

Isolating the Effects of EGR on HCCI Heat-Release Rates and NO_x Emissions

John E. Dec, Magnus Sjöberg and Wontae Hwang*

Sandia National Laboratories

This is a work of the U.S. Government and is not subject to copyright protection in the United States.

ABSTRACT

High-load HCCI operation is typically limited by rapid pressure-rise rates (PRR) and engine knock caused by an overly rapid heat-release rate (HRR). Exhaust gas recirculation (EGR) is commonly used in HCCI engines, and it is often stated in the literature that charge dilution with EGR (or high levels of retained residuals) is beneficial for reducing the PRR to allow higher loads without knock. However, EGR/retained-residuals affect other operating parameters such as combustion phasing, which can in turn influence the PRR independently from any effect of the EGR gases themselves. Because of the multiple effects of EGR, its direct benefit for reducing the PRR is not well understood.

In this work, the effects of EGR on the PRR were isolated by controlling the combustion phasing independently from the EGR addition by adjusting the intake temperature. The experiments were conducted using gasoline as the fuel at a 1200 rpm operating condition. EGR levels were increased from zero up to the amount required to replace all excess air with combustion products (*i.e.* stoichiometric intake) at a relatively high-load for HCCI (corresponding to $\phi = 0.42$ for no EGR). As the amount of EGR was increased, a systematic increase in fueling rate was required to maintain a constant IMEP_g. This occurred because the thermal efficiency decreased with EGR addition due to the lower specific-heat ratio (γ) of the EGR gases. Despite the increased fueling, a small reduction in maximum PRR was observed. This allowed the IMEP_g to be increased from 450 to 475 kPa for the same maximum PRR, albeit with a ~2% loss in thermal efficiency. The effectiveness of EGR for extending the high-load limit was also examined. Additionally, contrary to intuition, peak combustion temperatures actually increased with EGR because of the increased fueling required to maintain IMEP_g. This resulted in a significant increase in NO_x emissions, although NO_x was still very low compared to spark-ignition or diesel engines.

INTRODUCTION

Although HCCI engines offer substantial benefits in terms of providing high efficiency with low NO_x and particulate emissions, control of autoignition timing and extending the high-load limit can be challenging. One of the most widely used methods for controlling autoignition timing over the load/speed map is the use of exhaust gas recirculation (EGR). Adding cooled EGR to intake gases can suppress autoignition, retarding the combustion phasing [1,2]. Alternatively, hot EGR, usually in the form of retained hot residuals (internal EGR) is often used to advance combustion phasing, or to initiate autoignition of high-octane fuels (*e.g.* gasoline) in engines with the relatively low compression ratios that are often used to allow switching to spark-ignition at high loads [3-5]. In addition, combinations of cooled and internal EGR can provide even greater control capability [6,7].

The use of EGR has also been associated with extending the high-load limit of HCCI engines [9-11]. High-load HCCI is typically limited because the pressure-rise rate (PRR) with combustion becomes very rapid, causing engine knock. In these previous works, high levels of cooled EGR were generally found to reduce the knocking propensity, allowing higher loads, compared to dilution only with air at the same fueling rate. However, the reason for the benefit was unclear. These works suggested that EGR should reduce the combustion heat-release rate (HRR) due to lower combustion temperatures and the reduced oxygen concentration [8-10], but one article also noted that EGR retards the combustion phasing and that this could be one of the main reasons for the observed reduction in the HRR and associated PRR [8]. Even without the use of EGR, retarding the combustion phasing significantly reduces the PRR [12]. In another work, combustion phasing was controlled independently of EGR addition, and EGR was found to have almost no

* Currently at General Electric Global Research Center.

effect on the 10 – 90% combustion duration [11]. Finally, it should be noted that chemical-kinetic computations have shown that EGR addition slows HCCI combustion chemistry [13]. This would increase the burn duration for perfectly homogeneous HCCI, but EGR's effect in a real engine is less clear, since the burn duration is often dominated by the amount of thermal stratification [14,15].

Other important considerations for the use of EGR in HCCI engines are its effects on thermal efficiency, and emissions of hydrocarbons (HC), carbon monoxide (CO), and nitrogen oxides (NO_x). Changes in thermal efficiency with EGR can result from changes in both combustion efficiency and thermodynamic efficiency, with changes in the thermodynamic efficiency resulting mainly from changes in combustion phasing, heat transfer, and γ (the ratio of specific heats, c_p/c_v). Previous studies have shown mixed results on the effect of EGR on the thermal efficiency [8,11], probably because the various contributing factors have different trends for different operating conditions. Combustion efficiency generally increases with high levels of EGR since the HC and CO in the recirculated exhaust gas get a second chance to burn (or alternatively, the total amount of gases passing through the engine is reduced) [8,11]. However, γ decreases due to the shift from diatomic to triatomic molecules as EGR dilution is substituted for air dilution. This has significant potential for reducing the thermodynamic efficiency [11]. Changes in heat transfer with EGR can also be significant, and they can occur in two ways. First, if post-combustion temperatures are reduced by EGR (as might be expected due to the higher heat capacity of the gases) heat transfer losses would be reduced, as suggested in Ref. [11]. Second, if the PRR is so rapid that the engine is knocking, heat-transfer rates can be quite high due to the flows associated with the acoustic resonance of the knock. If EGR addition reduces the PRR sufficiently to stop the knocking (either by retarding the timing or by directly reducing the HRR), heat transfer losses will be greatly reduced, increasing the thermodynamic efficiency.¹ Finally, with the exception of greatly reducing the heat transfer as just discussed, if EGR or other means are used to retard the combustion phasing, the effective expansion ratio will be reduced, decreasing the thermodynamic efficiency.

Emissions of HC have been found to decrease with EGR addition, in agreement with the observed improvement in combustion efficiency [8,11]. Similarly, CO emissions were generally found to decrease [8,11]. However, under some conditions, where combustion phasing became more retarded with EGR addition [8], they remained nearly constant. NO_x emissions were generally found to decrease with EGR addition [8,11], but in some cases NO_x remained nearly constant or showed inconsistent trends that included some increases.

These inconsistent findings on effects of EGR on NO_x, thermal efficiency, and PRR likely result because EGR

acts on the combustion and thermodynamic cycle in multiple ways, as indicated in the preceding paragraphs. In practical applications, these multiple effects of EGR can become convolved, making it difficult to determine specific cause-and-effect relationships.

The objective of the current work is to systematically investigate the effects of EGR on the combustion HRR, thermal efficiency, and NO_x emissions in an HCCI engine. Two key points are to determine whether EGR is beneficial for 1) extending the high-load limit by lowering the PRR to reduce knock² and 2) maintaining low NO_x at high loads, independent of its effects on combustion phasing. As discussed above, EGR can retard the combustion phasing [1], which is beneficial for extending the high-load limit and for maintaining low peak-combustion temperatures and the associated NO_x formation [12,16]. However, these benefits occur regardless of whether the combustion retard is produced by EGR or other means. In the current work, the direct effects of EGR on the PRR and NO_x are isolated by controlling the combustion phasing independently from the EGR addition. This is accomplished by systematically increasing the intake temperature to maintain constant phasing ($\pm 0.15^\circ$ CA) of the 50% burn point (CA50) as EGR levels were swept over a wide range. It should be noted that changing the intake temperature to control CA50 independently of EGR addition is analogous to adjusting the relative amounts of hot retained residuals, cooled EGR, and excess air in practical HCCI engines that use variable-valve timing control.

Another secondary effect of EGR is that it can affect the engine power output, measured here by the gross indicated mean effective pressure (IMEP_g).³ Additionally, the changes to the initial charge temperature required to maintain CA50 can also affect the IMEP_g, mainly through changes in the intake density. To isolate the direct effects of EGR on PRR and NO_x from these effects on IMEP_g, data were acquired for EGR sweeps using three different fueling strategies: constant charge/fuel mass ratio, constant fueling rate, and constant IMEP_g. For each fueling strategy, EGR levels were swept from zero (excess air dilution only) to near-stoichiometric conditions where nearly all the excess air was replaced by recirculated combustion products. A research-grade gasoline was used for all tests.

The experimental HCCI research engine facility used for this study is described in the next section, along with a description of the operating procedure and data acquisition techniques. This is followed by a discussion of the selection of the base operating condition and the notation used for EGR addition. Next is the results and discussion section, with the main data presentation being preceded by a short review of the effects of EGR on combustion phasing and the effect of combustion phasing on PRR. In the final section, the investigation is summarized and conclusions are drawn.

¹ Retarding combustion timing without EGR addition, such as by reducing the intake temperature, can also reduce the PRR and eliminate knocking, with a corresponding increase in thermal efficiency.

² A reduction in PRR could also be used to improve stability by allowing more advanced combustion phasing.

³ IMEP_g includes the compression and expansion strokes only; pumping work is not included.

EXPERIMENTAL SETUP

ENGINE FACILITY

The HCCI research engine used for this study was derived from a Cummins B-series six-cylinder diesel engine, which is a typical medium-duty diesel engine with a displacement of 0.98 liters/cylinder. Figure 1 shows a schematic of the engine, which has been converted for single-cylinder operation by deactivating cylinders 1–5. The engine specifications and operating conditions are listed in Table 1. The configuration of the engine and facility is nearly the same as for previous studies (e.g. Refs. [1,2,12,17-19]).

The active HCCI cylinder is fitted with a compression-ratio (CR) = 14 custom piston as shown in Fig. 1a. Compared to the broad shallow bowl used in most previous studies, e.g. [1,17], this piston features a smaller-diameter hemispherical bowl in the center and a broad flat squish region. This change in bowl geometry was made to facilitate safer operation for concurrent testing of a variable valve actuation system (not reported here). Comparison tests showed minimal differences in performance and emissions compared to the CR = 14 shallow-bowl piston (see Ref. [1] for a schematic). A more complete discussion of the comparison between pistons may be found in Ref. [18]. Both of these custom-designed CR = 14 pistons provide a small topland-ring crevice, amounting to only 2.1% of the top-dead-center (TDC) volume, including the volume behind the top piston ring.

TABLE 1. Engine Specifications and Operating Conditions

Displacement (single-cylinder)	0.981 liters
Bore	102 mm
Stroke	120 mm
Connecting Rod Length	192 mm
Geometric Compression Ratio (current study)	14:1
No. of Valves	4
IVO	0° CA*
IVC	202° CA*
EVO	482° CA*
EVC	8° CA*
Swirl Ratio	0.9
Fueling system	Fully Premixed
Engine Speed	1200 rpm
Intake Pressure	100 kPa (abs.)
Intake Temperature	143° - 192°C
Coolant Temperature	100°C
Fuel	Research-grade Gasoline

* 0° CA is taken to be TDC intake. The valve-event timings correspond to 0.1 mm lift.

As shown in Fig. 1b and discussed in Refs. [1,19], the engine facility is equipped with two EGR loops and the capability of supplying simulated EGR (combustion products) by adding N₂, CO₂ and H₂O. For the current study, all EGR was supplied using the cooled EGR loop. The EGR cooler is necessary for maintaining the desired in-

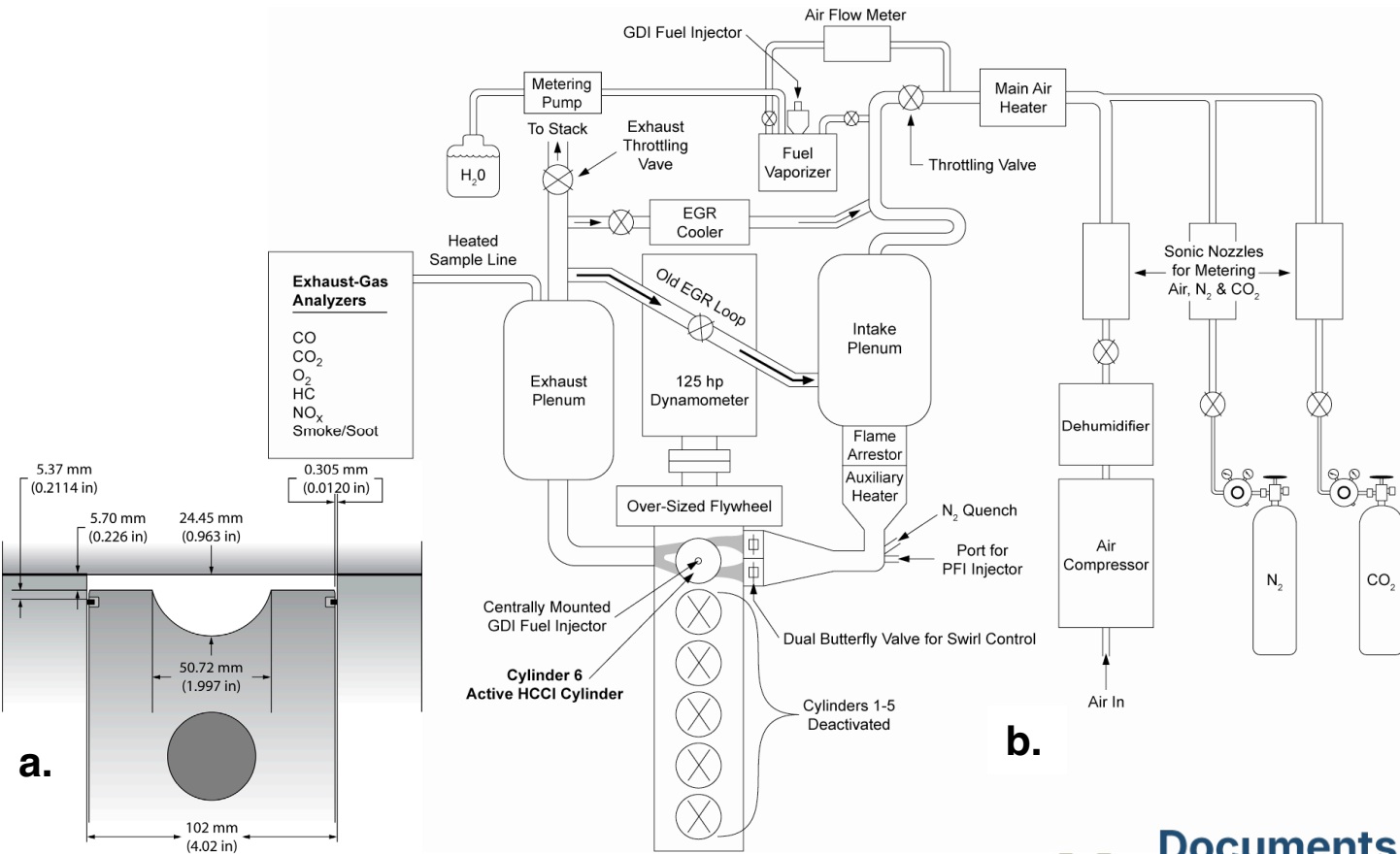


Figure 1. Schematic of the HCCI engine facility, and combustion chamber geometry of the CR = 14 piston, at TDC

take temperature as EGR levels are increased. In addition, this loop introduces the EGR well upstream of the intake plenum, so the EGR travels with the intake air through a series of bends before reaching the intake plenum, to insure that the intake charge is well mixed.

For this study, the intake charge was fully premixed using the fueling system shown at the top of the schematic in Fig. 1b. This fueling system consists of an electrically heated fuel vaporizer and appropriate plumbing to ensure thorough premixing with the air and EGR upstream of the intake plenum. A positive displacement fuel flow meter was used to determine the amount of fuel supplied, and the supplied air was precisely metered by a sonic nozzle (Fig. 1b). Supplied fuel/air mixtures were determined from these two measurements. The fuel was a research-grade gasoline supplied by Chevron Phillips Chemical Co., with RON = 90.8, MON = 83.4, and antiknock index $(R + M)/2 = 87.1$.

Prior to starting the experiments, the engine was fully preheated to 100°C by means of electrical heaters on the “cooling” water and lubricating-oil circulation systems. In addition, the intake tank and plumbing were preheated to a minimum of 60°C to avoid condensation of the fuel or water from the EGR gases. An auxiliary heater mounted close to the engine provided precise control of the intake temperature to maintain the desired combustion phasing. Intake temperatures ranged from 143° - 192°C depending on EGR levels. All data were taken at an engine speed of 1200 rpm with an intake pressure of 100 kPa, simulating naturally aspirated conditions.

DATA ACQUISITION

Cylinder pressure measurements were made with a transducer (AVL QC33C) mounted in the cylinder head approximately 42 mm off center. The pressure transducer signals were digitized and recorded at 1/4° CA increments for one hundred consecutive cycles. The cylinder-pressure transducer was pegged to the intake pressure near bottom dead center (BDC) where the cylinder pressure reading was virtually constant for several degrees. Intake temperatures were monitored using thermocouples mounted in the two intake runners close to the cylinder head. For all data presented, 0° crank angle (CA) is defined as TDC intake (so TDC compression is at 360°). This eliminates the need to use negative crank angles or combined bTDC, aTDC notation.

For all experiments, the intake temperature was adjusted to maintain CA50 at the desired crank angle (370° CA for most experiments). CA50 was determined from the cumulative apparent heat-release rate (AHRR), computed from the cylinder-pressure data (after applying a 2.5 kHz low-pass filter [17]). Computations were performed for each individual cycle, disregarding heat-transfer and assuming a constant ratio of specific heats [16]. The average of 100 consecutive individual-cycle CA50 values was then set to 370° CA. The reported PRRs are computed from the same low-pass filtered pressure data. For each cycle, the maximum PRR was analyzed separately with a

linear fit over a moving $\pm 0.5^\circ$ CA window. Similar to CA50, these individual-cycle values were then averaged over the 100-cycle data set.

A second method of computing the HRR was used for the plots presented in Fig. 6. Here, the heat release was computed in a more refined way from the ensemble-averaged pressure trace, using the Woschni correlation for heat transfer [16]. Using the ensemble-averaged pressure trace has benefits from the standpoint of reduced noise on the heat-release traces. On the other hand, it can lead to overestimated burn durations if the cycle-to-cycle variations are large. However, for the condition where this was applied, the phasing was fairly stable with the standard deviation of CA50 over 100 fired cycles being only about 0.8° CA.

Charge temperatures during the closed part of the cycle (*i.e.* compression/expansion) are computed using the ideal-gas law in combination with the measured pressure (ensemble-averaged over 100 cycles), the known cylinder volume, and the trapped mass. The average molecular weight used for the calculation during the compression stroke corresponds to that of the trapped gases - including fresh air, EGR, retained residuals, and fuel. During the combustion event, and in proportion to the mass fraction burned, the molecular weight in the calculations is gradually changed to that of the measured exhaust composition.⁴ This exhaust molecular weight is then used for the remainder of the expansion stroke.

Exhaust emissions data were also acquired, with the sample being drawn from the exhaust plenum using a heated sample line (see Fig. 1b). CO, CO₂, HC, NO_x, and O₂ levels were measured using standard exhaust-gas analysis equipment. In addition, a second CO₂ meter monitored the intake gases just prior to induction into the engine. This allowed the EGR fraction of the intake gases to be computed from the ratio of the intake and exhaust CO₂ concentrations. The amount of EGR was also computed directly from the difference between the measured intake-air flow for operation without and with EGR. Agreement was good between the two methods for determining the amount of EGR.

BASE CONDITION AND EXPERIMENTAL PROCEDURE

As outlined in the introduction, the primary need to better understand the effects of EGR on HCCI-engine performance and emissions involves its use for high-load operation. Therefore, the base condition for this study was selected to be a relatively high load for HCCI operation, corresponding to a fueling rate giving $\phi = 0.42$ for zero EGR (air dilution only). Another important consideration is that the combustion be stable from cycle to cycle and that it not be knocking. The latter is very important since changes in knock intensity can have a relatively large effects on the heat transfer and therefore the thermal effi-

⁴ Dissociation of the combustion products is neglected since it is expected to be small due to the relatively low peak combustion temperatures.