

Analysis of Cyclic Variability and the Effect of Dilute Combustion in a Gasoline Direct Injection Engine

Nicholas Matthias, Thomas Wallner, and Riccardo Scarcelli
Argonne National Lab.

ABSTRACT

The pressing need to improve U.S. energy independence and reduce climate forcing fossil fuel emissions continues to motivate the development of high-efficiency internal combustion engines. A recent trend has been to downsize and turbocharge automotive spark-ignited engines coupled with direct fuel injection to improve engine efficiency while maintaining vehicle performance. In-line with recent trends in state-of-the-art engine technology, the focus of this study is lean and EGR dilute combustion in a gasoline direct injection (GDI) engine.

The lean and dilute operating limits are defined by combustion stability typically in terms of COV_{IMEP} so experiments were carried out on an automotive size single-cylinder research engine to characterize combustion stability. From a 20,000 cycle sequence analysis, lean operating conditions exhibit binary high- to low-IMEP cycle sequences. This may be because the cycle-to-cycle feedback mechanisms are physically limited to one or two cycles. Longer sequences are observed with EGR likely because the EGR loop introduces a much longer transient feedback mechanism.

Results from a sensitivity study with induced variability show that COV_{IMEP} is far less sensitive to ignition perturbation than to injection perturbation while the sensitivity is similar in terms of efficiency. This is true for both lean and EGR dilute operation; however, one difference that stands out with EGR is the sensitivity of COV_{IMEP} to ignition perturbation. Increased sensitivity to ignition perturbation is likely the result of longer ignition delay which is critical for early flame kernel development.

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INTRODUCTION

With a share of approximately 60% of the total fuel energy used in the transportation sector in 2011, light-duty vehicles are the single largest consumer. Due to anticipated improvements in light-duty vehicle fuel economy as a result of CAFE and greenhouse gas emissions standards, this share is expected to drop to less than 50% by 2040. In order to comply with fuel economy regulations, new light-duty vehicle fuel economy numbers are expected to increase from 32.5 mpg in 2012 to 47.3 mpg in 2025 with a continuous increase up to 49.0 mpg in 2040 [1].

The market share of light-duty vehicles powered by a spark ignition internal combustion engine in 2010 was approximately 98%. While the EIA reference scenario (Figure 1) predicts an increased market share of FlexFuel, hybrid, plug-in hybrid, diesel as well as electric vehicles, more than 90% of light-duty vehicles sold in 2040 are predicted to be (partially) powered by a spark-ignition engine. That being the case, improvements in

gasoline internal combustion engine efficiency have immediate effect and are a critical building block to meeting future efficiency and emissions regulations.

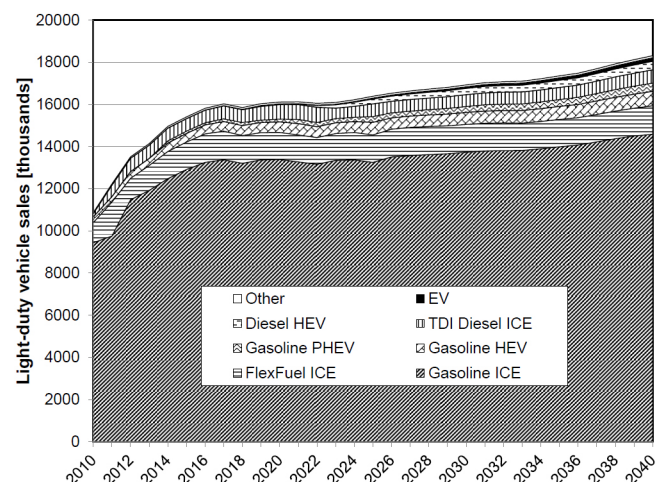


Figure 1. Sales of alternative fuels and powertrains in the light-duty vehicle sector [1]

Dilute combustion, in the form of lean or EGR diluted operation, is known to offer efficiency improvements which are in some cases also paired with emissions benefits. However, dilute combustion is also known to deteriorate combustion stability and combustion systems are currently limited by combustion stability issues. The most common measure of combustion stability is the coefficient of variation of the indicated mean effective pressure, COV_{IMEP} , defined as standard deviation divided by mean value of IMEP:

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{\mu_{IMEP}} \quad (1)$$

Typical target values of COV_{IMEP} stated in the literature depend on specific applications and are in the range of 2-10% with modern production engines typically at the lower range of this scale [2, 3].

Various research groups have analyzed cyclic variability in different ways. Samuel et al. looked at COV of in-cylinder pressure on a crank angle basis as well as comparing GDI and throttle body injection (TBI) in terms of COV_{IMEP} over a range of loads [4]. They found that pressure variations were minimal at the end of compression independent of engine type; however, COV_{IMEP} is higher for GDI than TBI [4]. Kaul et al. studied cyclic variability in a multi-cylinder GDI engine by analyzing cycle-by-cycle trends in heat release over a large number of cycles [5]. The statistical analysis aimed at next-cycle prediction and ultimately next-cycle control to improve overall stability [5]. The cycle sequence analysis performed in the current paper is modelled after the work of Kaul et al. Another study went beyond natural cyclic variability by actually inducing variability in engine control parameters. This work performed by Stevens et al. also included statistical analysis [6] and provided a basis for the perturbation experiments reported in the current paper.

This study focuses on the fundamental characteristics of combustion stability in a gasoline direct injection (GDI) engine and evaluates the connection between cyclic variability and engine efficiency. The study further analyses the effect of EGR diluted and lean operation on combustion stability and the subsequent impact on efficiency.

EXPERIMENTAL SETUP

Experiments were carried out on a single-cylinder research engine coupled to a dynamometer intended for steady-state operation. The engine design is representative of a modern GDI engine used for automotive applications; specifications are shown in Table 1. The engine has a 4-valve, 40° pent roof combustion chamber design with central spark plug and injector. It also has a pressure transducer port fitted with an AVL GU21C for high-speed in-cylinder data acquisition. Intake and exhaust ports are fitted with Kulite ETL-179B-190M-2BarA

and Kulite EWCT-312M-3.5BarA sensors respectively for high speed pressure data which are used as boundary conditions for 3D-CFD simulations [7]. Compressed air is supplied to the test cell and used as combustion air for boosted as well as throttled engine operation, although the 2000 RPM, 6 bar IMEP operating point examined in this study is throttled exclusively.

Table 1. Specifications of single-cylinder engine used for steady-state experimental results

Displacement	0.6264 L
Bore	89.04 mm
Stroke	100.6 mm
Compression Ratio	12.1 : 1
Intake Valve MOP	100 °CA ATDC
Exhaust Valve MOP	255 °CA ATDC
GDI Injector*	6 hole, solenoid
Injection Pressure	150 bar
Spark Plug	NGK-R dual fine wire, 0.7 mm gap
Fuel	EPA Tier II EEE

*Ford P/N CJ5Z-9F593-A

Between the intake and the exhaust is a cooled EGR loop. It is connected immediately downstream of the exhaust port and as such is considered high pressure EGR. A precisely positioned poppet valve (along with the respective intake and exhaust pressures) controls the EGR mass flow rate. Exhaust gases pass through a heat exchanger supplied with engine coolant at 85 °C which does the majority of cooling and then it passes through a section of stainless steel tubing in open air where some additional cooling takes place. The combination of these two effects maintains the EGR temperature below 85 °C as it enters the intake runner. The EGR rate is measured using high range and low range CO₂ analyzers sampling from the exhaust and intake respectively. The intake sample is drawn downstream of the EGR loop using a perforated sample probe to ensure that fresh air and EGR are well mixed. With these two measurements and ambient CO₂ which is measured daily in fresh intake air, the EGR rate is calculated based on Equation 2.

$$EGR \% = \frac{CO_{2intake} - CO_{2ambient}}{CO_{2exhaust} - CO_{2ambient}} * 100 \quad (2)$$

Note that external EGR is measured while internal EGR due to residual gas at exhaust valve closing is not taken into account when determining EGR rates.

Other emissions are measured along with CO₂ in the exhaust sample using a Pierburg AMA-2000 emissions bench. Nitrogen oxide (NO_x) concentration is measured using a CLD PM 2000 NO/NO_x analyzer, hydrocarbons using a FID PM-2000 calibrated for CH₃, O₂ using a BINOS-2000, and CO₂ using a BINOS-2000. The emissions bench is calibrated daily using zero and span gases of known concentration.

RESULTS

Overall Efficiency and Stability Trends

A basic assessment of general trends of dilute operation on efficiency, emissions and combustion characteristics is summarized in Figure 2. Ignition delay is the time window in terms of crank angle between the ignition event and when 10% of the total heat release has occurred (10% MFB). The time window between 10% and 90% heat release (10 to 90% MFB) is taken to be combustion duration. At the 2000 RPM, 6 bar IMEP operating point, dilution was accomplished through EGR up to 20% and lean operation up to a relative air/fuel ratio (λ) of 1.5. Data shown represents the arithmetic average of three data sets collected over a 30 second time interval with high-speed data sampled over 500 cycles. All operating points were run at maximum brake torque (MBT) spark timing.

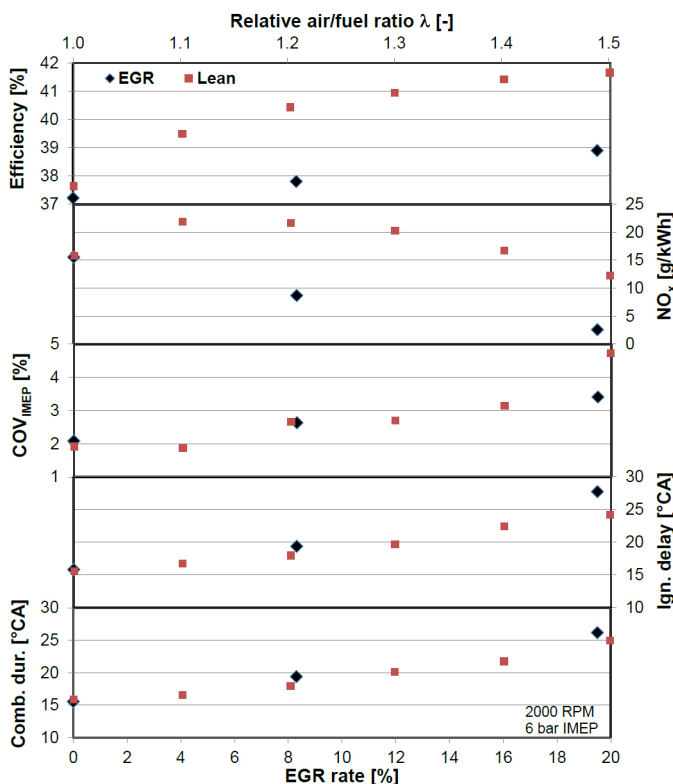


Figure 2. Trends of efficiency, NO_x emissions, combustion stability, ignition delay, and combustion duration with increasing dilution at 2000 RPM, 6 bar IMEP

Starting from a baseline efficiency of approximately 37% at stoichiometric operation without EGR, efficiency increases to approximately 39% at 20% EGR while reducing NO_x emissions from over 15 g/kWh to below 3 g/kWh. However, the combustion stability decreases from a COV_{IMEP} around 2% at the baseline point to 3.5% at 20% EGR. The EGR addition also increases ignition delay as well as combustion duration almost by a factor of 2.

In comparison, lean operation results in a more significant increase in indicated efficiency with a peak at 41.7% at $\lambda = 1.5$.

Over the air/fuel ratio range an increase in NO_x emissions compared to stoichiometric operation is observed and only at the $\lambda = 1.5$ point NO_x emissions dropped slightly below the baseline. Lean operation also continuously degrades combustion stability with a COV_{IMEP} almost at 5 at $\lambda = 1.5$. In terms of combustion stability operation at 20% EGR is similar to lean operation at $\lambda = 1.4$. Both ignition delay and combustion duration increase moderately with lean operation.

As previously mentioned, baseline data was collected at MBT spark timing. Combustion phasing also significantly influences combustion stability as shown in Figure 3 for stoichiometric operation at 0% EGR. Peak efficiency is achieved at a combustion phasing with a 50% mass fraction burned (MFB) location around 7.5 °CA ATDC at a COV_{IMEP} around 2. Advancing or retarding combustion phasing from the efficiency-optimized MBT timing not only negatively affects efficiency but also results in decreased combustion stability. Data was generated by advancing and retarding the spark timing in 2 °CA steps as indicated in the plot. Retarding spark timing by 6 °CA results in an approximately 7 °CA later combustion phasing and an increase in COV_{IMEP} from 2% to 2.75% with an associated 1% efficiency loss. Similar trends can be observed when advancing combustion phasing. Achieving MBT combustion phasing is therefore not only desirable from an efficiency point of view but also ensures optimal combustion stability.

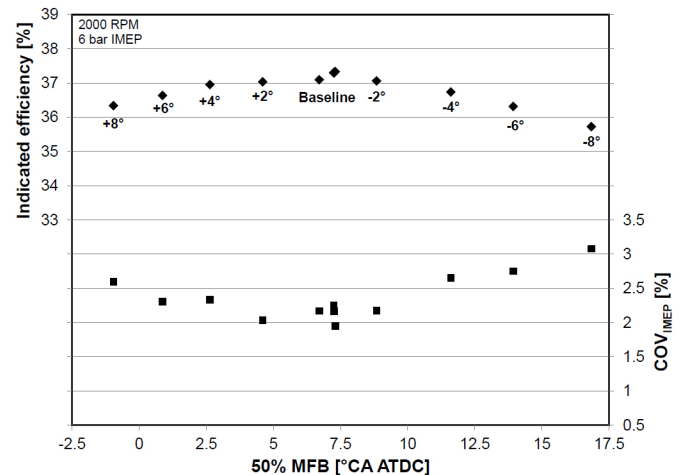


Figure 3. Influence of combustion phasing on indicated efficiency, and combustion stability at 2000 RPM, 6 bar IMEP

Figure 4 shows the correlation between λ and engine efficiency as well as combustion stability in a close range around stoichiometric operation. Data was generated by reducing and increasing fuel injection duration in 2% steps as indicated in the plot. As shown earlier, lean operation causes increased efficiency with a simultaneous deterioration in combustion stability. However, fuel rich operation causes the anticipated decrease in efficiency, while offering significant improvements in combustion stability. With a 6% increase in injection duration causing operation at $\lambda = 0.89$ a COV_{IMEP} around 1.2% can be sustained but with an efficiency loss of almost 4%. On the