

Energy Distribution Analysis in Boosted HCCI-like / LTGC Engines - Understanding the Trade-Offs to Maximize the Thermal Efficiency

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ABSTRACT

A detailed understanding of the various factors affecting the trends in gross-indicated thermal efficiency with changes in key operating parameters has been carried out, applied to a one-liter displacement single-cylinder boosted Low-Temperature Gasoline Combustion (LTGC) engine. This work systematically investigates how the supplied fuel energy splits into the following four energy pathways: gross-indicated thermal efficiency, combustion inefficiency, heat transfer and exhaust losses, and how this split changes with operating conditions. Additional analysis is performed to determine the influence of variations in the ratio of specific heat capacities (γ) and the effective expansion ratio, related to the combustion-phasing retard (CA50), on the energy split. Heat transfer and exhaust losses are computed using multiple standard cycle analysis techniques. The various methods are evaluated in order to validate the trends.

This work focuses on explaining the trends in thermal efficiency and the various energy-loss terms for independent sweeps of fueling rate, intake temperature and engine speed. Trends in thermal efficiency can be well-explained by considering variations in combustion efficiency, CA50 retard, γ and heat transfer. By identifying the energy losses, these results provide a new understanding that can help to optimize the thermal efficiency across the load/speed range in LTGC engines. Of particular importance, a picture is provided of how the heat transfer varies with changes in engine operating conditions. For example, results indicate that CA50 and the magnitude of the acoustic oscillations (*i.e.* knock) are fundamental parameters affecting the heat transfer.

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INTRODUCTION

Current engine research is largely oriented toward improving the overall thermal efficiency while meeting stringent emissions regulations and maintaining acceptable combustion noise levels. Toward this goal, engines utilizing Low-Temperature Gasoline Combustion (LTGC), including Homogeneous Charge Compression Ignition (HCCI) and partially stratified variations of HCCI, are attractive because of their superior fuel efficiency, particularly compared to conventional Spark Ignition (SI) engines but also to the latest generations of emissions-compliant Diesel engines [1, 2].

Because the charge in an LTGC engine is sufficiently well-mixed and globally lean or dilute (typically $\phi_m < 0.5^1$), combustion temperatures are low, and NO_x and soot formation zones are essentially avoided [2, 3]. This eliminates the need for the costly aftertreatment systems for soot and NO_x . Expensive high-pressure and fast-response fuel-injection technology is also not required for in-cylinder pollutant management. Thus, in LTGC engines, thermal efficiency is not

compromised to minimize NO_x and particulate emissions. Without this constraint, a higher potential exists to finely optimize the control parameters to always provide the highest efficiency.

Peak gross-indicated thermal efficiencies (η_{th})² as high as 48.3% have been previously reported for the mid-size (one-liter displacement) boosted-LTGC engine used in this study with a 14:1 Compression Ratio (CR) [4]. The results of the present study report a peak- η_{th} of 49.4% for this same engine using a CR = 16:1. η_{th} above 50% have also been achieved using a larger engine size with a higher CR [5]. It is of interest to further extend the η_{th} for a given engine geometry in order to demonstrate the full potential of boosted-LTGC engines. Perhaps more important is to maintain high efficiencies across the load/speed range of the engine. To achieve these goals, it is necessary to understand in detail the factors affecting the η_{th} and how their contributions vary with changes in the engine operating parameters. There are three main factors that affect the η_{th} [2, 4, 6]: 1) the combustion efficiency (η_{comb}), 2) the thermodynamic efficiency and 3)

1. ϕ_m : charge-mass equivalence ratio as introduced with Equation 2.

2. Since this study focuses on combustion-related phenomena in a single-cylinder research engine, gross-indicated thermal efficiencies, denoted as η_{th} , are used rather than brake or net thermal efficiencies, as explained in greater detail in the Experimental Setup section. η_{th} will be defined with Equation 4.

the wall Heat Transfer (HT)³. These factors are affected by several sub-effects that tradeoff differently depending on the operating parameter that is varied.

In LTGC engines, η_{comb} is an important factor affecting the η_{th} for low loads, or more correctly low- ϕ_m [2, 4]. A peak-charge temperature above 1500 K is required to prevent incomplete bulk-gas CO-to-CO₂ reactions [2, 7, 8, 9, 10]. Below this temperature, the η_{comb} drops off quickly and the η_{th} is severely deteriorated [8]. Above this temperature, the η_{comb} is generally relatively high (above ~96%), and the main sources of HC and CO are incomplete combustion of gases in or out-gazing from the top-land crevice and mixture near the walls [11, 12]. For higher load conditions, higher thermal environment still favors oxidation and higher η_{comb} [13]. However, the increasing rate of η_{comb} is much lower and for these relatively high- η_{comb} conditions, η_{comb} was found to not be any longer a decisive factor influencing the variations in η_{th} (change in η_{comb} too small) [4]. Also, the need to use Exhaust Gas Recirculation (EGR)⁴ helps improve η_{comb} by providing a second chance for the HC and CO in the EGR to burn [13].

As given by the theoretical equation for isentropic compression/expansion (Equation 1), the thermodynamic efficiency depends on two parameters: CR and γ , the ratio of specific heats (c_p/c_v). The higher the CR or γ , the higher the thermodynamic efficiency, leading to more work extracted and less energy lost into the exhaust gases.

$$\eta_{\text{thermodynamic}} = 1 - \frac{1}{\text{CR}^\gamma - 1} \quad (1)$$

For a given engine geometry, CA50 affects the effective expansion ratio and thus the thermodynamic efficiency: retarding the CA50 after Top Dead Center (TDC) lowers the expansion efficiency [2, 4, 6]. Alternatively, to maximize the thermodynamic efficiency, it is desirable to advance as much as possible toward TDC, while considering the CA50 limitation imposed by the need to avoid excessive combustion noise/knock [2, 14]. However, as shown by Dec et al. [4], for relatively high η_{comb} conditions, there is little potential benefit to advancing CA50 beyond about 366-368°CA, even without knock, since the piston motion per Crank Angle (CA) degree is small near TDC, and HT losses increases, as discussed below.

γ , is sensitive to the charge composition and temperature: γ decreases with higher cycle temperatures, and with higher EGR and ϕ_m (more tri-atomic molecules). Thus, leaner mixtures with lower temperatures and less EGR tend to improve the thermodynamic efficiency through a more advantageous γ value.

Concerning the HT, it is caused mainly by convection from the hot in-cylinder gases to the combustion chamber walls. HT is typically lower for LTGC engines than for conventional engines because of the lower combustion temperatures, a more quiescent flow field, reducing the convective HT-coefficient and the absence of radiation from soot⁵. However, HT still constitutes a significant part of the lost energy. Modeling works have shown that HT is sensitive to various main engine parameters [6, 15, 16]. In particular, HT was found to increase with CA50 being more advanced. Hence, this effect acts to mitigate the gain in η_{th} produced by the better expansion ratio, particularly as CA50 is closely approaching TDC, where the gain in expansion ratio is small. If CA50 is not sufficiently retarded to slow down the reaction rate, the combustion process can generate relatively strong acoustic oscillations (*i.e.* knock) [17]. If knocking conditions are encountered, the acoustic oscillations further enhance the HT and can lead to significant decrease in the η_{th} [4, 18, 19]. The knock-limit often dictates the high-load boundary of the engine and also restrains the η_{th} . As pointed out by Saxena and Bedoya [6], a better quantitative understanding of how ringing/knock impacts the HT is required. More generally, for practical operations of LTGC engines, there is a lack of understanding regarding variations of HT with operating parameters and how these variations affect the η_{th} , in contrast to the η_{comb} and the thermodynamic efficiency which are fairly well understood.

This summary of the factors influencing the η_{th} in LTGC engines highlights the fact that the multiple tradeoffs can take place. In this context, the primary objective of the current study is to provide a detailed understanding of the trend in η_{th} with changes in key operating parameters by systematically analyzing variations in the main energy-loss terms. In this study, the parameters that have been selected are: CA50, ϕ_m , intake temperature (T_{in}) and engine speed. The effect of CA50 includes a characterization of the effect of knock on the HT (and η_{th}) and an evaluation of two commonly employed knock-detection metrics (the knock index [20] and the ringing intensity [21]). Then, ϕ_m is swept with CA50 adjusted to prevent knock. The analysis of the effects of these two fundamental parameters (CA50 and ϕ_m) will help to explain the influence of T_{in} and engine speed. Before presenting the results, the experimental setup is introduced in the next section. Following this, in the “Data Analysis” section, the methodology developed to compute the amount of energy lost through HT and into the exhaust is explained in detail.

EXPERIMENTAL SETUP & DATA ACQUISITION

Engine Facility

The LTGC research engine used for this study was derived from a Cummins B-series six-cylinder diesel engine with a displacement of 0.98 liters/cylinder. Figure 1a shows a schematic of the engine, which has been converted for single-cylinder operation. The engine specifications and main operating conditions are listed in Table 1. Data for the present study have been obtained using two different

5. In LTGC engines, the ability to phase most of the combustion sequence closely after TDC, due to the fast reaction rate, can also contribute to overall efficiency by combustion igniting “well” before TDC, so that no negative work is avoided, reducing the time for HT at higher in-cylinder temperatures.

3. In practice, the blow-by losses also affect the η_{th} through losses of pressure and fuel mass. In the analysis conducted in this study, its contribution was not considered. Blow-by was measured in our engine for a range of conditions and was found to be minimal. For instance, a change in boost pressure from 1 to 2 bar increases the blow-by from 0.28 to 0.30% of the total mass. Blow-by losses decrease by 0.41% for an increase in engine speed from 600 to 1800 RPM. Since the change in measured η_{th} will be about half of the variation in the loss of mass, this effect in the noise of the η_{th} measurements and can reasonably be neglected.

4. For boosted-LTGC engines, cooled-EGR is used as a control strategy to delay the CA50 in order to prevent knock with increased fueling [13], when it is not practical to further decrease T_{in} .

pistons, giving geometric CRs of 14:1 and 16:1, as shown in Figure 1b. Both pistons provide a small top ring-land crevice, amounting to less than 0.9% of the TDC volume.

As shown in Figure 1a, the EGR loop is equipped with a cooler, a gas-to-water heat exchanger and a gas/liquid separator (not shown) to remove the condensed water. The flow rate of cooling water can be adjusted to control the temperature of the EGR gases down to temperatures as low as 30°C. The EGR is introduced well upstream of the intake plenum, to insure that the intake charge is well mixed. With this configuration, the exhaust pressure must be greater than the intake pressure (P_{in}) for EGR to flow into the intake. The required back pressure was achieved by throttling the exhaust flow using the valve shown in Figure 1a. This typically resulted in the exhaust pressure being about 2 kPa greater than P_{in} ⁶. Intake air was supplied by an air compressor and precisely metered by a sonic nozzle as shown in Figure 1a. The EGR fraction was varied by adjusting the amount of supplied air, and then adjusting the exhaust throttle to maintain the desired P_{in} . In this article, all pressures are absolute.

Table 1. Engine specifications and operating conditions.

Displaced (single cylinder)	0.981 liters
Bore	102 mm
Stroke	120 mm
Connecting rod length	192 mm
Geometric compression ratio	14:1 & 16:1
Approximate surface to volume ratio at TDC [1/cm]	2.65 (CR = 14:1) 2.96 (CR = 16:1)
Number of valves	4
I/O	0°C/A *
IVC	202°C/A *
EVO	482°C/A *
EVC	8°C/A *
Swirl ratio	0.9
Fueling system	Fully PM and GDI
GDI injector	Bosch, 8-hole
Included angle	70°
Hole size	Stepped-hole, min. dia. = 0.125 mm
Injection pressure	120 bar
Engine speed	1000 to 2000 RPM
Centerline T_{in} †	30 to 70°C
P_{in} (abs.)	2.0 and 2.4 bar
Coolant and oil temperature	~ 100°C

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

† T_{in} is measured in the center part of the duct in the two runners about 120 mm from the intake valves.

6. This exhaust pressure is much lower than would be required to drive an actual turbocharger. However, the current work focuses on changes in IMEP_g which is hardly affected by using a higher, more-realistic exhaust pressure, as was shown in a previous study [13]. In this earlier work, a realistic back pressure, considering a 40% combined turbo and compressor efficiency for a P_{in} = 2.0 bar condition, reduced the IMEP_g by < 0.2% (or about a 0.1%-unit decrease in η_{in}), which is in the noise of the data. Net IMEP was reduced by about 4%. Thus, not considering the higher exhaust pressure is not expected to significantly affect the ensuing results.

In this study, both PreMixed (PM) and Direct Injection (DI) fueling were used. The PM fueling system, shown at the top of the schematic in Figure 1a consists of a GDI (Gasoline-DI) injector mounted in an electrically heated fuel-vaporizing chamber and appropriate plumbing to ensure thorough premixing with the air and EGR upstream of the intake plenum. The DI fueling is accomplished using a second GDI injector mounted centrally in the cylinder head. The amount of fuel supplied was measured with a volumetric fuel flow meter, calibrated in-house, taking into account liquid-compressibility effects and changes in density with temperature. The measurement error and repeatability of the fuel flow rate spans between +/- 0.2%. Depending upon the dataset presented in the results section, two different gasoline blends were used. Neither gasoline blend contained ethanol. The fuel specifications are listed in Table 2.

Table 2. Fuel properties.

	Chevron-Phillips Research-Grade Gasoline *	Haltermann Solutions Certification Gasoline
Antiknock Index (R+M)/2	86.9	92.7
RON	91.0	96.6
MON	82.7	88.7
Specific gravity	0.746	0.743
Carbon [wt%]	86.37	86.49
Hydrogen [wt%]	13.55	13.34
Oxygen [wt%]	0.08	None detected
A/F Stoichiometric	14.60	14.56
Lower Heating Value, gas-phase [MJ/kg]	43.10	42.97
LHV for stoichiometric charge [MJ/kg]	2.763	2.762
Hydrocarbon type [vol%]		
Aromatics	24.5	34.3
Olefins	5.1	0.12
Saturates	70.1	64.0
Distillation [°C]		
5%	56.9	43.3
10%	63.4	52.8
30%	78.9	80.6
50%	96.1	106.1
70%	114.1	117.8
90%	147.3	159.4
95%	164.9	172.8

* Based on analysis provided by Paragon Labs, Inc.

Prior to starting the experiments, the engine was preheated to 100°C by means of electrical heaters on the “cooling” fluid and lubricating-oil systems. In addition, the intake tank and plumbing were preheated to a minimum temperature of 60°C to avoid condensation of the fuel in the case of the PM fueling system. The minimum T_{in} in the case of the DI injection was 30°C. An auxiliary heater mounted close to the engine provided precise control of the T_{in} .

This work uses an equivalence ratio based on total charge mass, rather than air alone. This equivalence ratio, referred to as the charge-mass equivalence ratio (ϕ_m), is defined by Equation 2⁷, where

7. This definition is similar to the one used by other authors [3, 22]. It differs from ϕ , which additionally accounts for the residuals. For our engine, the residual gas fraction is small (lower than 4%-mass) so that $\phi_m \approx \phi$.

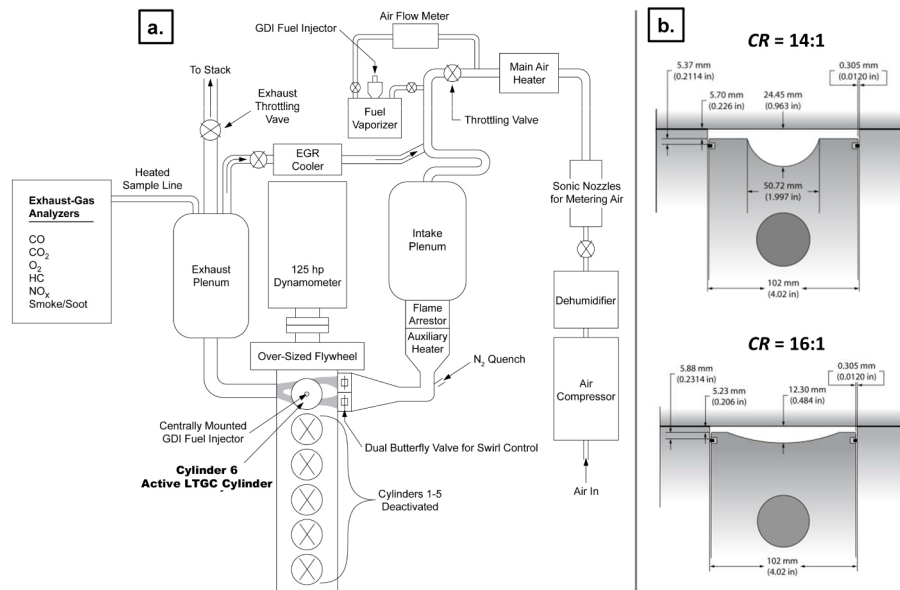


Figure 1. Schematic of the LTGC engine facility (a.) and combustion chamber geometries of the CR = 14:1 and 16:1 at TDC (b.).

F/C is the mass ratio of fuel and total charge gas (*i.e.* fresh air and EGR), and $(F/A)_{\text{stoich}}$ is the mass ratio of stoichiometric fuel/air mixture for complete combustion. This provides a convenient and consistent way to compare data with the same supplied energy content per unit charge mass (*i.e.*, the same dilution level) for operating conditions with different fuels and different EGR levels. Note that ϕ_m is the same as conventional air-based ϕ when no EGR is used. It should also be noted that the air-based ϕ is < 1 for all conditions presented.

$$\phi_m = \frac{(F/C)}{(F/A)_{\text{stoich}}} \quad (2)$$

Prior to starting the experiments, the engine was preheated to 100°C by means of electrical heaters on the “cooling” fluid and lubricating-oil systems. In addition, the intake tank and plumbing were preheated to a minimum temperature of 60°C to avoid condensation of the fuel in the case of the PM fueling system. The minimum T_{in} in the case of the DI injection was 30°C. An auxiliary heater mounted close to the engine provided precise control of the T_{in} .

Data Acquisition

Cylinder pressure (P_{cyl}) 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 P_{in} near Bottom Dead Center (BDC) where the P_{cyl} reading was virtually constant for several degrees. Cycle-average firedeck temperatures were monitored with a K -type thermocouple embedded in the cylinder head so that its junction was about 44 mm off the cylinder center and 2.5 mm beneath the surface. Surface temperatures were estimated by extrapolating the thermocouple reading to the surface, using the thickness of the firedeck and assuming that the back surface was at the cooling-water temperature. T_{in} was monitored using

thermocouples mounted in the two intake runners close to the cylinder head at the flow-centerline. Flow-centerline exhaust temperature was also measured with a thermocouple located about 200 mm from the exhaust valves. For all data presented, 0°CA is defined as TDC intake (so TDC compression is at 360°CA).

The CA of the 50% burn point (CA50) was used to monitor the combustion phasing. CA50 was determined from the cumulative Apparent Heat-Release Rate (AHRR), computed from the low-pass filtered cylinder-pressure data described below. Computations were performed for each individual cycle, disregarding HT and assuming a constant ratio of specific heats [23]. The average of 100 consecutive individual-cycle CA50 values were then used to monitor CA50 during engine operation and for the values reported. The low-pass filter was set so that frequencies below 2.5 kHz were unaffected, while frequencies from 2.5 to 3.9 kHz were progressively attenuated using a Gaussian roll-off function. This smoothly truncates the frequency power spectrum at the near-zero energy region between the lower frequencies due to combustion and the higher frequencies due to knock [8],[20].

The operating range of LTGC engines is often limited by the occurrence of knock, characterized by a distinctive sharp sound, an increase in HT that reduces the η_{th} and could result in engine physical damage over time. In this work, the correlation for Ringing Intensity (RI, Equation 3) developed by Eng [21] is used to indicate the propensity for engine knock. The RI, expressed in MW/m², is a metric designed to correlate with the acoustic energy of the resonating pressure wave.

$$RI \approx \frac{1}{2\gamma} \cdot \frac{\left(\beta \cdot \left(\frac{dP}{dt} \right)_{\text{max}} \right)^2}{P_{\text{max}}} \cdot \sqrt{\gamma \cdot R \cdot T_{\text{max}}}$$

In this equation, $(dp/dt)_{\max}$, $P_{\max}$ and $T_{\max}$ are the peak values of the PRR (in real time: kPa/ms), cylinder pressure, and the mass-averaged charge temperature, respectively, and R is the gas constant. The Peak-PRR (PPRR)⁸ was obtained from the individual low-pass-filtered pressure traces (with a linear fit over a moving $\pm 0.5^\circ\text{CA}$ window), and converted to real-time using the average crankshaft speed, together with the peak mass-averaged temperature and pressure from the ensemble-averaged pressure trace. Based on our experience with this engine at all boost levels tested (*i.e.* $P_{\text{in}} = 1.0$ to 3.2 bar [13]) and mainly at 1200 RPM, using $\beta = 0.05$ ms as a constant (following Eng [21]), a RI criterion of 5 MW/m^2 was found to reasonably correlate over a range of conditions to the onset of an audible knocking sound and the appearance of obvious ripples on the pressure trace (maximum amplitude $> \sim 1$ bar). For comparison, this RI threshold of 5 MW/m^2 selected for operation without knock corresponds to about $8 \text{ bar/}^\circ\text{CA}$ at 1200 RPM (or 58 bar/ms), naturally aspirated. With higher boost levels, higher PRRs can be used for the same audible knocking sound. This is taken into account in Equation 3 by the $P_{\max}$ term in the denominator. The use of a time-based PRR, as opposed to a CA-based one, is intended to eliminate the engine speed dependence on the onset of knock [21]. However, evidence provided in a recent study [20] and supported in the current study, suggest that the RI correlation might not be effective for large changes in engine speed.

Exhaust emissions data were also acquired, with the sample being drawn just downstream of the exhaust plenum using a heated sample line (See Figure 1a). CO , CO_2 , HC , NO_x , and O_2 levels were measured using standard exhaust-gas analysis equipment. For all data reported (even for the knocking cases), NO_x were well below (typically more than an order of magnitude) below the US-2010 limit (*i.e.* 0.27 g/(kW.h)). Soot emissions were monitored using an AVL smoke meter and found to be below the detectability limit for all data reported. A second CO_2 meter monitored the intake gases just prior to induction into the engine, which allowed the EGR fraction of the intake gases to be computed. The uncertainty in the exhaust gas measurement is reported to be less than $\pm 1\%$.

Data Analysis

Energy distribution analysis was performed on the “closed” part of the cycle from BDC-compression (180°CA) to BDC-expansion (540°CA). Considering the first law of thermodynamics, the supplied energy is split into the four following energy pathways⁹:

1. Gross-indicated thermal efficiency
2. Combustion inefficiency
3. Heat transfer
4. Exhaust losses (Exh-L)

HT and Exh-L have been evaluated with different methods presented below. This cross-checking was necessary in order to validate the observed trends.

Gross Indicated Thermal Efficiency (η_{th}) and Combustion Efficiency (η_{comb})

The gross-indicated thermal efficiency, denoted as η_{th} , is the metric of primary interest in this study. The intention is to focus on the combustion-related phenomena and to avoid too much dependence on engine technology. In contrast to the brake thermal efficiency, the indicated thermal efficiency does not include the mechanical losses (friction, accessories driving forces, transmission losses, *etc.*). The term “gross” (as opposed to “net”) means that only the “closed” part of the cycle is considered. Thus, the efficiency does not include the pumping losses during the breathing part of the cycle (intake and exhaust strokes). All η_{th} reported in this article are based on the gross indicated work, as determined from the IMEP_g (gross-Indicated Mean Effective Pressure) and are defined as:

$$\eta_{\text{th}} = 100 \times \frac{\text{Gross indicated work}}{\text{LHV}_f \times m_f} \quad (4)$$

where LHV_f is the Lower Heating Value of the fuel supplied, and m_f is the mass of fuel.

η_{comb} is defined as:

$$\eta_{\text{comb}} = 100 - 100 \times \frac{m_{\text{HC}} \times \text{LHV}_f + m_{\text{CO}} \times \text{LHV}_{\text{CO}}}{\text{LHV}_f \times m_f} \quad (5)$$

where m_{HC} and m_{CO} are the mass of the corresponding pollutant species determined from emission measurements.

Heat Transfer (HT)

Three methods have been evaluated to estimate the HT:

Method 1 (Instantaneous):

The total-HT is estimated by considering the general expression for instantaneous, spatially averaged HT (Equation 6). This equation is particularly representative of the global as well as local HT behavior in HCCI-like engines because of the high degree of homogeneity in the HT throughout the combustion chamber for such engines, as demonstrated by Chang et al. [24].

$$Q_w[J] = \frac{60/360}{\text{RPM}} \times \int_{180}^{\text{EVO} = 482} h(\theta) \times A(\theta) \times (T_{\text{gas}}(\theta) - \overline{T_w}) \cdot d\theta \quad (6)$$

In equation 6, Q_w is the total-HT (here in [J]), h is the HT-coefficient deduced from a global model detailed below, A is the exchange surface area, and T_{gas} is the mass-average charge temperature which varies over the cycle and is obtained from the ideal gas law considering real gas properties (*i.e.* variable γ) and $\overline{T_w}$ is the cycle averaged wall surface temperature which was taken to be the extrapolated firedeck temperature as introduced in the “Data Acquisition” section. Regarding HT coefficient, several empirical /

8. The time-based PRR is equivalent to $(dp/dt)_{\max}$ while the CA-based PRR is equivalent to $(dp/d\theta)_{\max}$.

9. See Footnote #3 regarding the blow-by.

semi-empirical correlations have been proposed and modified to estimate this quantity [18, 23, 24, 25, 26, 27, 28, 29, 30]. Among them, the original Woschni correlation [23, 25, 28] has been used in this study because it is still relevant for determining HCCI-like HT as it is shown in the following.

The Woschni correlation is derived from the dimensionless expression of the form $Nu = C \times Re^m$ (Nu and Re being respectively the Nusselt and Reynolds numbers, C and m the scaling constants) for forced steady turbulent convection HT. The Woschni correlation is expressed as (Equation 7):

$$h_{Woschni}(\theta) \left[W/K.m^2 \right] = C_m \times B^{-0.2} \times P_{cyl}(\theta)^{0.8} \times T_{cyl}(\theta)^{-0.53} \times [C_1 \times \bar{P}_S + C_2 \times (P_{cyl}(\theta) - P_{cyl}^{motor}(\theta))]^{0.8} \quad (7)$$

where C_m is the main coefficient, the diameter of the bore B is the characteristic length scale, T_{cyl} is the mass-average gas temperature. T_{cyl} and P_{cyl} take into account the change in the physical properties of the charge-gas (density, viscosity and heat conductivity) with temperature and pressure. In addition, the temperature term includes the influence of radiation on the HT in a lumped form with the convective effect [25]. However, the radiation effect is likely to be small because of the soot-free and low combustion temperatures in typical LTGC engines.

The term in the square bracket to the power 0.8 ($0.8 = m$ in the Nusselt expression stated above) represents the mean gas velocity. The first part of this term ($C_1 \times \bar{P}_S$) represents the gas velocity for a motored engine (unfired) with C_1 described by Equation 8 which takes into account the mean piston speed $\bar{P}_S$ and the swirl ratio S .

$$C_1 = 2.28 + 0.308 \times \frac{\pi \cdot B \cdot RPM/60 \cdot S}{\bar{P}_S} \quad (8)$$

The second part of the velocity term in Equation 7 represents the combustion-induced gas velocity scaled with C_2 . It is proportional to the difference between the P_{cyl} trace under fired and motored conditions. For every operating condition (every set of T_{in} , P_{in} , RPM), 100 consecutive motored pressure traces are acquired, pegged, filtered and averaged, with only supplying dry air to the engine. However, this motored P_{cyl} is not directly used in Equation 7, otherwise this would result in an artifact during the compression stroke. Indeed, the combustion-induced term in Equation 7 would return a non-zero value during compression because the pressure signals of the fired and motored cycles do not overlap as seen in Figure 2. This is because γ of the charge is always lower for the fired case (presence of fuel, residuals and eventually EGR, with higher gas temperature). Therefore, as shown in Figure 2, the “motored P_{cyl} ” used in Equation 7 first tracks the P_{cyl} of fired case during the compression stroke up to the point where both traces intersect. Then, Equation 7 uses the actual measured motoring P_{cyl} .

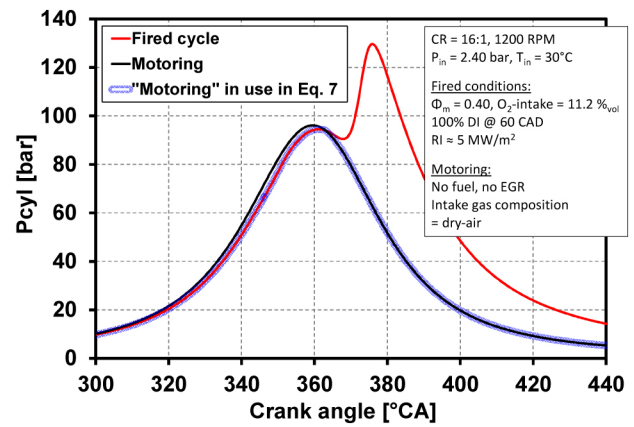


Figure 2. Comparison of the P_{cyl} traces in motoring and fired conditions and illustration of the “motoring” P_{cyl} used in the HT correlation.

The values for the coefficients C_m and C_2 in Equation 7 must still be determined. The main scaling coefficient, C_m , is engine specific [24, 25]. In contrast, C_2 , the scaling coefficient for the combustion-induced velocity term, is sensitive to the intensity of the turbulence generated by the combustion event [25] and is believed to be somewhat dependent to the combustion mode (flame propagation, diffusive flame, sequential autoignition, mixed-modes) and the presence of combustion-induced acoustic oscillations [18].

It has often been reported that using the Woschni correlation with the original values for C_m and C_2 (adjusted for a Diesel engine in 1967 [25]) is inappropriate to assess the HT for HCCI-like LTGC engines [24, 29, 30, 31]. Chang *et al.* [24] conducted measurements of HT in an HCCI engine and modified the Woschni correlation accordingly. They found that C_2 needs to be significantly reduced (*i.e.* divided by 6) for HCCI-like combustion compared to the Woschni original value. They also use the instantaneous chamber height as the characteristic length scale (the bore in original Woschni) and the temperature exponent is recalibrated to be -0.73 . A comparison between the original Woschni and this modified version by Chang *et al.* will be provided in a following subsection.

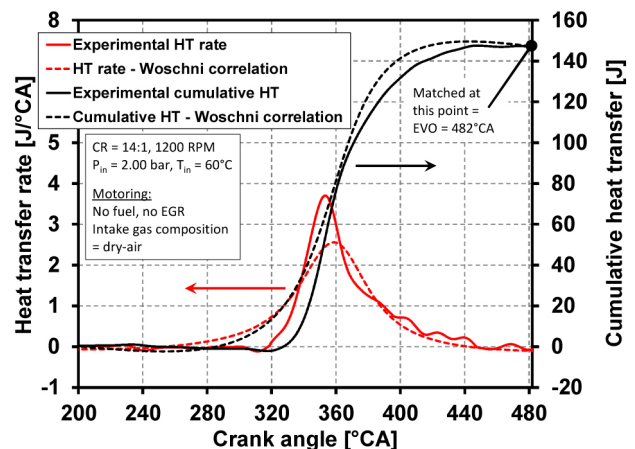


Figure 3. Comparison under motored condition of the HT-rate and cumulative-HT between the experiment and the computation after adjusting C_m in the HT correlation to match the experimental total-HT.

The values of the coefficients C_m and C_2 are adjusted to our engine platform and operating conditions using a two steps approach explained in the following. In a first step, C_m is adjusted by matching the output of Equation 6 to the total-HT determined experimentally by computing the AHRR on a motored cycle (Figure 3). In this manner, C_2 is not participating since the combustion-induced velocity term is equal to zero.

In a second step, with C_m fixed at the value computed from the motored P_{cyl} , the value of C_2 is determined from the fired cycle data. This is accomplished by adjusting C_2 until the cumulative-HR corrected for HT matches the burned energy as described by Equation 9. This adjustment is illustrated in Figure 4. In our methodology, at the start of the combustion (first point shown in Figure 4), the cumulative corrected-HRR is set to zero and the integration is performed until the end of combustion (second point in Figure 4). Note that the exact values for the limits of integration are not critical since the corrected-HRR is nearly flat beyond these points (Figure 4a).

$$\begin{aligned} m_f \times LHV_f \times \eta_{comb} &= (cum. gross. HR)_{end\ of\ comb} \\ &= (cum. AHR + cum. HT)_{end\ of\ comb} \end{aligned}$$

(9)

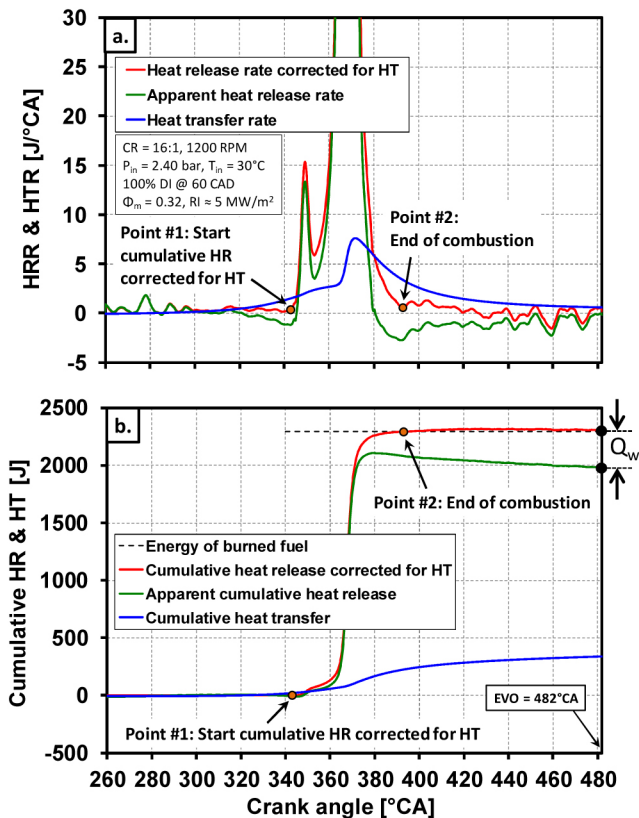


Figure 4. Heat release rate (a.) and cumulative heat release (b.) profiles (apparent and corrected for HT).

It was found that the optimization of the value of C_2 needs to be repeated for every experimental point so that the energy balance described by the Equation 9 is always satisfied. Similarly, C_m needs to be readjusted only when one of the main operating conditions (e.g. T_{in} , P_{in} , RPM) is modified. The value of coefficients C_m and C_2 was found to significantly vary as operating conditions were changed.

This will be shown in the results section for C_2 in Figure 7. Modifying the original correlation and accurately calibrating the coefficients and exponents to make it predictive over the wide range of operating conditions covered in this work is out of the scope of this study. One major concern is the need to include in the combustion-induced velocity term, a parameter to model the effect of the intensity of the acoustic oscillations, as done by Tsurushima *et al.* [18]. In this current work, the information that is not captured by the terms in the correlation like knock is included by a variation of the value of the coefficients. Carefully adjusting the coefficients for every operating point provides a mean to determine fairly well the HT. Indeed, the energy balance equation is always satisfied and as the example in Figure 4a shows, the HRR is well-corrected (the corrected-HRR is nearly flat before and after the combustion event). Moreover, as it will be discussed in conjunction with Figure 6, the total-HT derived from the Woschni correlation was found to be in good agreement with that determined by two others methods introduced below. Finally, trends in the variation of the HT-coefficients provide additional information as will be discussed.

Method 2 (Global Average):

As evident from Figure 4b, the total-HT can also be simply estimated by the difference between the total energy of the burned fuel and the cumulative of the AHRR (Equation 10) until EVO (*i.e.* 482°CA).

$$Q_w = m_f \times LHV_f \times \eta_{comb} - cum. AHRR_{@EVO}$$

(10)

Note that for methods 1 and 2, the computation cannot be performed for CA later than EVO because it is the latest CA for which the ideal gas law still holds (assuming a constant charge-mass during the cycle). This implies that the HT from EVO to BDC is not considered. Thus, the HT is likely to be under-valued for these methods, but, estimates indicate that HT from EVO to BDC represents no more than 1% of the total fuel energy.

Method 3 (Global Average):

The fraction of HT (HT in [% of total supplied energy]) can be estimated as the remainder of the first law energy balance equation (Equation 11), knowing the fraction of heat lost into the exhaust Exh-L (detailed in the next subsection).

$$\eta_{th} + (100 - \eta_{comb}) + HT + ExhL = 100$$

(11)

Exhaust Losses (Exh-L)

Three methods have also been evaluated to estimate the amount of energy lost into the exhaust.

Method 1:

The energy balance equation 11 is applied to indirectly calculate the Exh-L, using the HT-losses obtained with equation 6 (Method 1 or 2 for HT).

Method 2:

The measured exhaust temperature T_{exh} is used to compute the sensible enthalpy of exhaust gas Δh_{exh} in [J]. The enthalpy change between the enthalpy in the exhaust gas (final enthalpy) and a reference-state enthalpy (initial enthalpy) taken at BDC-intake (i.e. 180°CA) is calculated. This is illustrated in Figure 5 (between points 1 & 2) and defined by equation 12,

$$\Delta h_{exh} = m_c \times (c_p (exh. gas @ T-exh) \times T_{exh} - c_p (int. gas @ T-BDC) \times T_{BDC}) \quad (12)$$

where m_c is the charge-mass per cycle and c_p is the specific heat capacity at constant pressure which depends on the gas composition and the temperature. T_{BDC} has been estimated using the procedure developed by Sjöberg and Dec [32] accounting for HT during induction.

One drawback of this method is that the measured- T_{exh} is sensitive to the location of the temperature sensor in the exhaust runner. T_{exh} is affected by the amount of HT during the exhaust stroke¹⁰ and in the exhaust system before reaching the thermocouple. Therefore, the Exh-L would tend to be underestimated.

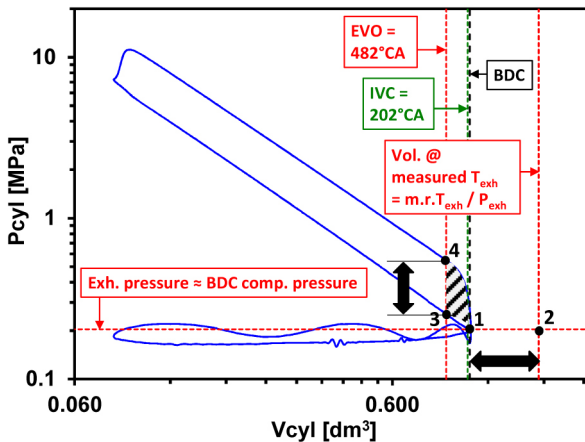


Figure 5. Log-log PV cycle representing the computation of the Exh-L 1) through the measured exhaust temperature (between points 1 & 2: method 2) and 2) through the computed EVO temperature (between points 3 & 4: method 3).

Method 3:

A third method to compute the Exh-L is introduced to tackle the drawback of using the measured T_{exh} . Instead of using the measured T_{exh} , the mass-averaged charge temperature at EVO obtained using cycle analysis is used. The computation is also performed at EVO because it is the last CA for which the ideal gas law still holds. The internal energy change Δu_{exh} in [J] is then computed (Equation 13) between the conditions at the CA of EVO and the corresponding CA in the compression stroke at iso-volume (as seen in Figure 5 between points 3 at 238°CA & 4).

$$\Delta u_{exh} = m_c \times (c_v (exh. gas @ T-EVO) \times T_{EVO} - c_v (int. gas @ T-238) \times T_{238}) - W_{(EVO \rightarrow BDC \& BDC \rightarrow 238)} \quad (13)$$

where c_v is the specific heat capacity at constant volume which depends on the gas composition and temperature. From this internal energy change, it is necessary to subtract the amount of work represented by the cross-hatched area in Figure 5 between BDC to the points 3 and 4, otherwise the Exh-L would be significantly overestimated. However, HT between points 3 - 4 to BDC is still not accounted. Thus, the Exh-L is slightly overestimated.

Comparison of the Different Methods

Figure 6 compares the output of the different methods in terms of HT for an example of a ϕ_m sweep that will be fully described in the “Results” section. Indirect methods using the Exh-L and the energy balance equation 6 are shown with dashed-lines/open-symbols while the methods directly estimating the HT are shown with continuous-lines/ filled symbols. Reasonable agreement is obtained between these different methods. The trends are similar, particularly among the methods that directly compute the HT (curves nearly parallel). Between all methods, the magnitude of the HT is within 6-7% at most, and less than 3% without considering the method using the T_{exh} . The HT-trend determined using T_{EVO} has a slightly higher slope. As mentioned earlier, the method using T_{exh} underestimates the Exh-L. As a result, HT tends to be overestimated. Also, generally the trends appear noisier and the slope is also significantly different from the other methods as seen in Figure 6.

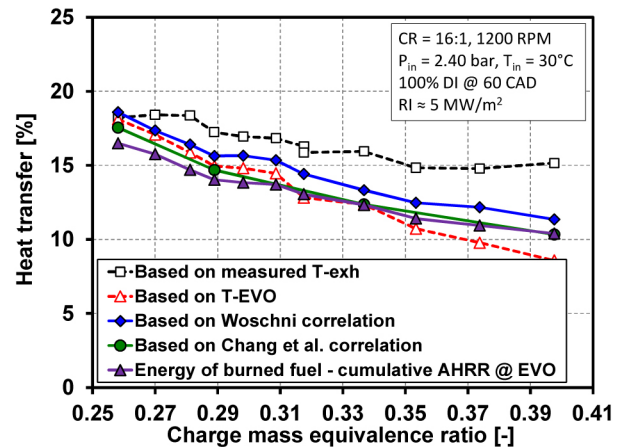


Figure 6. Comparison of the different methods to estimate the total-HT. Example of a ϕ_m sweep at $RI = 5 \text{ MW/m}^2$.

The HT determined from the other methods is likely to be underestimated because they do not consider the HT from EVO to BDC. As mentioned earlier, this omission is responsible of an error less than 1%. A comparison was performed between the Woschni correlation and its modified version by Chang *et al.*, introduced earlier. As for the Woschni correlation, we found that the coefficients C_m and C_2 were also needed to be significantly varied in order to satisfy the energy balance equation. Overall, as seen in Figure 6, even

10. Considering the breathing part of the cycle is not desirable in this study because the energy distribution analysis is carried out on the “closed” part of the cycle.