

Effects of Charge Motion, Compression Ratio, and Dilution on a Medium Duty Natural Gas Single Cylinder Research Engine

Jennifer Wheeler, Joshua Stein, and Gary Hunter
AVL Powertrain Engineering Inc

ABSTRACT

Recent advances in natural gas (NG) recovery technologies and availability have sparked a renewed interest in using NG as a fuel for commercial vehicles. NG can potentially provide both reduced operating cost and reductions in CO₂ emissions. Commercial NG vehicles, depending on application and region, have different performance and fuel consumption targets and are subject to various emissions regulations. Therefore, different applications may require different combustion strategies to achieve specific targets and regulations. This paper summarizes an evaluation of combustion strategies and parameters available to meet these requirements while using NG in a spark ignited engine.

A single-cylinder research engine using a modified diesel cylinder head was employed for this study. Both stoichiometric combustion with cooled exhaust gas recirculation (EGR) and lean-burn were evaluated. Testing quantified the effects of dilution (EGR or excess air), swirl ratio, and compression ratio on the operating limits, engine-out emissions, and fuel efficiency at operating conditions typical for heavy-duty vehicle applications.

The data acquired indicates that lean-burn is a viable option for moderate NOx emission applications such as Euro IV and V. Stoichiometric combustion enables the use of a three way catalyst which provides low tailpipe NOx as required for more stringent standards. Stoichiometric combustion with cooled EGR enabled greater full load torque capability than lean combustion, but lean combustion provided improved fuel efficiency. In both cases, higher compression ratio resulted in an increase of efficiency but reduced full load capability, and increased swirl extended the dilution capability.

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BACKGROUND

Over the last 20 years there has been a transformation in NG recovery technology that has significantly increased access to natural gas reserves. Consequently gas prices have fallen to levels not seen in over 10 years. Although natural gas has been used for a long time as a transportation fuel, energy density and distribution have restricted this usage to niche applications.

Recently the advent of more stringent emissions legislation has caused the cost of the aftertreatment system for diesel fuelled vehicles to increase dramatically. Consequently, the business case for NG fuelled vehicles has become attractive.

In large power generation and pipeline pumping applications, dedicated NG engines have been available for many years. However the market for smaller commercial dedicated NG engines is still developing; therefore the engines currently available fall into 2 categories. The first is converted gasoline engines; the second is converted diesel engines.

Few gasoline engines exist with specific displacements above 0.75 liters (l) per cylinder. Thus most NG engines above 0.75l per cylinder are based on diesel units. When converting a diesel engine to run on natural gas a number of factors need to be considered, including engine thermal load, compression ratio (CR), method of gas introduction, ignition system, and emissions requirements.

Diesel engines run with excess air, however, most on-road spark ignited engines run at stoichiometry to enable use of three way catalyst (TWC) aftertreatment technology. If a base diesel engine is operated as a spark-ignited (SI) NG engine at stoichiometry, the thermal load on the engine is higher than that of the initial design intent of the engine. Therefore methods need to be taken to reduce combustion temperature. This is typically done with either the addition of exhaust gas recirculation (EGR) or with excess air. Lean burn NG engines have a thermal efficiency advantage over stoichiometric engines; however the exhaust emissions prevent compliance with the most stringent emissions standards with the exception of lean NOx aftertreatment such as selective catalytic reduction (SCR).

(SCR) system. If EGR is applied to a stoichiometric engine then a relatively inexpensive and highly effective TWC can be used.

Previous studies comparing lean burn and EGR show that EGR with a TWC can offer large reductions in NO_x and HC emissions [1, 2]. For US 2010 and EU Euro VI on road emissions, the EGR converted diesel engine is emerging as the mainstream solution. Recent examples include the Cummins Westport ISX12G and the Scania 9L Euro VI.

INTRODUCTION

The effects of dilution, CR, and swirl ratio on net mean effective pressure (NMEP) capability, thermal efficiency, engine-out emissions, and combustion stability in a diesel engine converted to run on natural gas (NG) are examined in this paper. Both the stoichiometric mixture with cooled EGR and the lean burn mixture without EGR dilution strategies were investigated.

The main limiting factors for NG combustion are exhaust gas temperature (EGT), pre-ignition (PI), knock, combustion stability, and compliance with emissions regulatory standards. In addition, adherence to these limitations is continually being balanced with optimal engine performance and fuel consumption goals.

The chemistry of NG combustion produces relatively low total hydrocarbon (THC) and very low particulate matter (PM) emissions. The engine knock tolerance with NG is much higher than with gasoline due to its high octane (~120 RON) [3], enabling SI combustion in large displacement cylinders utilizing converted diesel engines without excessive knock.

As dilution is introduced in SI applications, reduction in fuel consumption is realized due to reduction in combustion and burned gas temperature, which leads to reductions in heat transfer losses and losses due to variation in specific heat and molecular dissociation effects [3,4]. Moreover, cooled EGR or excess air dilution reduces the tendency for PI and further improves knock tolerance, which leads to improved combustion phasing and stability. Increased dilution also reduces pumping losses, which further improves thermal efficiency.

However, increased dilution presents challenges regarding combustion stability. Flame propagation rate declines significantly at the dilute limit where the flame temperature falls below ~1900 K [4], and there is an upswing in THC. Conversely, because of the lower combustion temperatures there is a decline in NO_x formation. Overcoming the challenges presented with dilute combustion can be facilitated with the addition of increased turbulent kinetic energy (TKE), by way of increasing charge motion. An increase of TKE results in higher flame speed and shortens the burn duration, compensating for

the effects of dilution on these parameters. Although increased swirl ratio was used in this study to increase the bulk TKE, tumble charge motion, which is common in modern gasoline turbocharged direct injection (GTDI) engines, could also have been used to accomplish the same result.

The effects of both lean and EGR dilution strategies on engine performance, fuel consumption, and emissions are discussed in detail in this paper.

EXPERIMENTAL SET-UP

Engine Hardware

The current work used the AVL 580 single-cylinder research engine (SCRE) and combustion system hardware from a production 4.5L 4-cylinder Diesel engine. [Table 1](#) provides specific geometrical parameters of the test engine and combustion system.

Table 1. Engine specifications

Displaced volume	1.125 L
Stroke	127 mm
Bore	106.5 mm
Connecting Rod	203 mm
Compression ratio	10:1, 12:1
Number of Valves	4
EVO/EVC	45° bBDC/15° aTDC @ 0.1 mm lift
IVO	0° bTDC/60° aBDC @ 0.1 mm lift

Valve timings were set at fixed values, with the intake valve opening (IVO) occurring at 0° CA aTDC at 0.1 mm of lift, and the exhaust valve closing (EVC) occurring at 15° CA aTDC at 0.1 mm of lift. Intake and exhaust valve open duration was 240° CA. Minimal valve overlap was important to prevent blow-through of the NG fuel under boosted conditions and this was used to define the fixed valve timing. The valve lifts and durations are typical of a spark ignited engine application and were chosen based on hardware availability.

The test cell used was one of AVL's single cylinder research test cells, which is equipped with all necessary test and measurement equipment and single cylinder controls functionality, including boosted air and external EGR systems.

The pistons used for testing incorporated a concentric bowl-in-piston design with 10:1 and 12:1 CR, as shown in [Figure 1](#). These two pistons were designed with a 35.5% squish region, corresponding to a bowl diameter of 78 mm which was limited by the existing diesel piston mold geometry. Greater bowl depths were not possible due to a minimum allowable metal thickness of 7 mm, and therefore the maximum bowl depth was limited to 20 mm.

The engine used a PFI injector setup specifically designed for use with NG. This consisted of two off-the-shelf Bosch NGI2 K injectors located downstream of the intake buffer tank and 560 mm upstream of the engine. This injector placement was chosen based on the desire to provide an adequate mixing length which resulted in stable steady-state operation, as transient response was not of interest for single-cylinder testing. The injector has a static flow rate (Q_{stat}) of 10.4 kg/hr at 7 bar injection pressure.

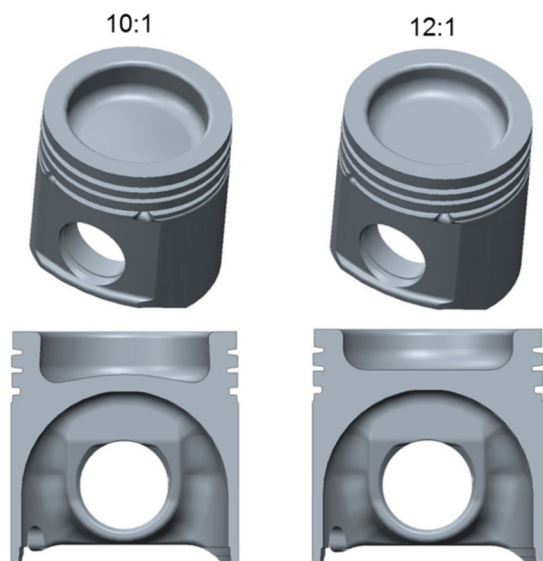


Figure 1. Piston Variations

The ignition coil used was a Bosch ZS-K 1X1E. This coil has an integrated power stage and therefore only requires the ECU signal. The coil also allows variable energy and duration. Spark energy of up to 132.4 mJ and spark duration of up to 2.21 ms is available using this coil. This potential is important in a highly dilute boosted application, as high cyclic variability can occur due to inconsistent flame kernel development. A higher spark energy and duration can help reduce the flame development time and improve combustion stability under highly dilute conditions.

A dedicated NG engine M14 spark plug was desired. However, due to packaging considerations, the spark plug chosen for this project was a Denso M12 iridium type (Denso IXU22). The spark protrusion was set to 3.3 mm. A 3 to 5 mm protrusion was desired for improved flame development. This spark plug had a 19 mm reach, and used a 1 mm gap. The gap was decreased to 0.375 mm to overcome challenges with misfiring under highly dilute boosted conditions.

Two cylinder heads with alternate intake port geometries were used to investigate the effect of increasing bulk in-cylinder charge motion on combustion behavior. The port geometry of the production 4.5L cylinder head results in a measured swirl ratio of 0.6. A second production cylinder head was modified to provide a higher swirl ratio of 2.3. A flow bench was used to

determine the flow coefficients and the swirl ratio for the two intake port variants. A paddle wheel and torque meter were used to independently quantify the in-cylinder charge motion for each intake port variant. These tools were used iteratively throughout the port design process to obtain the desired swirl ratio.

Test Conditions and Instrumentation

For this single-cylinder engine testing, intake and exhaust surge tanks were used upstream and downstream of the engine respectively to minimize the impact of pressure wave dynamics on engine performance. Air was supplied to the test cell at a pressure of ~8 bar, which was then manually regulated to ~4 bar. A 2nd pressure regulator and throttle valve allowed precise control of the air to the target pressure (boosted) or vacuum (part load). The air temperature was controlled using an 8 kW in-line air heater. A cooled EGR loop was available in the test cell, which draws exhaust gas from the exhaust surge tank and re-introduces it to the intake pipe upstream of the intake surge tank. The fresh air and recirculated exhaust gas are then mixed in the intake surge tank before being inducted into the combustion chamber. Figure 2 shows the layout of the engine in the test cell.

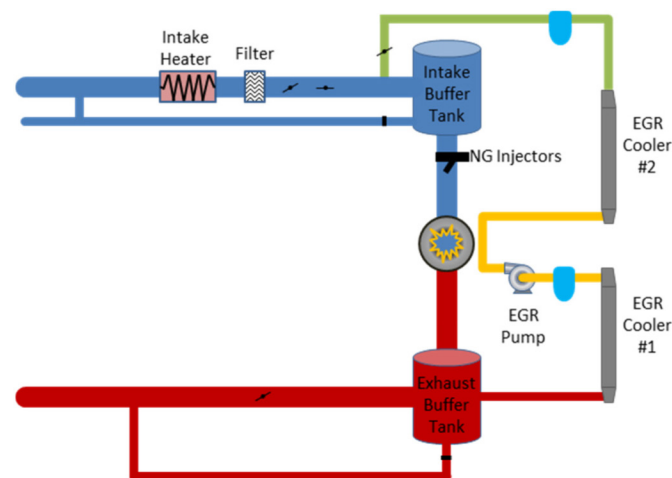


Figure 2. Test Cell Configuration

Filtered combustion air flows through a 3" Miriam laminar flow element (LFE), which measures the air flow rate and was installed 10 diameters after and 5 diameters before any piping bends. All engine-out gaseous emissions were sampled from the exhaust surge tank by an AVL CEB II emission bench, and smoke emissions were sampled downstream of the exhaust pressure valve and measured by an AVL 415S smoke meter. Temperatures and pressures were measured by an AVL PUMA 1.5 dynamometer operating system. The in-cylinder, intake, and exhaust pressure traces, as well as spark and injection signals were acquired with an AVL Indicomm high-speed data acquisition system.

The 95 methane number (MN) NG was provided at the test cell via utility pipeline. The supplied gas was then pumped and regulated to 550 kPa for use at the injector rail. An Endress-Hauser PROMASS 83A coriolis type mass flow meter was used to provide gas mass flow measurement data in kg/hr. The accuracy of the meter is $\pm 0.5\%$ of range (0 - 10 kg/hr).

A high-speed piezoelectric AVL GH14DK transducer was used in the cylinder to collect the indicating pressure data. AVL Indicom 2.3 software was used to supply the real-time raw and processed cylinder pressure curve data. Heat release analysis, including calculation of mass fraction burn rates, was performed using the first law of thermodynamics and a variable polytropic coefficient [3]. Peak combustion temperature was estimated using the ideal gas law based on the cylinder pressure vs. volume curve and the trapped mass. Heat losses to the combustion chamber surfaces were neglected.

The intake and exhaust manifolds were instrumented using Kulite high speed piezoresistive LP11DA transducers with an AE04 cooling adaptor used on the exhaust to maintain acceptable temperature conditions for low-pressure indicating.

The AE04 cooling adaptor was cooled with a water loop of nylon tubing fed by an electric pump mounted on a water reservoir.

For intake manifold pressure below ambient, a throttle valve was used upstream of the intake filter and heater, with closed loop control based on the manifold pressure set point. Boosted air flow was provided by an external boosting system consisting of two compressors located above the test cell. Pressure was controlled with a closed loop PID controller based on the intake manifold pressure set point. All air handling controllers were developed on the Motohawk ECU.

The charge air temperature at the manifold was heated to 40 °C at high loads using a single 8 kW electrical air heater installed in series with the intake system upstream of the plenum as shown in [Figure 2](#).

A Bosch LSU 4.2 lambda sensor was installed downstream of the exhaust orifice location. An ETAS ES635.1, which incorporates pressure compensation, was used for the lambda output signal analysis. An electronically controlled butterfly valve in the exhaust system was used to control the exhaust backpressure to be equal to the intake manifold pressure minus 10 kPa under boosted conditions.

Low temperature measurements were made with standard Omega RTDs and pressures were measured with standard low-pressure low speed transducers. Type K thermocouples were used in the exhaust system for high temperature measurements.

Test Methodology

This project focused on the comparison between stoichiometric combustion with cooled EGR and lean air-fuel combustion without EGR. Under lean combustion operation, the excess oxygen precludes the use of a TWC; therefore the engine-out NO_x levels must be reduced using a combination of lean air-fuel ratio and retarded spark timing. Because of the requirement of retarded spark timing to reduce NO_x emissions with lean burn combustion, a full spark sweep is necessary in order to determine the point that achieves the best net specific fuel consumption (NSFC) while meeting specified NO_x targets, subject to the combustion stability constraint. For consistency, spark sweeps were performed for both stoichiometric combustion with EGR and the lean air-fuel combustion tests. These were performed using both cylinder head configurations at 1200, 1800, and 2400 rpm at 5, 15, 20, 25, and 30 (1200 rpm only) bar loads.

Detonation borderline (DBL) was determined in a repeatable manner using visual inspection of the cylinder pressure trace to identify knocking. The operator advanced ignition timing until knock was visually observed, then the operator would advance ignition timing one additional step to ensure that the magnitude and frequency of the knocking on the trace increased. If so, then the previous step was determined to be the ignition timing to achieve DBL timing.

Because NMEP capability and fuel consumption were the main criteria of interest with the stoichiometric combustion with EGR strategy, there was no need to extend the spark sweep until the combustion phasing limit of 30° aTDC CA50 was attained (CA50 is the crank angle at which 50% of the mass fraction is burned). At low loads, spark sweeps were performed close to MBT timing, ranging from CA50 of 1 through 11 degrees aTDC. For high load the spark was retarded in 3 degree increments from DBL.

For the lean-burn combustion strategy, spark sweeps were performed starting at a CA50 of approximately 1° CA aTDC and continuing to the CA50 limit of 30° CA aTDC. For loads that resulted in knock limited operation, spark timing was set to DBL and then subsequently retarded in 6 degree increments until either the CA50 was equal to 30° CA aTDC or the coefficient of variation (COV) of NMEP was greater than 5%. Late combustion phasing leads to higher EGT and increased

COV of NMEP. Usually one of these two limits is reached prior to reaching a CA50 of 30 °CA aTDC. Otherwise, using this combustion phasing limit is an arbitrary way of limiting data collection and is frequently used in similar studies.

For all testing, the fuelling was set according to lambda as measured by the emissions bench, and measured by two additional independent methods to ensure consistency: a lambda sensor, and mass air/mass fuel calculation. The inlet temperature was set to 25°C for low loads and 40°C for boosted intake conditions. The start of injection (SOI) timing for the NG PFI injector was set to a fixed value of 300° CA bTDC compression.

The following parameters were set and controlled throughout the sweep:

- Inlet temperature = 25°C (low load) or 40°C (high load) +/- 2°C
- Coolant temperature = 90°C +/- 2.5°C
- Oil Temperature = 100°C +/- 2.5°C
- Lambda = 1.00 (stoich with EGR) or 1.00 to 1.80 (lean burn) +/- 0.01
- Exhaust Pressure = MAP minus 10 kPa +/- 0.5kPa

To correlate to realistic engine operation, each sweep was subject to the following constraints.

- Combustion phasing limit: 30°CA aTDC CA50
 - Action: Stop test
- Peak Pressure limit: 150 bar mean + 3 sigma peak pressure
 - Action: Retard combustion phasing to limit combustion pressure
- Exhaust gas temperature limit: 850°C exhaust temperature (corresponds to ~ 950°C turbine inlet temperature on multi-cylinder engine based on previous work.)
 - Action: Stop test
- Combustion Stability Limit: COV of NMEP = 5%
 - Action: Stop test

Engine load is represented by NMEP which is based on the 720°CA cylinder pressure curve and includes pumping losses. Fuel consumption and emissions are therefore presented on a net indicated specific basis, which is calculated as a function of NMEP. Since friction losses on a single cylinder engine are much greater on a relative basis than on a multi-cylinder engine, brake specific values are not comparable.

ENGINE TEST RESULTS

EGR Dilution

Representative results from testing at 1200 rpm and 20 bar NMEP at a CR of 12:1 are presented in this section. At this engine load, the spark timing was retarded from MBT due to knock. Therefore, the spark sweeps were started by setting timing at DBL and retarding in increments of 3° spark timing.

As shown in Figure 3, there is a significant reduction in NSFC with increasing levels of cooled EGR. This occurs for two reasons: 1) Cooled EGR reduces the rate of autoignition kinetics of the unburned end gas and thereby enables more advanced combustion phasing at DBL, as can be seen by the most advanced points of each sweep; and 2) Cooled EGR results in reduced combustion and burned gas temperatures due to dilution of the charge [6], which reduce heat transfer losses and result in reduced losses due to variation of specific heat and due to dissociation effects. The significantly reduced NOx emissions shown in Figure 3 also reflect the reduction in these temperatures.

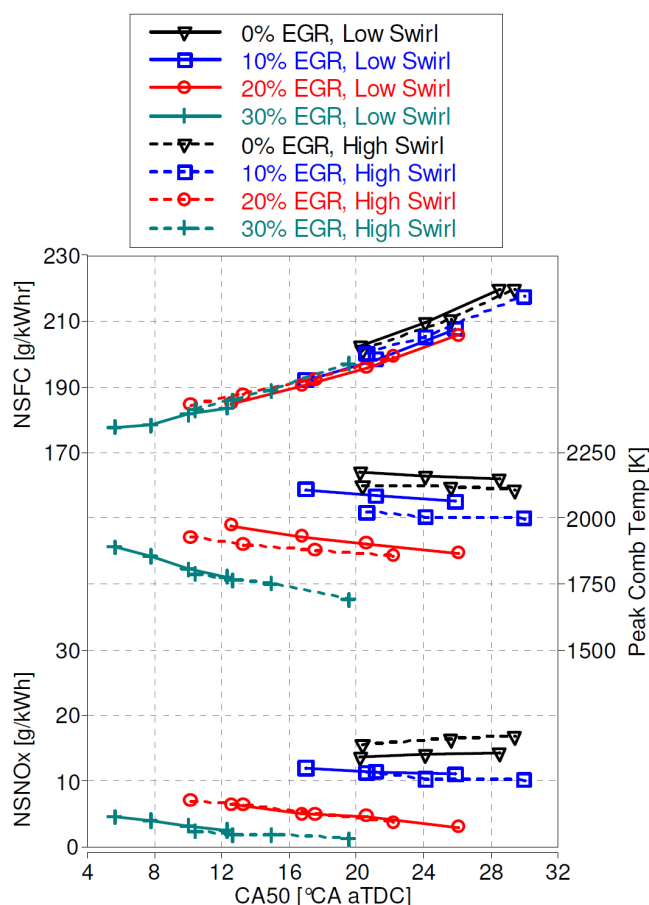


Figure 3. Effect of EGR and swirl ratio on fuel consumption, peak combustion temperature and NOx emissions at 1200 rpm, 20 bar NMEP and 12:1 CR