system and subsequent application to a single cylinder test engine. This paper outlines the challenges faced to meet the conflicting demands of a high flow rate injection system required for high power demands and the low flow metering accuracy for idle and low load operation. A natural gas direct injector is required to deliver high flow rates to effect direct injection at higher engine speeds with reduced injection durations to avoid displacement of in-cylinder air. In conjunction with this high flow rate the injector must provide controlled fuel metering for idle and low load operation. These factors create a challenging specification for injector turn down ratio.

For spark ignited applications, direct injection of natural gas offers performance benefits which reduce the torque and power loss typically associated with natural gas fuelling. To demonstrate the potential of such an application a prototype direct injector has been developed and tested on a 450 cc single cylinder research engine. Fuel injector bench test results along with fuel system modelling show the capability to fuel typical spark ignited automotive engines from idle to full load for both naturally aspirated and boosted applications. Notably improved injector durability is demonstrated for operation with dry gas through bench durability testing. Results from initial engine testing show significant engine performance improvement due to late cycle direct injection of natural gas when compared to manifold natural gas injection. Further performance improvement is anticipated through optimisation of both the engine and the injector to enhance mixing and combustion of natural gas. A direct injection CNG combustion system has been demonstrated on a naturally aspirated engine which may also be applied to a boosted engine, and which may bring particular benefits in low end torque and transient performance when implemented on a turbo-charged engine.

Introduction

The technology available for natural gas injection systems has predominantly been suited to aftermarket conversion from gasoline to dual fuel CNG/gasoline with fumigation or electronic manifold injection of natural gas. These systems have allowed the utilisation of natural gas as a transport fuel to spread despite the limited availability of factory fitted natural gas systems, but have enjoyed limited opportunity to optimise the performance realised with natural gas. The growing market for natural gas vehicles creates the potential for original vehicles developed for optimised performance with natural gas. This paper provides an insight into the benefits available from a direct injected natural gas combustion system and reports the bench test assessment of the fuel injection system and the dynamometer testing of the combustion system on a single cylinder research engine.

Catania, Misul, Spessa and Vassallo (1) identify the main disadvantages of CNG engines as decreased engine power output, potential increased NOx emissions and the weight and volume of the vehicle fuel storage. Decreased engine power output is due to reduced volumetric efficiency as a result of the gaseous state of the fuel in the engine inlet. Increased NOx may result from the requirement for advanced ignition due to the lower heat release rate. In this paper engine performance is measured for manifold injection of gasoline and natural gas. This provides a baseline performance to measure the benefits of direct injection against.

Kim (2) notes the benefits of direct injection of CNG for both full load and lean operation. Late cycle direct injection of CNG increased the engine IMEP by up to 15% while reducing ppm NOx although increasing ppm HC emissions. The testing was, however, limited to low speed operation. This paper includes results from a single cylinder research engine tested to 5000 rpm to assess performance improvements at higher speeds from the direct injected natural gas system. The results of the direct injected natural gas combustion system are compared with manifold injected natural gas operation and with manifold injected gasoline operation allowing analysis of the effects of both the fuel characteristics and the effect of improved charge trapping.

Kekedjian and Krepec (3) note a number of the issues associated with developing a DI gaseous injector. These included control of nozzle leakage, nozzle valve seat wear, injector needle bouncing, solenoid temperature and solenoid optimisation. The Orbital prototype DI CNG injector has been developed to achieve durability suitable for combustion development through some modification to the injector and through modifications to the injector driver waveform.

Pischinger, Umierski and Hüchtebrock (4) suggest that a variable compression engine will be beneficial for exploiting the greater knock tolerance of natural gas in a bi-fuel vehicle taking advantage of the increased octane number for CNG. They further suggest that a turbocharged lean burn concept is the most promising way to exploit all the characteristics of CNG. The testing in this paper was limited to a lower compression and natural aspiration, but the results do suggest that direct injection could be usefully combined with turbo charging.

In this paper test data is presented for a fuel system developed to meet the requirements for both naturally aspirated and boosted engines with fuel metering capability from 4 mg to 80 mg per cycle. This provided a prototype system with performance and durability suitable for on-engine combustion testing. The capability of the fuel system was demonstrated through on-engine testing. The performance of a 450 cc single cylinder research engine was improved by up to 10% through the direct injection of natural gas without other modification of the engine to take advantage of the combustion characteristics of natural gas. The greatest increase was achieved at low speeds with partial benefit retained at 5000 rpm. Further development is anticipated through optimisation of the injector nozzle, the engine compression ratio and in-cylinder motion.

DI CNG Injection system Development

Two key areas were addressed to develop a prototype DI CNG injection system. The fuel metering capability to cover the fuelling requirements from low to high load, and the injector durability when operated on dry gas and so without the lubrication provided by gasoline.

Fuel Metering

The time available for injection of gaseous fuels is significantly shorter than that available for injection of liquid fuels if the displacement of air is to be avoided. Figure 1 shows the available injection window for direct injected natural gas compared to that for direct injected gasoline and for inlet port fuel injection. Conventional inlet port injection of gasoline may occur at any time during the engine cycle. This provides the maximum possible time for fuel metering at high engine load. Typically the fuel may be injected onto the back of the inlet valves to promote evaporation prior to induction into the engine. Direct injection of gasoline has a reduced window to avoid injection during the exhaust stroke when unburnt fuel may short circuit straight to the exhaust as well as requiring injection to be complete before ignition. Direct injection of gaseous fuels has a similar constraint but may be further limited to injection during the compression stroke. Constraining the injection of gaseous fuel to the compression stroke avoids displacement of air from the engine cylinder but significantly reduces the time available for injection, increasing the flow rate required to meter the fuel required for full load operation. For boosted applications of direct injected natural gas, the injection duration may approach that of direct injected gasoline since the cylinder charging can be maintained by increased intake pressure.

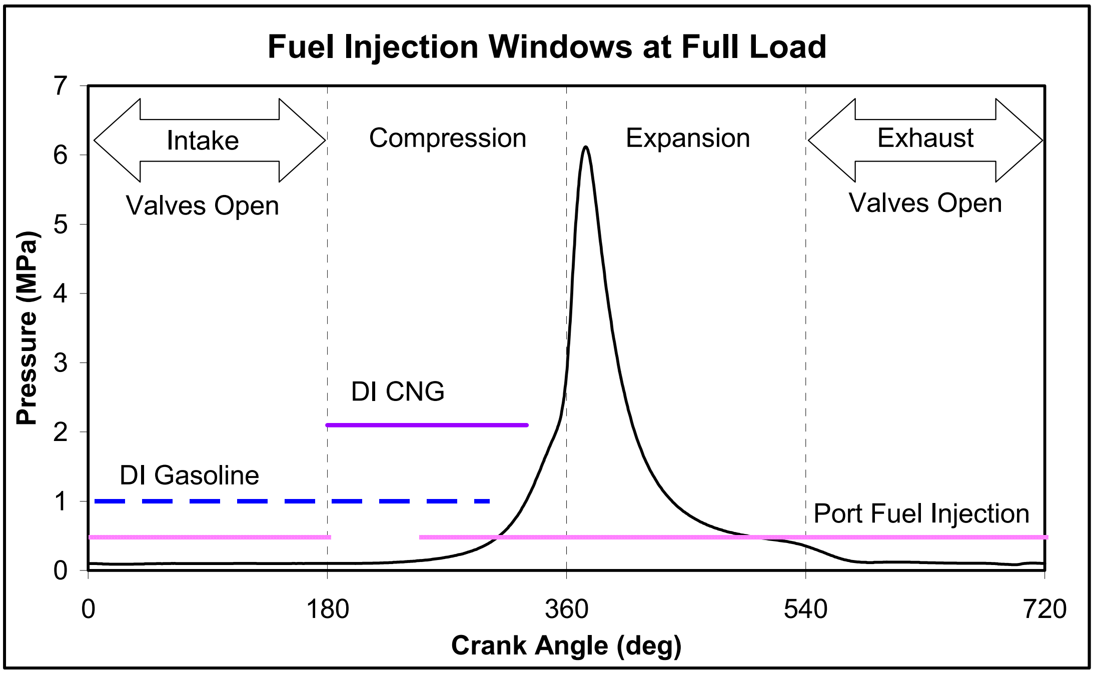

Figure 1 Fuel Injection Windows at Full Load

A prototype DI CNG injector has been developed to fuel a typical 1.8 litre four cylinder four stroke engine. The target flow for this injector was based on the nominal natural gas fuel requirements of the engine as shown in Table 1.

Table 1 Nominal CNG Fuel Requirements for a 450 cc cylinder

Requirement	Naturally Aspirated	Boosted
Minimum Fuel	4.0 mg/pulse	4.0 mg/pulse
Maximum Fuel	36 mg/pulse	80 mg/pulse
Maximum Fuel Duration	4.0 msec	9.0 msec
Nominal fuel flow	9 g/sec	8.9 g/sec

The prototype DI CNG injector was developed based on the Synerject Strata injector. The Strata injector has been developed for gasoline direct injection and has features which made it suitable for development of the prototype DI CNG injector. Initially developed for air-assist direct injection, the Strata injector power group and nozzle were suitable for flowing gas and provide a significantly larger flow area than typical liquid fuel metering injectors. The valve group of the Strata is designed for in-cylinder operation with a valve layout and materials suitable for exposure to combustion pressure and temperature. Table 1 lists the major modifications made to develop the DI CNG injector in comparison to the Strata injector as shown in Figure 2. The solenoid and leg envelope was unchanged from the Strata injector, allowing utilisation of common components already in production and proven in the field as well as facilitating the installation of the DI CNG injector in cylinder heads previously designed for OCP direct injection. The development of a prototype CNG variant of the existing Strata injector offers a relatively low cost 12 volt direct injector using existing production facilities.

Table 2 Injector changes for DI CNG

Feature	Strata	DI CNG
Gas Pressure	650 kPa	2200 kPa
Spring	14 N	40 N
Spring Adjustment	No	Yes
Armature Area	38.6 mm^2	43 mm^2
Stroke	0.23 mm	0.26 mm
Nozzle Area	2.39 mm^2	2.72 mm^2
Opening Time	1.12 ms	1.48 ms
Closing Time	1.27 ms	1.00 ms
Lubricant coating	No	Yes

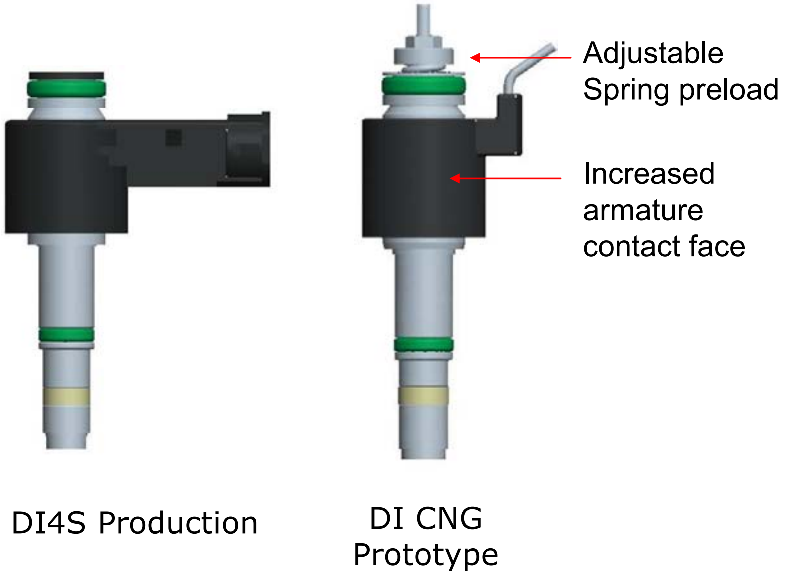

Figure 2 DI CNG Injector Modifications

The DI CNG injector flow was measured on bench using compressed air. The resulting fuel delivery was calculated for methane for two injection pressures. The full load target flow was achieved with a rail pressure of 2200 kPa while a pressure of 1200 kPa was used for metering the minimum fuel. Figure 3 shows delivery of fuel from 4 to 14 mg using 1200 kPa rail pressure and from 12 mg to 80 mg using 2200 kPa.

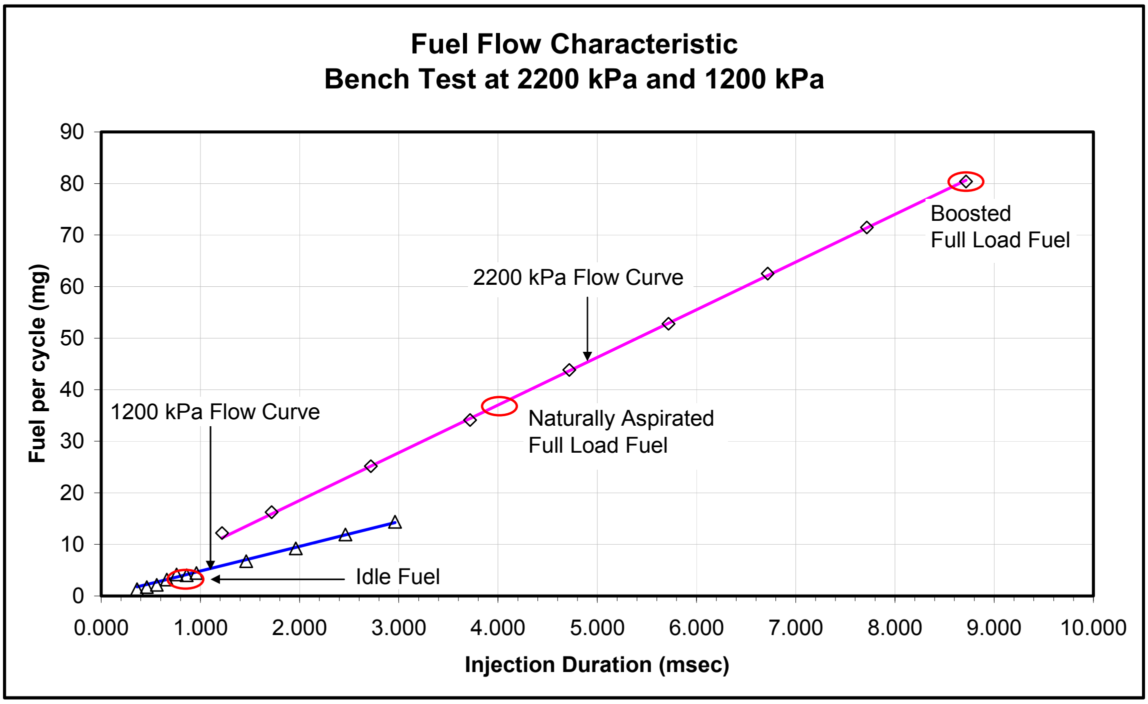

Figure 3   Injector Flow Characteristics

Controlling injection pressure to vary between two values is more problematic for a gaseous fuel than for a liquid fuel due to the compressibility of the gaseous fuel. A gaseous injection system has been designed to achieve rapid switching between injection pressures. This uses two volumes which are used to accelerate the transition between injection pressures, particularly for the transition from the higher to the lower pressure. Schematics of the system are shown in Figure 4 and Figure 5 for both low pressure and high pressure operation. Figure 4 shows the system at low pressure which is the preferred operating mode. In this condition the fuel rail is open to the low pressure volume. When high pressure is required as shown in Figure 5 the pressure in the rail is increased while the valve to the low pressure volume is closed so that it is maintained at the lower pressure.

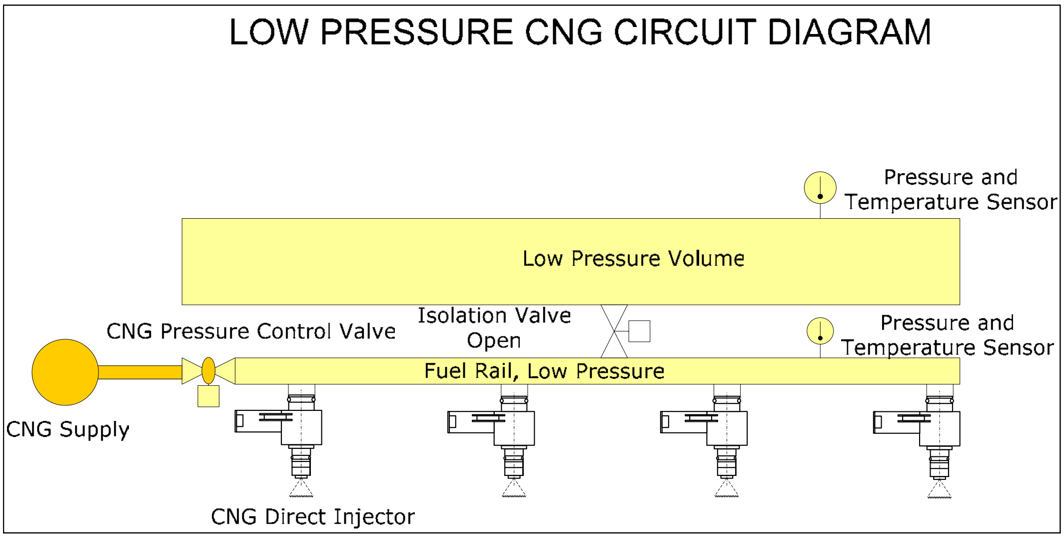

Figure 4 CNG Fuel System Schematic, Low Pressure

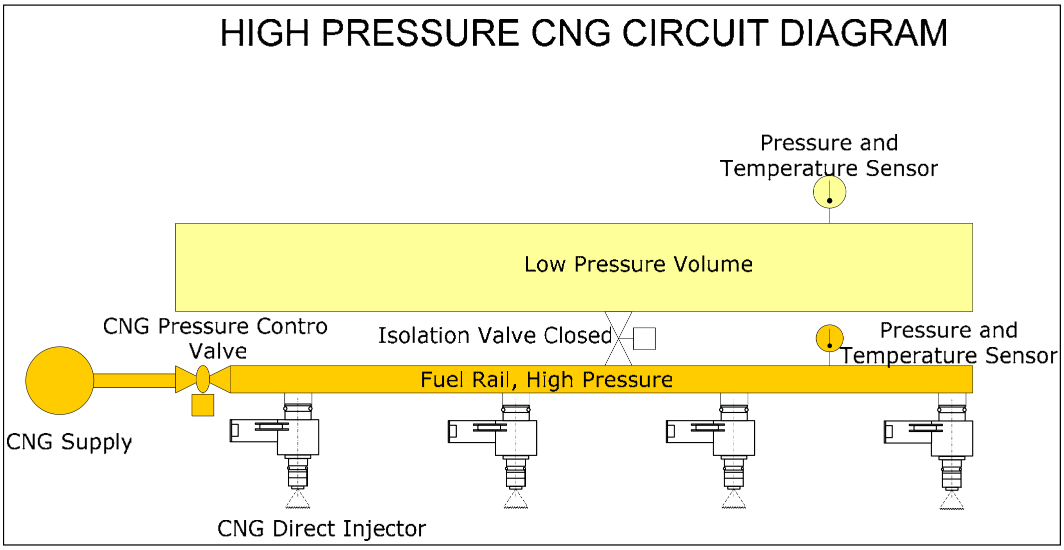

Figure 5 CNG Fuel System Schematic, High Pressure

The transition between high and low pressure was modelled for a typical four cylinder engine with an 8 mm diameter fuel rail connected to a 30 mm diameter low pressure volume. The transition shown in Figure 6 demonstrates the accelerated pressure change when the system is switched from high pressure to low pressure by closing the gas supply valve to the high pressure rail and opening the communication to the low pressure rail. The added volume allows the gas in the rail to expand, reaching the lower pressure in less than one engine cycle. This provides a much accelerated pressure drop when compared to the gradual pressure reduction by the consumption of gas for fuelling the engine. The transition from low to high pressure is much less problematic, as shown in Figure 7. This pressure rise can be achieved within one engine cycle by opening the supply valve to the high pressure rail while isolating the rail from the low pressure volume. The ability to transition between two pressures allows a lower pressure to be used to achieve the fuelling required for idle and low engine load while a higher pressure is used to achieve the flow rate required for direct injection in a limited window at high load.

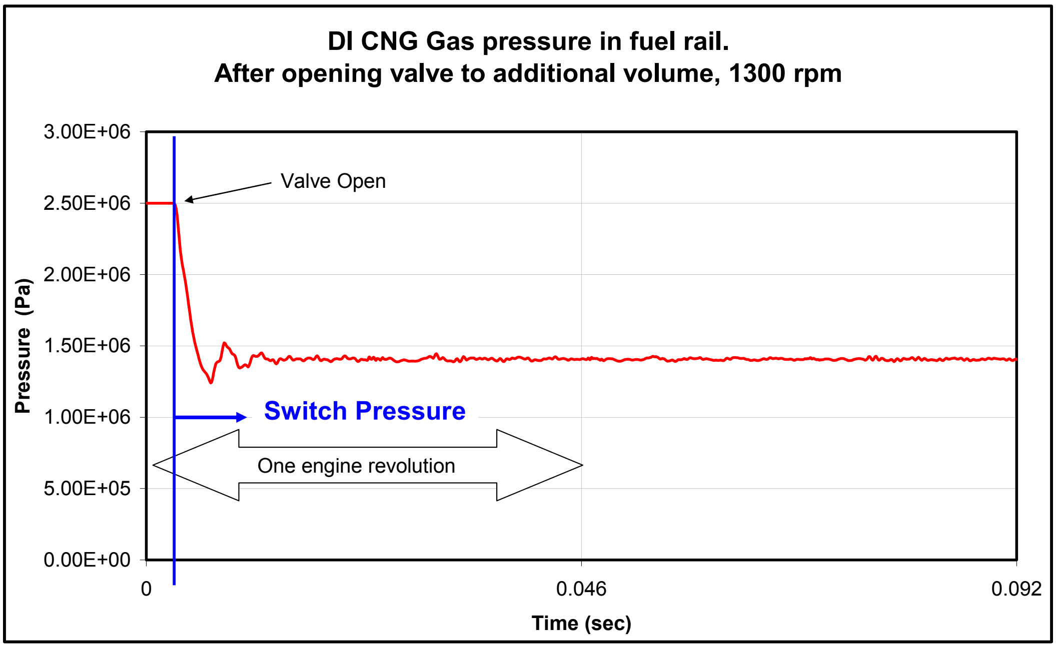

Figure 6 Rail Pressure Transition, High to Low Pressure

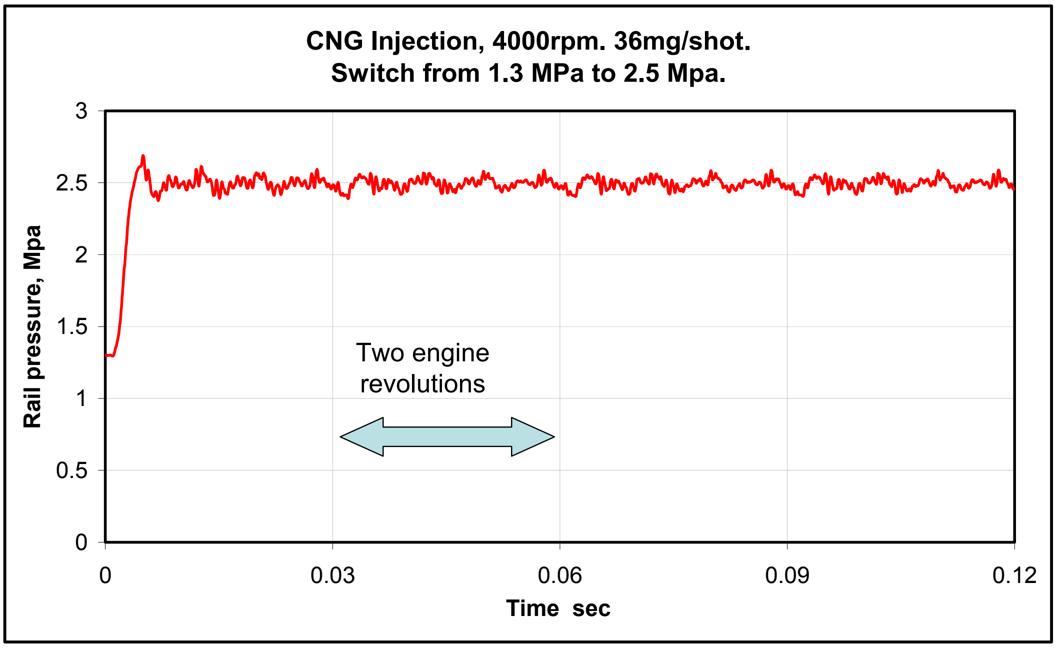

Figure 7 Rail Pressure Transition, Low to High Pressure

Injector Durability

The durability of the injector with dry gas was an issue due to the reduced lubrication. In the standard air-assist injector the fuel provides lubrication which is removed when operating on dry gas such as CNG. This effect was initially investigated on unmodified air injectors. The primary durability issue was the increase in stroke over time which led to a shift in metering. The major contribution to stroke increase was found to be the wear or deformation of the contact faces in the primary air gap of the solenoid. In the DI CNG injector the primary air gap in the magnetic circuit was increased in area to reduce the contact stress. In addition to this a modified injector driver waveform was developed to reduce the impact of the armature on opening and of the poppet on closing. Figure 8 shows the modified poppet movement and reduced impacts when operating with this injector driver.

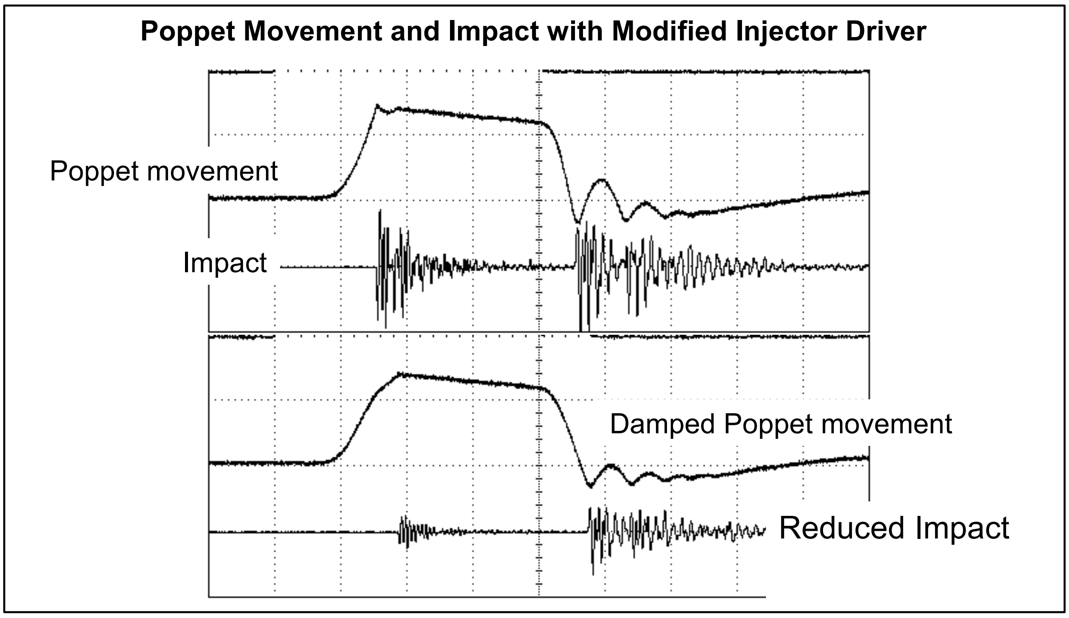

Figure 8 Injector Actuation with Modified Driver

Durability testing of injectors was conducted on bench with dry air. The injected airflow was measured periodically during the test to monitor changes and the injector stroke was measured before and after the test. Figure 9 and Figure 10 compare the performance of the prototype DI CNG injectors to that of the standard injectors. This shows that the stroke and flow of the standard injector increased over 20 million cycles of testing while DI CNG injectors remained unchanged over 26 million cycles. This provided a prototype DI CNG injector with the durability and flow capability for initial engine development.

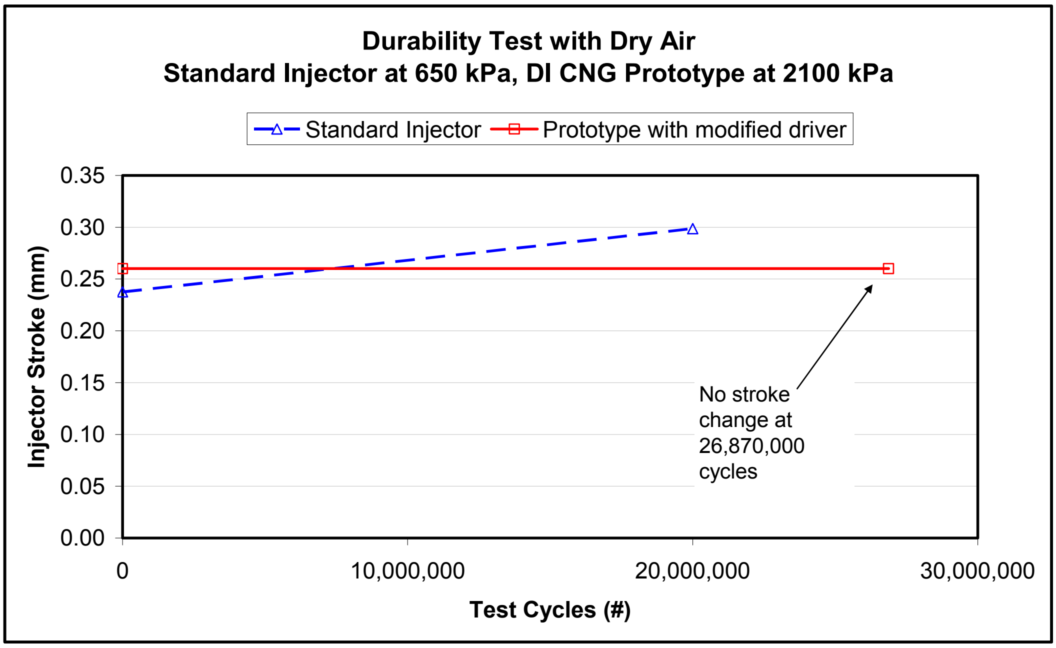

Figure 9 Injector Durability Test

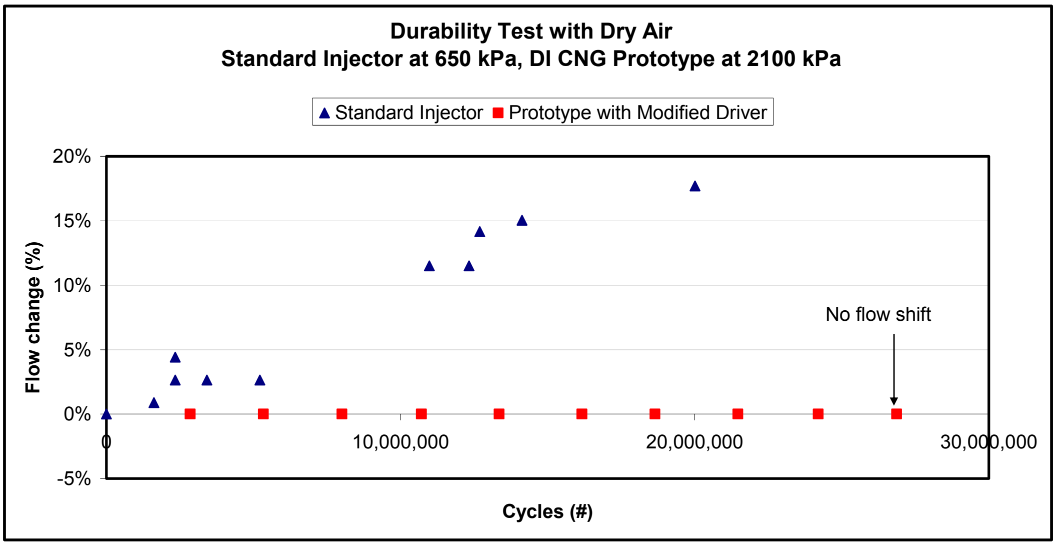

Figure 10 Injector Flow Shift

Engine Testing

Testing was undertaken on a naturally aspirated 450 cc single cylinder engine to compare the performance of DI CNG to performance with manifold injected CNG and to performance with manifold injected gasoline. Table 3 lists the main features of the test engine which incorporated a centrally positioned direct injector in close proximity to the spark plug. The engine was operated with a compression ratio of 10.7:1 to suit manifold injected gasoline. The compression ratio of the engine was not altered to utilise the combustion characteristics of CNG which may typically have a research octane number of 130 compared to a range of 91 to 98 for gasoline. Table 4 lists some typical values for CNG compared to gasoline. Figure 11 shows typical locations of the injector and spark plug centrally in the combustion chamber. The engine was configured with no swirl and low tumble as is typical for stratified operation with central injected spray guided gasoline direct injection.

Table 3 Development Engine Major Features

Feature	Value
Engine Displacement	450cc
Compression Ratio	10.7:1
Combustion Chamber	Pent-roof type with Orbital modifications
Fuel System	Orbital DI4 central injection
Spark Plug	M12
Inlet Valve Open	364 Deg atdc
Inlet Valve Close	583 Deg atdc
Exhaust Valve Open	133 Deg atdc
Exhaust Valve Close	357 Deg atdc

Table 4 Typical Fuel Properties

	CNG (Methane)	Gasoline (ULP)
Heating Value (MJ/kg)	~47.0	~42.5
Octane Rating (RON)	130	91
Octane Rating (MON)	115	81
Stoichiometric ratio (by mass)	17.2	14.6

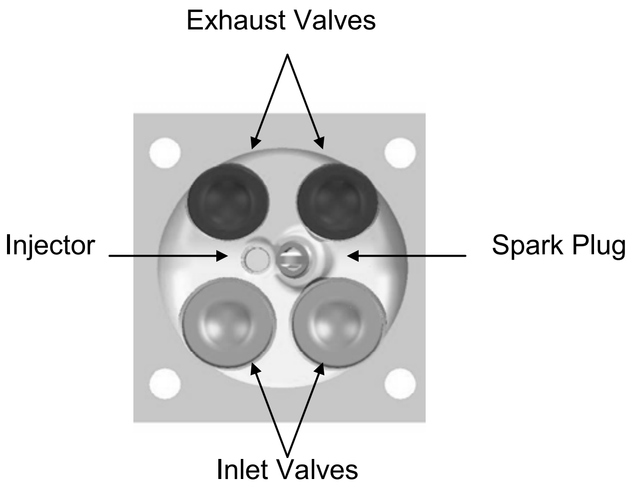

Figure 11 Central Direct Injection Layout

Manifold Injected Baseline Performance

Testing was conducted both with manifold gasoline injection and manifold CNG injection to determine the baseline performance for comparison with DI CNG operation. Figure 12 and Figure 13 show the baseline engine performance for both gasoline and CNG. N.IMEP was calculated from the logged cylinder pressure and utilised as the measure of engine performance. A performance drop of 9% to 13% was noted when operating with CNG. Operating richer than stoichiometry with gasoline produced increased performance, with a 0.4% to 6.7% benefit relative to stoichiometric operation with gasoline. Stoichiometric operation with gasoline was conducted with an AFR of 14.5:1 while stoichiometric operation with CNG was conducted with an AFR of 17.2:1 and rich operation with gasoline was conducted with an AFR of 12.5:1. A major contribution to the performance loss with CNG was the reduced engine airflow as shown in Figure 14. Engine airflow per cycle was measured using a LFE on the engine intake. For stoichiometric operation with CNG the engine airflow was reduced by approximately 9%.

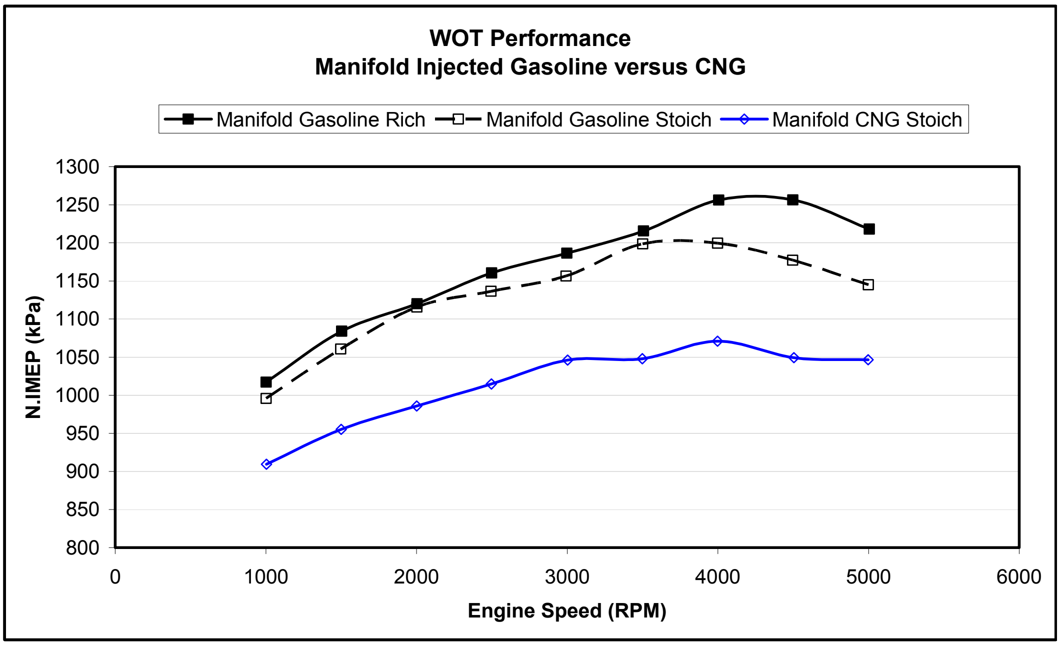

Figure 12 Baseline Engine Performance, Gasoline and CNG

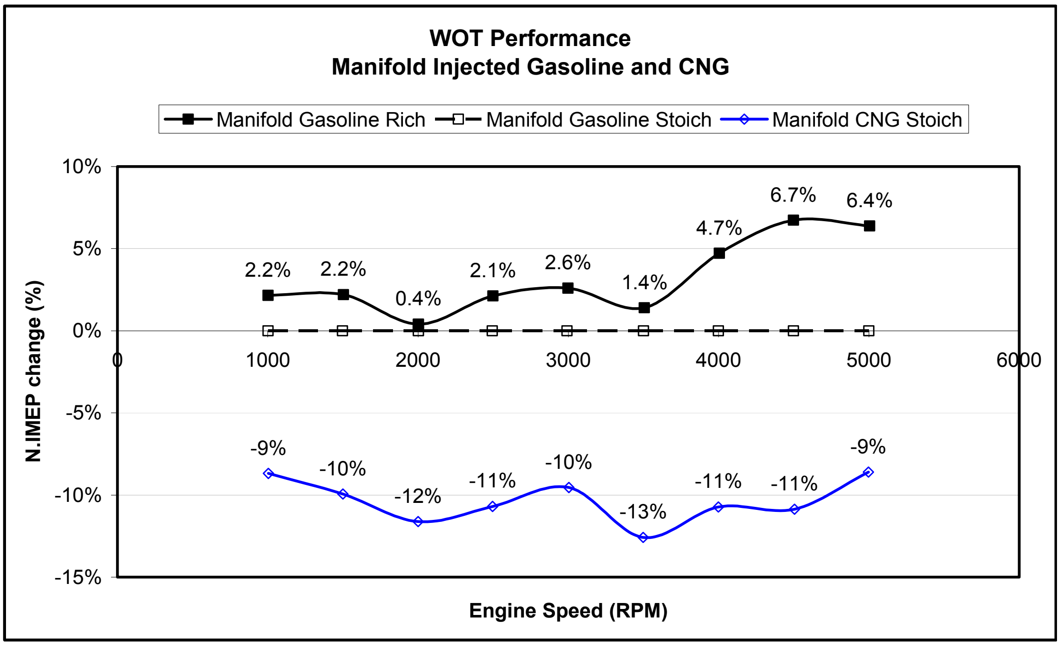

Figure 13 Baseline Performance Comparison

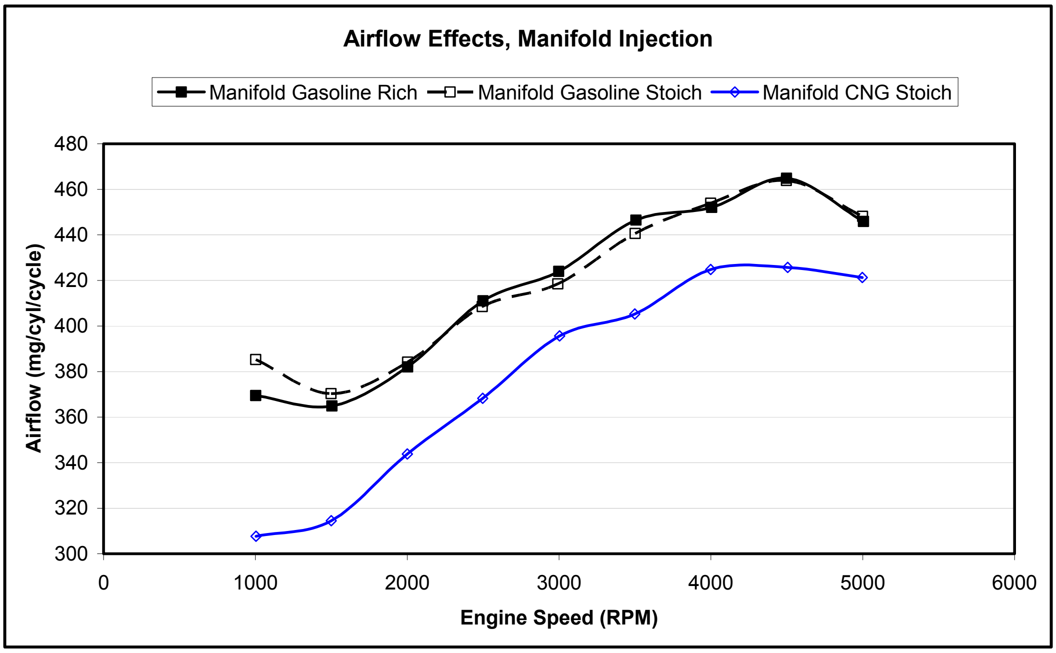

Figure 14 Manifold CNG Effect on Airflow

<u>Direct Injected CNG Performance</u>

Direct injection of CNG was investigated using a prototype DI CNG injector at 2000 kPa injection pressure. Testing was conducted in the SCRE engine as configured for manifold injected gasoline with a 10.7:1 compression ratio. As noted previously the engine had no swirl and low tumble and it is likely that increased in-cylinder motion would be beneficial for combustion of CNG. Furthermore this initial testing did not include optimisation of the injector nozzle geometry to improve the in-cylinder mixing of the gas. The primary objective of this testing was to investigate improved volumetric efficiency through increasing the quantity of trapped air. At each test speed the effect of injection timing on air displacement was investigated by scanning from early to late injection. At each injection timing the fuel pulse width was adjusted to maintain a stoichiometric AFR of 17.2:1 and the ignition advance was optimised for MBT. Figure 15 shows the effect of injection timing on the airflow and engine performance at 1000 rpm. The engine airflow increased as the start of injection was retarded toward inlet valve closure. The airflow increased up to 11% with injection after inlet valve closure with a corresponding 10% increase in N.IMEP as shown in Figure 16. Figure 17 shows a similar effect due to late fuel injection timing at 3000 rpm with a 9% increase in both engine airflow and performance as a result of injection after inlet valve closure.

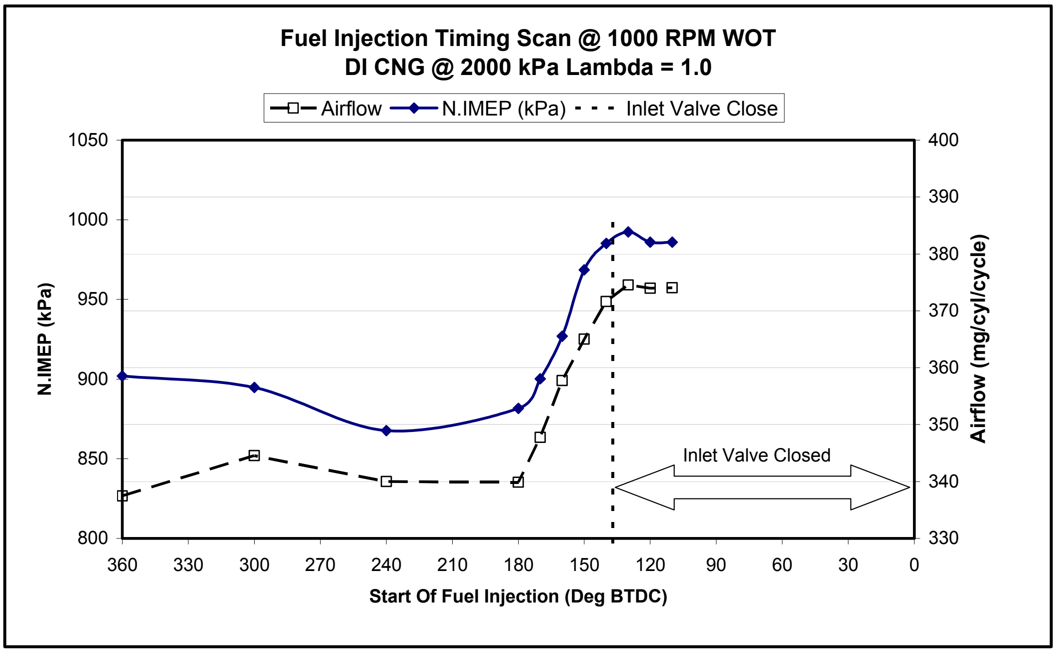

Figure 15 Injection Timing Effects at 1000 rpm

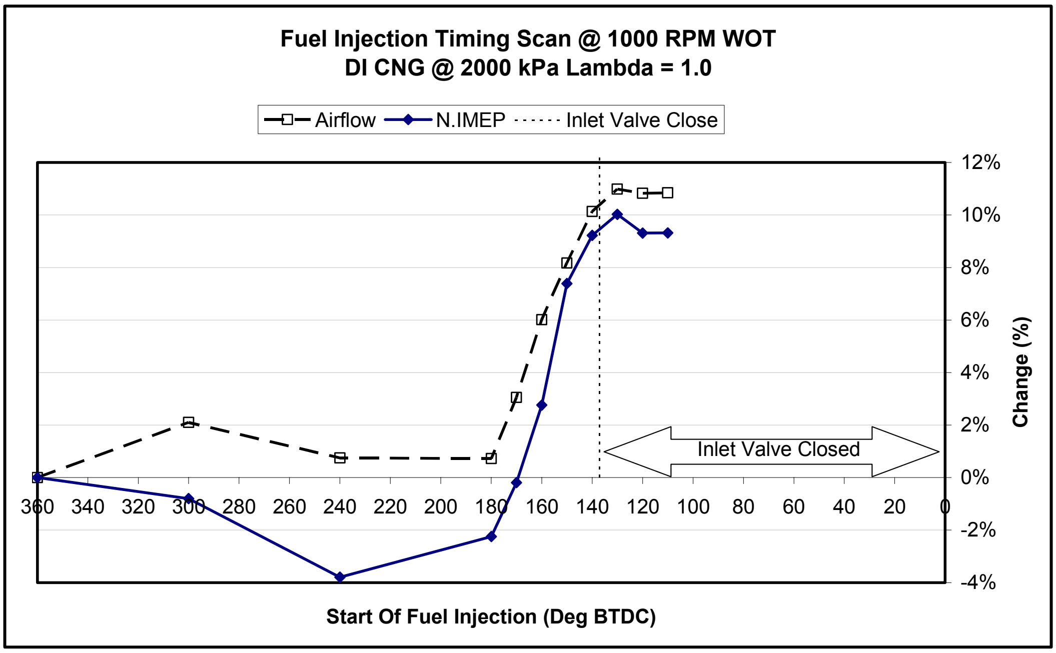

Figure 16 Injection Timing Effects at 1000 rpm

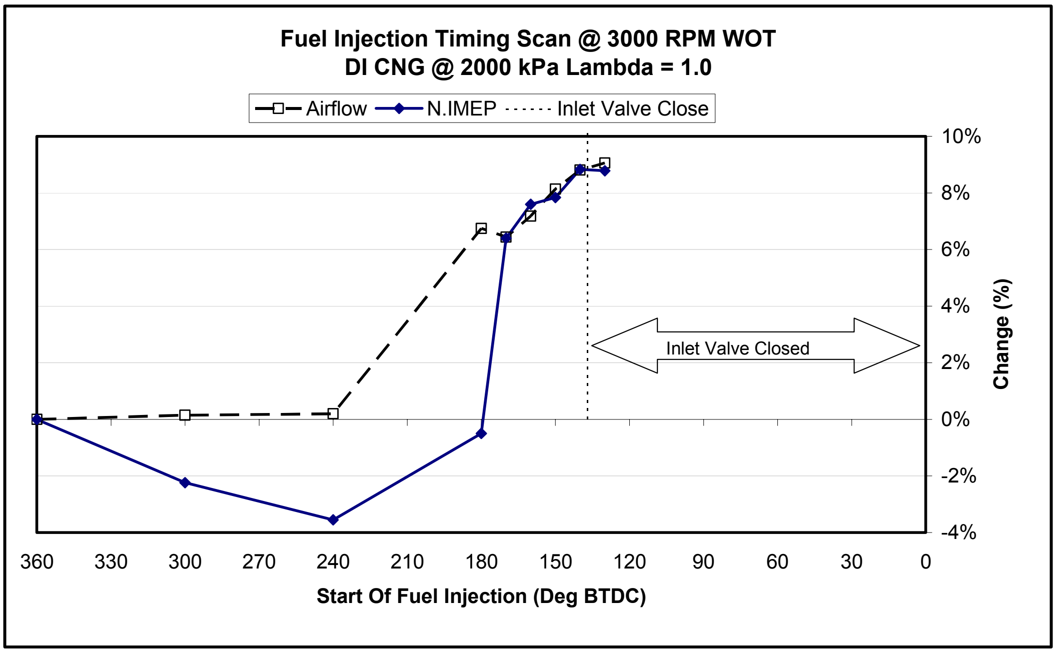

Figure 17 Injection Timing Effects at 3000 rpm

At higher speeds retarded injection timings continued to provide increased airflow, but did not result in the same gains in performance. Figure 18 shows the effect of injection timing on airflow and performance at 4000 rpm. At 4000 rpm the airflow increased by up to 10% with retarded injection timing but the performance gain was only 4%. The maximum N.IMEP was achieved with more advanced injection timing than the optimum for airflow. At 4000 rpm the optimum performance was achieved with an injection timing of 170 deg BTDC compared to optimum injection at 130 deg BTDC at 1000 rpm. Figure 19 shows the effect of injection timing on airflow and performance at 5000 rpm. At this speed the potential for increased airflow was again evident as the injection was retarded, but the engine performance dropped at later injection timings so that only a small increase in airflow could be utilised. This suggests that factors other than reduced volumetric efficiency are contributing to performance loss at higher speeds. Figure 20 compares the effect of injection start time on carbon monoxide emissions represented by Gross Indicated Specific CO for speeds of 1000, 4000 and 5000rpm. The rising CO at retarded injection timings indicates incomplete combustion due to the reduced mixing time. The combustion at higher engine speeds would be expected to benefit from shorter injection durations and increased in cylinder motion to promote mixing and burn rate.

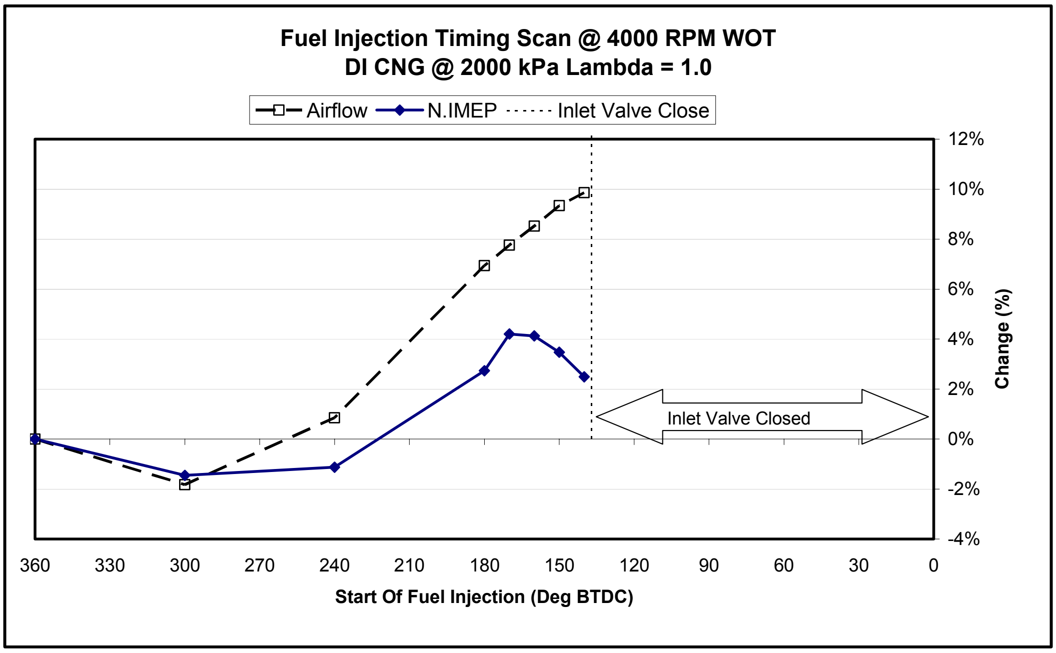

Figure 18 Injection Timing Effects at 4000 rpm

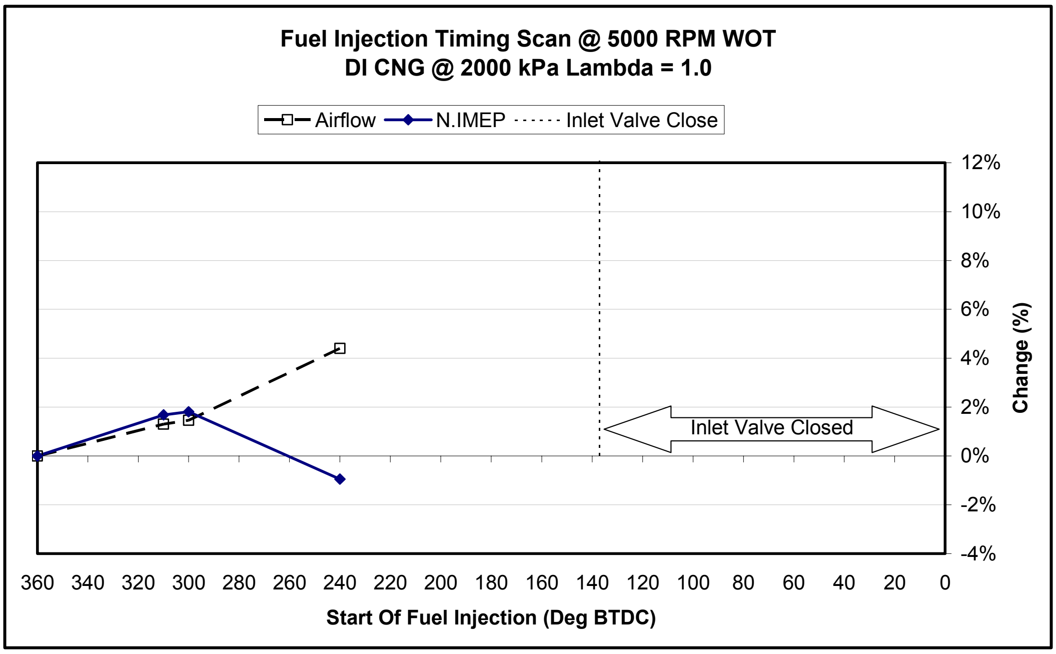

Figure 19 Injection Timing Effects at 5000 rpm

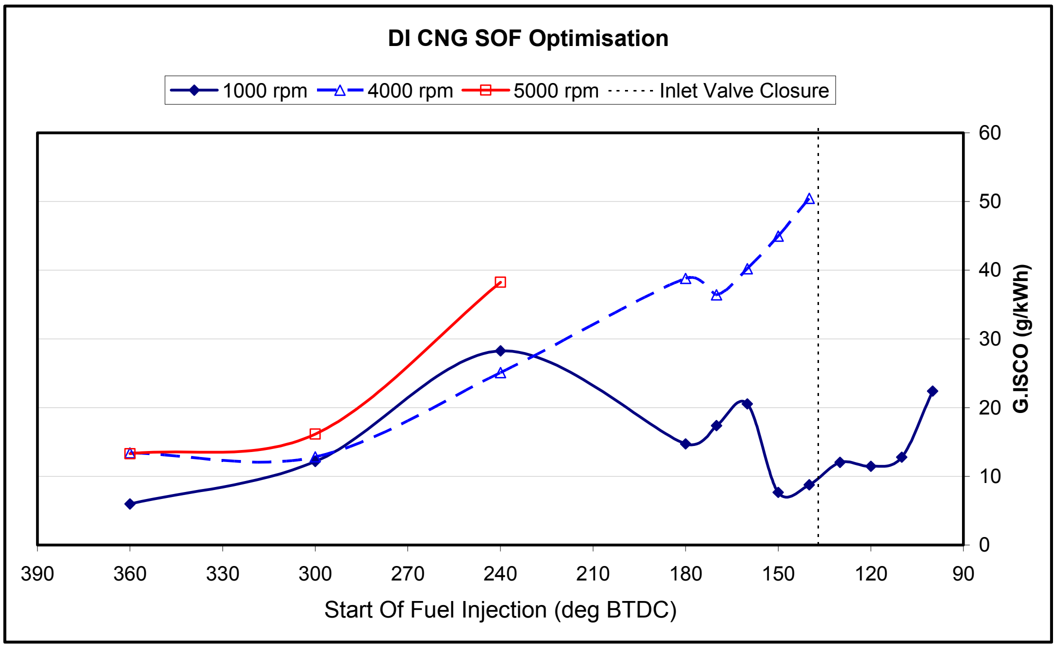

Figure 20 Injection Timing Effect on Carbon Monoxide

Figure 21 shows the optimised direct injection timings selected to achieve maximum N.IMEP. These show injection predominantly or partially after inlet valve closure for speeds below 4500 rpm. At 5000 rpm the optimum injection was prior to inlet valve closure.

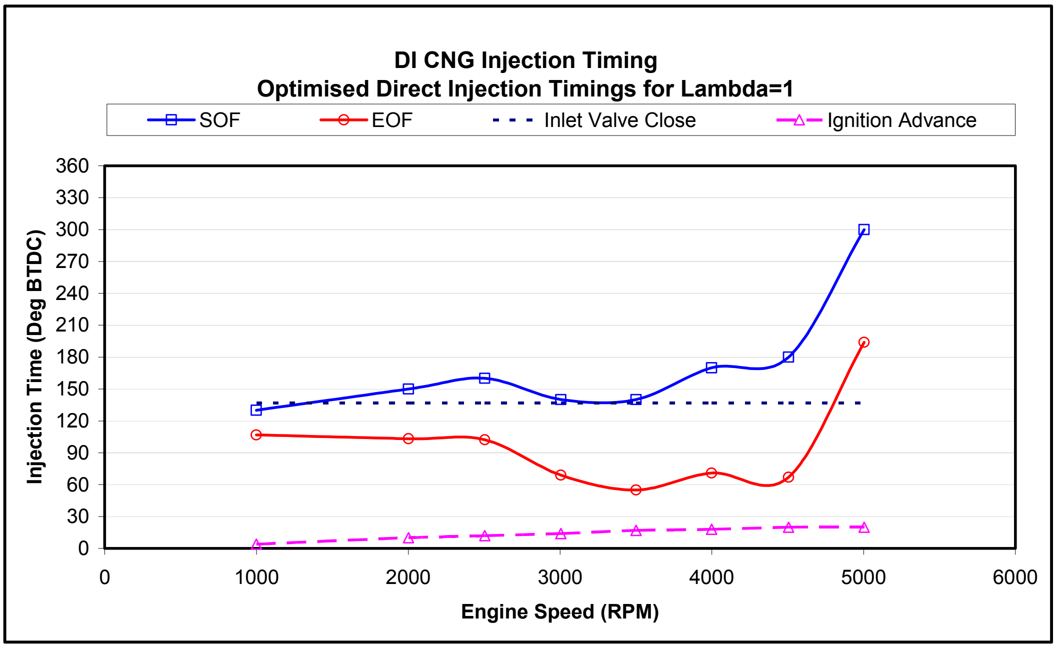

Figure 21 DI CNG Optimised Injector Timings

<u>DI CNG Performance compared to Manifold Injected Gasoline</u>

The major benefit attributed to direct injection of CNG was the improved engine airflow. Figure 22 and Figure 23 show the engine airflow for each major configuration tested. The baseline test with manifold gasoline injection at $\lambda=1$ is used as the reference. This shows that the engine airflow with late injected CNG matched that of manifold injected gasoline for speeds up to 4000 rpm. The drop in engine airflow at 4500 rpm and 5000 rpm was due to the reduced capability to utilise late injection at these speeds.

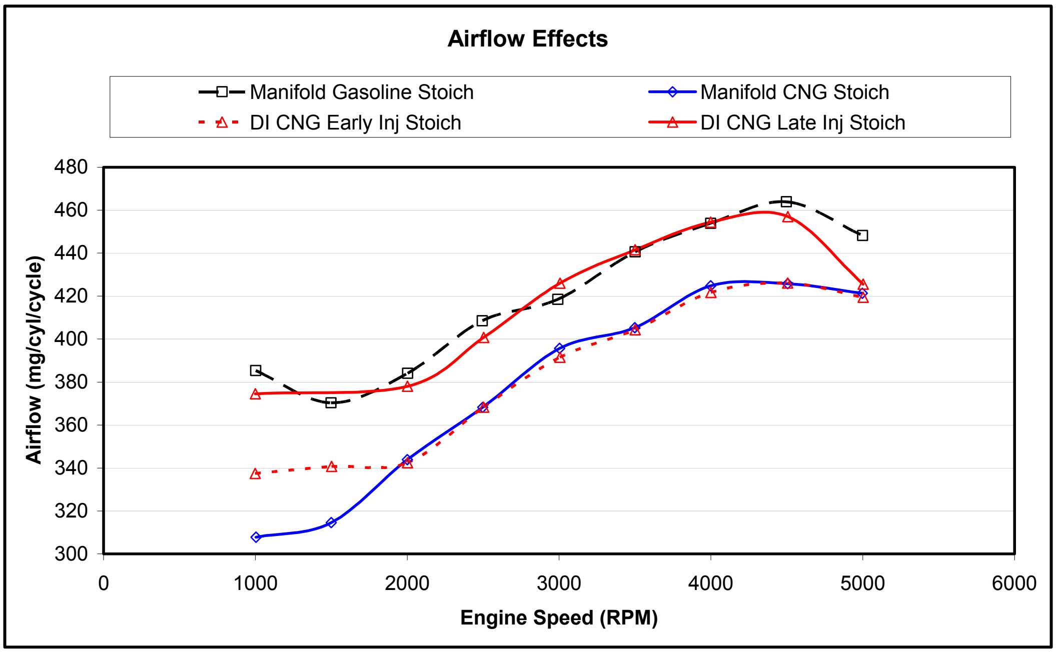

Figure 22 Airflow Effects

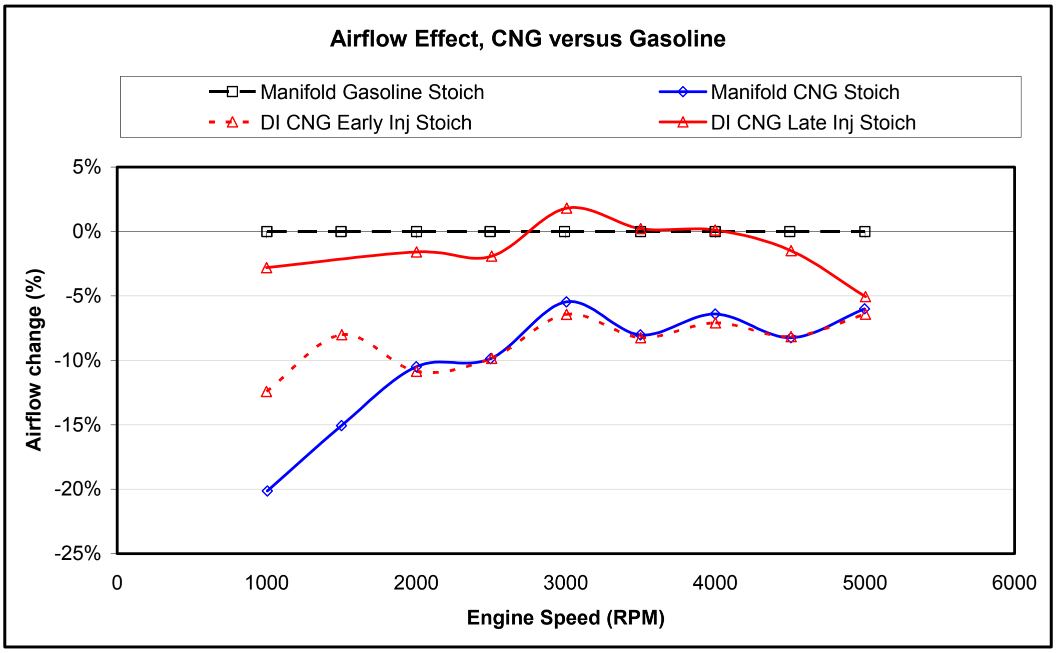

Figure 23 Airflow Effect Summary

The improved engine airflow with DI CNG resulted in improved performance compared to manifold injected CNG across all speeds tested. Figure 24 and Figure 25 compare the resulting engine performance with DI CNG to the performance with gasoline. The performance with DI CNG was within 3% of stoichiometric gasoline performance at low speeds and within 7% at higher speed. The ability to utilise late injection appeared to be limited at higher speeds by the time required for mixture preparation prior to ignition. The rich calibration with gasoline provided up to 7% benefit compared to stoichiometric calibration. Some further investigation into the optimum AFR for performance with DI CNG appears warranted.

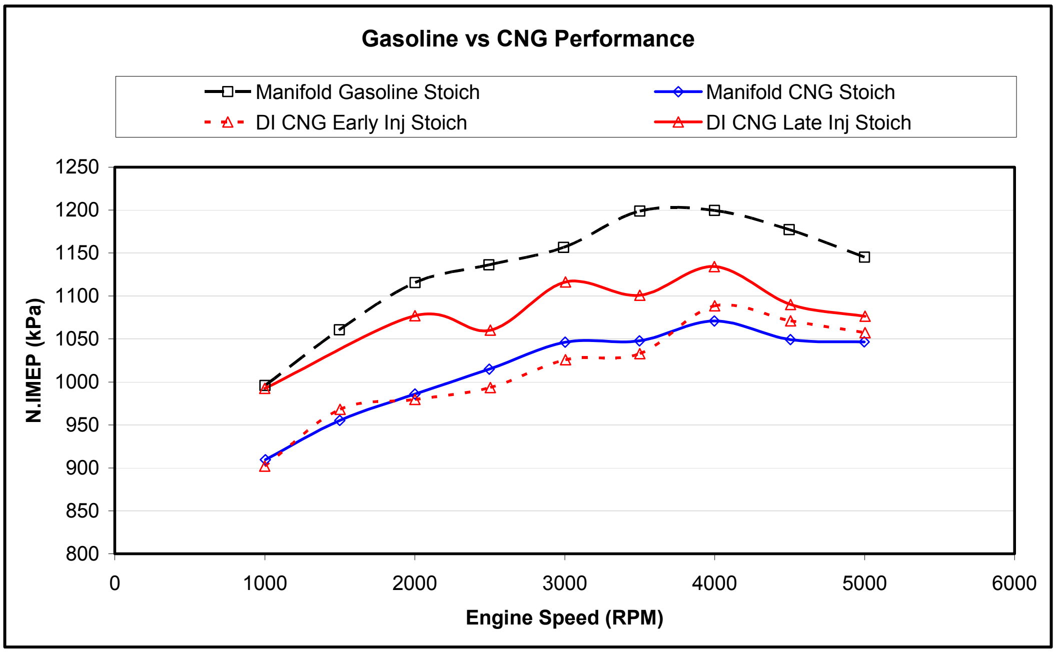

Figure 24 Performance Summary

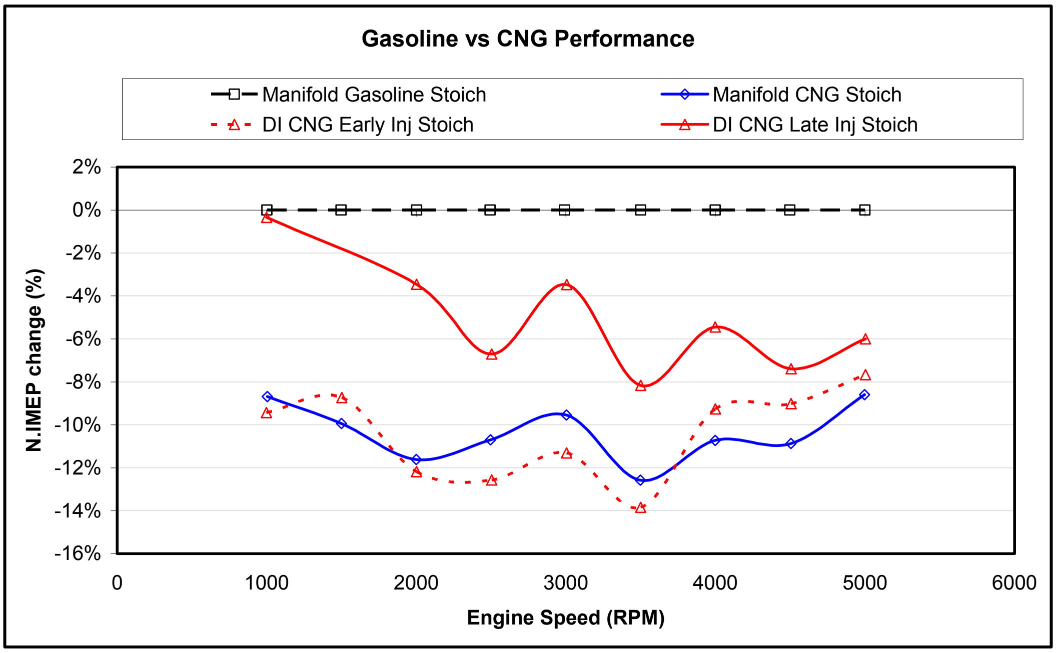

Figure 25 Performance Effect Summary

The maximum fuel used for the DI CNG application was 31.5 mg/cyl/cycle. This indicates that a target fuel capability of 36mg/shot was sufficient for a 450 cc DI CNG naturally aspirated engine. It would be expected that a multi cylinder engine would gain some cylinder charging at the tuned frequency of the inlet system, increasing the trapped airflow and hence the fuelling requirement. Future testing on a boosted test engine will confirm the typical fuelling required for boosted DI CNG applications.

Figure 26 shows the optimised ignition timings used for the stoichiometric calibrations of DI CNG, manifold injected CNG and gasoline. This shows the considerably advanced ignition used for CNG indicating the knock resistance of the fuel and the potential for increased compression ratio for a dedicated DI CNG engine.

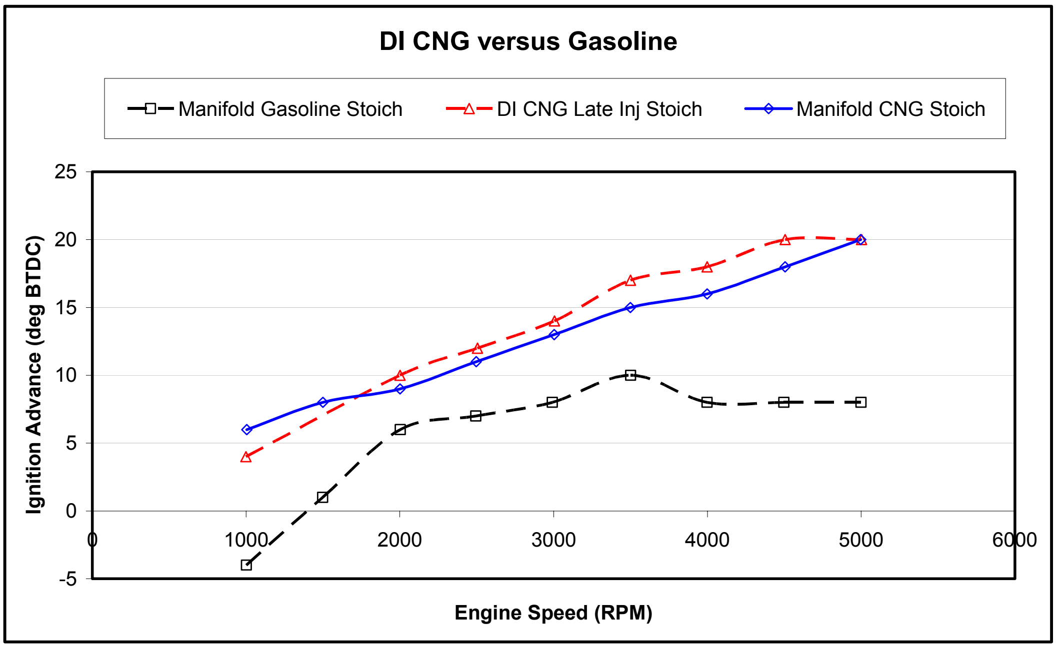

Figure 26 CNG effect on Ignition Advance

Conclusions

DI CNG Injector

A prototype direct injector for CNG has been developed which is suitable for fuelling a typical engine of up to 450cc capacity per cylinder. This injector has capability to fuel naturally aspirated engines and boosted engines of this capacity. The fuelling range from the minimum to maximum fuel requirement of the engine was achieved via variation of the gas injection pressure. A system for rapidly transitioning between pressures has been modelled showing transition between operating pressures within one engine cycle. The prototype injector durability has been tested to 26 million cycles on dry air showing good control of stroke and injected gas flow.

Engine Testing

Direct injection appears to have promising potential for improving the performance of natural gas engines through improved volumetric efficiency. Engine testing has been completed on a single cylinder research engine of 450cc capacity and 10.7:1 compression ratio. This showed the effect of reduced engine airflow and performance for manifold injection of CNG. Direct injection of CNG with late injection timings was shown to improve engine airflow up to 10%. At higher speeds the capability to utilise late injection timings was limited by the requirement to advance the injection to achieve suitable mixture preparation. The resulting benefit in engine performance was up to 10% at low speeds with the benefit reducing to 4% at 5000 rpm. These results were achieved with an engine compression ratio suited to operation with gasoline and may be improved on a dedicated CNG engine with optimised compression ratio and in-cylinder air motion to suit the combustion characteristics of natural gas. Further improvement would also be expected from optimisation of the direct injector nozzle to enhance mixture preparation, particularly at higher engine speeds.

The characteristic of maximised performance at low speeds may be particularly beneficial to the driveability of turbocharged engines which tend to achieve boosted performance at higher engine speeds. In addition, the combination of direct injection and turbocharging offers particular benefits which have been documented for gasoline fuelled engines (7). For a turbocharged direct injected CNG application the improved volumetric efficiency from direct injection could be used to improve low speed torque and to reduce the effect of turbo lag. The improved volumetric efficiency of DI CNG may be utilised in dual fuel applications to minimise the performance difference between gasoline and CNG where the engine compression ratio is selected to suit operation on gasoline.

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Definitions, Acronyms, Abbreviations

CNG	Compressed Natural Gas
NOx	Nitrous Oxide
IMEP	Indicated Mean Effective Pressure
DI	Direct Injected
AFR	Air Fuel Ratio
SCRE	Single Cylinder Research Engine
MBT	Minimum for Best Torque
BTDC	Before Top Dead Centre
ppm	parts per million
OCP	Orbital Combustion Process
LFE	Laminar Flow Element
WOT	Wide Open Throttle
SOF	Start Of Fuel

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