

Investigation of the Sources of Combustion Noise in HCCI Engines

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ABSTRACT

This article presents an investigation of the sources combustion-generated noise and its measurement in HCCI engines. Two cylinder-pressure derived parameters, the Combustion Noise Level (*CNL*) and the Ringing Intensity (*RI*), that are commonly used to establish limits of acceptable operation are compared along with spectral analyses of the pressure traces. This study focuses on explaining the differences between these two parameters and on investigating the sensitivity of the *CNL* to the ringing/knock phenomenon, to which the human ear is quite sensitive. Then, the effects of independently varying engine operating conditions such as fueling rate, boost pressure, and speed on both the *CNL* and *RI* are studied.

Results show that the *CNL* is not significantly affected by the high-frequency components related to the ringing/knock phenomenon. In contrast, *CNL* is found to be sensitive to increasing energy in the 0.4 to 2.0 kHz frequency range generated by the combustion-induced uniform pressure rise. Parametric investigations emphasize the fact that the *RI* and the *CNL* are designed to provide two distinctly different but complementary measurements. *RI* is designed to correlate with the sound produced by the resonating wave leading to knock. Hence, an *RI* limit appears to be a better criterion for avoiding knocking combustion and its detrimental effects (irritating noise, loss of thermal efficiency, engine damage). However, *RI* is not appropriate for estimating the overall noise of the combustion event since it does not consider the noise generated by the combustion-induced uniform pressure rise. In contrast, the *CNL* provides a valuable metric for the overall loudness of the combustion event. However, the *CNL* is not sensitive to knock-induced noise, so the irritating noise of knock is not accounted for in the *CNL*, and the *CNL* is not appropriate for estimating the onset of knock.

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INTRODUCTION

Concerns about CO₂ reduction, urban air pollution and economic issues have led to increased interest in developing combustion systems for engines that are both highly fuel efficient and have low levels of harmful pollutant emissions. Homogeneous Charge Compression Ignition (HCCI) and HCCI-like Low-Temperature Combustion (LTC) engines have been shown to be capable of providing high thermal efficiencies, ultra-low NO_x and particulate-matter emissions over a large range of operating conditions [1, 2, 3]. However, in spite of these advantages, several challenges remain to be addressed prior to its widespread implementation [3, 4, 5, 6]. Among them, the noise induced by the combustion process is a very important issue.

In most cases, the HCCI combustion event is more rapid than that of conventional engines. HCCI combustion is generally characterized by a relatively high Pressure Rise Rate (PRR), which potentially generates two types of combustion noise.

1. Like any type of piston-engine combustion, a "base" noise (sometimes called "rumble") is generated by the "normal" combustion process as a result of the uniform pressure rise associated with the combustion event. Compared to a Spark Ignition (SI) engine, the base noise of an HCCI engine might be expected to be greater, since the PRR tends to be higher. The noise could be somewhat closer to that of a conventional diesel-combustion process, which also features a relatively high PRR during the initial premixed phase of the combustion event.

2. The second combustion noise source for HCCI engines originates from the propensity of the combustion process to enter into the so-called “knocking” combustion regime. This potential for knock is particularly prevalent as operation is extended to higher loads, which can lead to higher PRRs if steps are not taken to control them. As the PRR increases, it begins to generate acoustic pressure oscillations in the combustion chamber, called ringing in this article¹. If the ringing becomes too intense, it eventually results in a distinctive, sharp and unpleasant sound, commonly known as knock.

Both, the base noise (induced by the normal combustion process) and the development of these acoustic pressure oscillations that can lead to knock, cause vibrations in the engine structure (block) that are subsequently radiated into the surrounding environment. Because of this, combustion-generated noise is a concern for HCCI-type engines. Furthermore, since noise radiated from the engine compartment is among the important factors driving combustion-system design [7], there is a need to address the issue of combustion-generated noise from HCCI-type engines at a fundamental level.

Knock in HCCI Engines

In addition to its unpleasant sound, the occurrence of knock is associated with a substantial enhancement of the heat-transfer losses, believed to be due to a breakdown of the thermal boundary layer by the velocity oscillations associated with the resonating pressure waves [8]. If knock becomes strong enough, generally due to either the load increasing or the combustion phasing not being retarded sufficiently after Top Dead Center (TDC), the knock-induced heat-transfer losses can actually cause a decrease in the thermal efficiency [9]. Finally, in extreme cases, it is considered that intense HCCI knock, could result in engine physical damage over time [10] (somewhat similar to the effect of knock in SI engines).

The induction of knock in HCCI-like engines differs from the process causing knock in SI engines. For the latter, knock is a consequence of end-gas autoignition before the combustion by flame propagation is complete. Because the mixture is stoichiometric, the autoignited end-gas combusts very rapidly, causing a localized rapid pressure rise that excites acoustic resonance frequencies in the cylinder. In HCCI engines, the mixture is dilute, either by being very lean or with high Exhaust Gas Recirculation (EGR) levels. Moreover, at the time of autoignition, the entire combustion chamber contains significant thermal stratification due to heat transfer/convection effects [11]. This results in a slower and more progressive autoignition than in the end gas of SI engines.

The process that leads to HCCI knock is also more progressive than that of SI engines [12, 13]. As the fueling rate is increased and the mixture becomes less dilute, or the combustion phasing is less retarded, the combustion rate becomes more rapid, as indicated by a higher PRR. Low-amplitude ripples appear on the pressure trace, and a spectral analysis of the pressure trace (*i.e.* Fast-Fourier-Transform of the pressure signal) shows a corresponding low power spectral density at the frequency of these ripples. If the PRR is not too rapid, this can occur without the aforementioned detrimental consequences of the knock phenomenon. As the PRR is progressively increased due to further increases in the fueling or decreases in combustion-timing retard, the magnitude of these ripples increases as does the corresponding spectral power content at these frequencies [12, 13, 14, 15]. Eventually, the magnitude of the acoustic pressure oscillations reaches the point that the engine is said to be knocking.

The exact mechanism whereby this more rapid combustion excites acoustic resonances is not completely understood. It is likely that initial oscillations are triggered by a relatively high combustion rate of some localized zones at multiple locations within the charge [3, 16, 17, 18], that interact with each other in a way that excites the resonant modes of the cylinder. The characteristic frequencies (*i.e.* resonance modes) excited can be well estimated using the solution of the wave equation for a right circular cylinder (Draper's equation²) [12, 13, 14, 18, 19]. The majority of the power spectral density of the pressure oscillations and the corresponding sources of the noise from the knock phenomenon are mainly contained in the first circumferential resonant mode (centered at ~ 3 to 6 kHz, depending on the in-cylinder temperature and cylinder bore diameter) [13, 14].

Engine Noise - Perception and Metrics

In addition to the total acoustic power of a sound, as perceived by the human ear, which determines the loudness of an event, the human ear is also subjectively sensitive to the “quality” of the sound. Because of this, the sound generated by the knock phenomenon, which is emitted at higher frequencies and characterized by intermittence/irregularity, is perceived as more irritating than the engine base combustion noise. Accordingly, to improve the perception of the noise emitted by HCCI-like engines, it is of primary importance to ensure that the engine is not knocking. Once care has been taken to avoid knock, it might also be desirable to reduce the engine base noise. However, the acceptable overall loudness from the combustion without engine knock is not well defined, since acceptable noise levels depend on the engine structure, the surrounding equipment in the engine compartment, and perhaps most importantly, the application.

1. Terminology used in this article:

Ringing is defined as the acoustic pressure oscillations developing in the combustion chamber at the specific resonant frequencies of the cylinder. Knock is defined as the situation for which the ringing intensity becomes strong enough to cause any of the following detrimental effects: sharp noise, eventual deterioration of the engine performances, possible engine failure.

2. Draper's equation is presented in the next section, Equation 1.

Several cylinder-pressure-derived metrics are commonly employed to characterize the combustion process in HCCI-like engines. They are also valuable for adjusting and optimizing the control parameters to keep the engine operating well and to avoid engine knock and excessive base noise. The three main metrics: Peak Pressure Rise Rate (*PPRR*), Ringing Intensity (*RI*) and Combustion Noise Level (*CNL*), are introduced immediately below, and detailed descriptions of the *RI* and *CNL* are provided in the next section.

1. The crank angle based *PPRR* (*i.e.* bar/°CA) is perhaps the most commonly used metric. *PPRR* is simply the maximum rate of pressure rise during the combustion event. Early HCCI work used *PPRR* almost exclusively to set a limiting criterion for the harshness of HCCI combustion [3]. However, it was found that a single value of the *PPRR* did not provide consistent results over changes in many engine parameters such as speed and boost pressure. It has now largely been superseded by the other techniques.
2. The *RI*, as proposed by Eng [12], is a quantity that is correlated with the acoustic energy of the resonating pressure wave (expressed in MW/m²) and is designed to quantify the propensity of the combustion to produce acoustic oscillations. It is a convenient and well-accepted metric to forecast excessive oscillations (*i.e.* knock) [3, 13, 20, 21]. By setting a criterion for the maximum allowable *RI*, knocking combustion can be fairly well avoided over a wide range of operating conditions [1].
3. Finally, the *CNL* is a commonly used metric designed to estimate the intensity of the combustion-induced noise emitted through the engine structure. The cylinder pressure is converted to the frequency domain where it is filtered by a transfer function to account for the noise attenuation by the engine structure and how the transmitted noise is perceived by the human ear [22, 23].

Other metrics have been developed to characterize the ringing/knock phenomenon, mainly based on the analysis of the pressure fluctuation amplitudes of the acoustic oscillations [8, 12, 13, 17, 21, 24, 25]. A similar metric related to the magnitude of the acoustic oscillations, the Knock Index (*KI*), is introduced in this study. The *KI* is a quantitative measure of the power spectral content of the pressure trace at the frequencies associated with the first resonant mode. It is defined rigorously in the Data Analysis section, and it will be applied for a specific dataset.

Several studies have investigated how the three main metrics (*PPRR*, *RI*, *CNL*) compare over a wide range of operating conditions (speed/load) and LTC modes [2, 22, 26, 27]. They all generally agree that for small changes about a given engine operating point, variations in *CNL* correlate well with changes in either the *PPRR* or the *RI* (*e.g.* *CNL* increases when either *PPRR* or *RI* is increased). However, there are some significant discrepancies with larger changes in the operating conditions,

or when combustion modes or engine platforms are changed. Poor correlations between the *CNL* and the *RI*, and a somewhat better correlation of the *CNL* with the *PPRR*, are reported. For instance, the *CNL* substantially increases with engine speed and more moderately with boost pressure for a given *PPRR* or *RI*. It was therefore concluded in the aforementioned studies that neither the *PPRR* nor the *RI* well represents the overall noise emitted by the combustion process.

Objectives of the Current Study

The *CNL* has proven to be a useful metric for the noise produced by conventional SI and diesel engines, and it is widely used by industry. With this success, it has been suggested that *CNL* should be applied to advanced LTC engines as a potential alternative to *RI* or *PPRR*, or to introduce an additional constraint parameter in terms of a maximum allowable *CNL*. Accordingly, several studies in the recent literature [2, 22, 26, 27], have compared these three metrics as discussed in the preceding paragraph. However, it is unclear whether establishing an allowable *CNL* limit would be effective for avoiding knocking operation across the operating range, which is of primary importance. Therefore, it is of interest to better understand the sources of the combustion-generated noise and how the various metrics respond to this noise (base combustion noise and noise from acoustic oscillations, *i.e.* knock), over a wide range of operating conditions. The objective of the current study is to provide this fundamental understanding, which is required to correctly interpret and apply the *CNL* and *RI* to HCCI and HCCI-like LTC engines.

In the following section the experimental facility and data-acquisition techniques are described, and detailed descriptions of the various cylinder-pressure analysis techniques are provided. Then, the results of the current study are presented in two parts. In the first part, the study particularly focuses on investigating the sensitivity of the *CNL* to the ringing/knock phenomenon, to which the human ear is quite sensitive. To do so, raw and low-pass filtered pressure traces that truncate the knock frequency components are compared using a dataset for which the knock-onset point is well defined. Further analysis leads to the identification of the frequency range responsible for the overall loudness of the combustion process. In the second part of the results, the main engine operating parameters (*i.e.* load and speed) are varied over a wide range (*IMEP_g* from 2.5 to 16.0 bar and engine speed from 600 to 2400 RPM). This is particularly relevant in the context of the development of high-load, high-speed HCCI engines. The individual effects of the equivalence ratio and the boost pressure on the *CNL* are also investigated and interpreted. Finally, in the last section, the results are summarized, conclusions drawn and areas where further research is needed are identified.

Experimental Setup & Data Acquisition

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-a](#) shows a schematic of the engine, which has been converted for single-cylinder operation by deactivating cylinders 1-5. The engine specifications and main operating conditions are listed in [Table 1](#). Additional information on the facility may be found in Ref. [\[28\]](#).

Data for the present study have been obtained using two different pistons, giving geometric Compression Ratios (CR) of 14:1 and 16:1 as shown in [Figure 1-b](#). The $CR = 14:1$ piston was used in several previous studies, like in Ref. [\[9\]](#), and some of the data acquired in this aforementioned reference have been re-analyzed in terms of CNL . Additionally, new data have been acquired with a recently installed $CR = 16:1$ piston. The $CR = 14:1$ custom piston provides an open combustion chamber with a large squish clearance and a quasi-hemispherical bowl ([Figure 1-b](#)). Comparison tests showed minimal differences in performance and emissions compared to the $CR = 14:1$ broad shallower-bowl piston as used in earlier studies, e.g. Ref. [\[29, 30, 31\]](#). The $CR = 16:1$ custom piston is also characterized by a broad shallow bowl as shown by [Figure 1-b](#). Both of these pistons provide a small top ring-land crevice, amounting to less than 0.9% of the TDC volume. Although one to one CR comparison is not included in this paper, differences

in terms of CNL were found to be small (e.g. less than 1 dB for a typical intake-boosted operating point with $P_{in} = 2.4$ bar, $T_{in} = 30^\circ\text{C}$, early DI fueling and over a large range of fueling rates).

As shown in [Figure 1-a](#), the EGR loop is equipped with a cooler (a gas-to-water heat exchanger), and the EGR is introduced well upstream of the intake plenum. 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. With this configuration, the exhaust pressure must be greater than the intake pressure for EGR to flow into the intake. The required back pressure was achieved by throttling the exhaust flow using the valve shown in [Figure 1-a](#). Intake air was supplied by an air compressor and precisely metered by a sonic nozzle as shown in [Figure 1-a](#). Intake pressures varied from 1.0 bar (simulating naturally aspirated conditions) to 2.4 bar for the current study. All pressures given are absolute.

For operation without EGR, the air flow was adjusted to achieve the desired intake pressure, as measured by a pressure transducer on the intake runner. For operation with EGR, the air flow was reduced from the amount required to achieve the desired intake pressure with air alone, and the valve on the EGR line was opened. The exhaust backpressure throttle valve was then adjusted to produce enough EGR flow to reach the desired intake pressure. This typically resulted in the exhaust pressure being about 2 kPa greater than the intake pressure. For consistency, the back pressure was maintained at 2 kPa above the intake pressure, even when EGR was not used. The EGR fraction was varied by adjusting the amount of supplied air, and then adjusting the exhaust throttle to maintain the desired intake pressure.

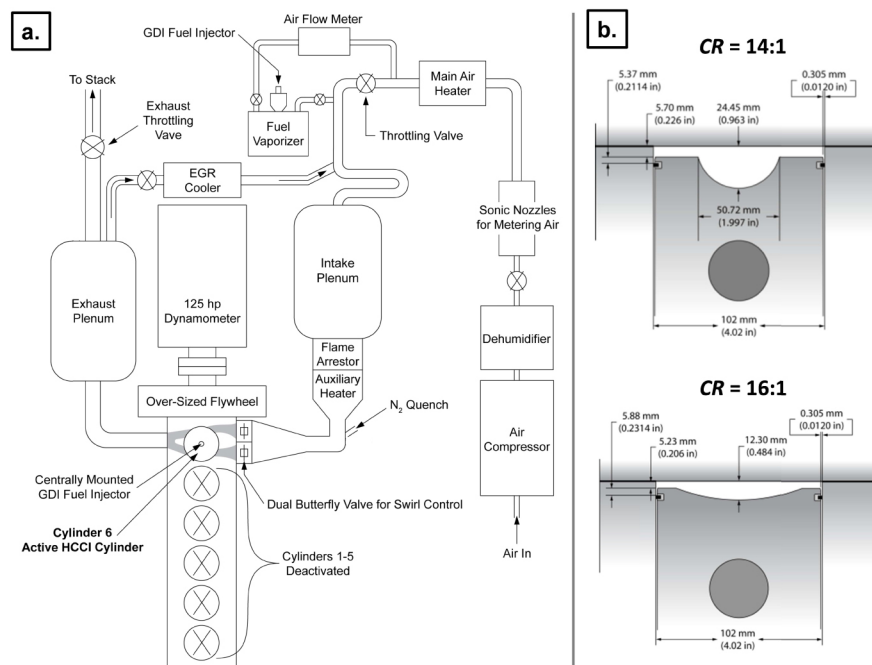


Figure 1. Schematic of the HCCI engine facility (a.) and combustion chamber geometries of the $CR = 14:1$ and $16:1$ at TDC (b.).

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
Number of valves	4
IVO	0°CA *
IVC	202°CA *
EVO	482°CA *
EVC	8°CA *
Swirl ratio	0.9
Fueling system	Fully premixed 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	600 to 2400 RPM
Centerline intake temperature [†]	30 to 181°C
Intake pressure (abs.)	1.0 to 2.4 bar
Coolant and oil temperature	~ 100°C

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

[†] Intake temperature is measured in the center part of the duct in the two runners about 120 mm from the intake valves.

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.

The premixed fueling system, shown at the top of the schematic in Figure 1-a consists of a GDI 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. A positive displacement fuel flow meter was used to determine the amount of fuel supplied.

Depending upon the dataset presented in the results section, two different gasolines were used, but not compared. They include a research-grade regular gasoline supplied by Chevron-Phillips Chemical Co. and a high-octane certification gasoline from Haltermann Solutions. Neither gasoline contained ethanol. The fuel specifications are listed in Table 2.

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 1, where F/C is the mass ratio of fuel and total inducted charge gas (i.e. fresh air and EGR), and $(F/A)_{stoich}$ 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, so combustion is never oxygen limited.

$$\phi_m = \frac{(F/C)}{(F/A)_{stoich}} \quad (1)$$

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 temperature of 60°C to avoid condensation of the fuel in the case of the premixed fueling system. The minimum intake temperature in the case of the DI injection was 30°C. 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 30 to 181°C in this present study. All data were taken at an engine speed of 1200 rpm unless otherwise specified, particularly for the case for which the engine speed effect on the CN_L and R_I was investigated.

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. Centerline 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°CA).

The crank angle of the 50% burn point (CA_{50}) was used to monitor the combustion phasing. CA_{50} 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 heat transfer and assuming a constant ratio of specific heats [32]. The average of 100 consecutive individual-cycle CA_{50} values were then used to monitor CA_{50} during engine operation and for the values reported. The reported PRR and RI are computed from the same low-pass-filtered pressure data. For each cycle, the $PPRR$ was analyzed separately with a linear fit over a moving $\pm 0.5^\circ\text{CA}$ window (5 points)³. Similar to CA_{50} , these individual-cycle values were then averaged over the 100-cycle data set. Examples of the effect of the filtering algorithm will be shown later in relation to Figure 7 and Figure 18. 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. As evident in Figure 7 and Figure 18 (and also demonstrated in [28]), 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.

Data Analysis Methods

Ringing Intensity (RI)

The operating range of HCCI engines is often limited at high loads by the occurrence of the knock phenomenon. It originates from conditions producing a relatively high PRR which triggers pressure oscillations in the combustion chamber at specific resonant modes. If the PRR is not controlled, these acoustic oscillations can reach unacceptably high levels causing the distinctive sharp and unpleasant sound, commonly referred as engine knock. The knock effect can cause a substantial increase in the heat transfer, which under certain conditions, if it becomes sufficiently strong, can cause a decrease in the thermal efficiency [8, 9]. Such a knock increase can occur if load is increased without retarding CA_{50} to control the PRR, or if the combustion phasing becomes too advanced at a given load⁴. Finally, it is considered that intense HCCI knock (due to high PRRs producing strong pressure oscillations) could result in engine physical damage over time [10].

On the other hand, for a given engine load, in order to maximize the thermal efficiency, it is desirable to advance the combustion phasing (when CA_{50} occurs later than $\sim 366^\circ\text{CA}$) to take advantage of the higher expansion ratio [9]. However, advancing CA_{50} can increase the combustion rate leading to high PRRs and eventually knock⁵. As a result, it is appealing to operate the engine as close as possible (considering a safety margin) to the knock onset limit.

It has now been well established [1, 10, 12, 20, 33, 34] that the acceptable ringing limit (*i.e.* the onset of knock) for HCCI engines is not correctly captured by setting a fixed criterion on the maximum allowable PRR ($dP/d\theta$, where θ is a variable representing $^\circ\text{CA}$) under different engine speeds and boost pressures. In these aforementioned references, it was observed during the experiments that with higher boost levels, the combustion chamber could withstand higher PRRs without producing an undue increase in the characteristic audible knocking sound. This is thought to be the result of the increased dampening effect of the pressure waves by the higher in-cylinder density [12, 34]. Moreover, during HCCI combustion, it is likely that weak pressure oscillations (*i.e.* “light” ringing) occur even without any of the detrimental consequences of the knock phenomenon (*i.e.* characteristic knocking sound, lost in thermal efficiency, engine physical damage) [12, 13]. Consequently, the pathway leading to knock is progressive and a criterion needs to be defined to set a boundary between non-knocking and knocking combustion regimes, considering a safety margin.

In this work, the correlation for ringing intensity (Equation 2) developed by Eng [12] is introduced to indicate the propensity for engine knock. The RI , expressed in MW/m^2 , is a measure that correlates with the acoustic energy of the resonating pressure wave. In other words, the RI was designed to correlate with the intensity of the sound generated by the acoustic oscillations. It does not consider the noise emitted the normal combustion event.

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

This correlation includes the main in-cylinder conditions, namely $(dP/dt)_{\max}$, $P_{\max}$, and $T_{\max}$ which are the peak values of the PRR (in real time: kPa/ms), pressure, and mass-averaged temperature, respectively; γ is the ratio of specific heats (c_p/c_v), and R is the gas constant. The $PPRR$ ⁶ was obtained from the low-pass-filtered pressure traces as previously mentioned, and converted to real-time using the average crankshaft speed. The use of a time-based $PPRR$ as opposed to the crank angle based one is intended to eliminate the engine speed dependence [12]. The use of the low-pass filtered pressure

3. The $PPRR$ magnitude from the low-pass filtered pressure trace was not found to be significantly affected by the exact size of the moving average window.

4. An example will be provided in the results section in relation to Figure 4.

5. This aspect will be further discussed in the results section.

6. The time-based $PPRR$ is equivalent to $(dP/dt)_{\max}$ while the crank angle-based $PPRR$ is equivalent to $(dP/d\theta)_{\max}$.

trace for the calculation of the RI was preferred because it seems to give more consistent results. The reasons for this choice are discussed in [Appendix 1](#), which also provides an analysis of the filtering effect on the calculation of the RI . For this study, the computation of the RI using [Equation 2](#) was performed based on the $PPRR$ averaged over 100 cycles together with the peak mass-averaged temperature and pressure from the ensemble-averaged pressure trace⁷.

In [Equation 2](#), the term β is an empirically determined scaling coefficient. It relates the $PPRR$, which is considered by Eng [12] to be the main input force that drives the combustion chamber into acoustic resonance, to the largest peak-to-peak amplitude of the high frequency pressure oscillations: $\Delta P = \beta \cdot (dP/dt)_{max}$. According to Eng [12], the amplitude of pressure oscillations is on the order of 5% of the $PPRR$. However, the range of operating conditions over which this assumption is valid is not clearly specified. For instance, as reported by Maria et al. [20], it is expected that β should significantly decrease with peak pressure (and therefore with the boost level). Indeed, from our experimental observation, for any given $PPRR$, the amplitude of the pressure oscillations ΔP decreases as the cylinder pressure increases. However, it is unclear whether this indicates a need to adjust β with boost, or if the increase in P_{max} with boost in the denominator of [Equation 2](#) is sufficient to compensate.

Based on our experience with this engine at all boost levels tested [1] (i.e. $P_{in} = 1.0$ to 3.2 bar) and mainly at 1200 RPM, using $\beta = 0.05$ ms as a constant (like Eng), a RI criterion of 5 MW/m² was found to qualitatively and reasonably correlate over a range of conditions as being just below the onset of an audible knocking sound and the appearance of obvious ripples on the pressure trace. For comparison, this RI threshold of 5 MW/m² selected for operation without knock corresponds to about 8 bar/°CA at 1200 RPM (or 58 bar/ms), naturally aspirated. In the present study, the accuracy of this criterion to detect the onset of knock is only analyzed in detail for one dataset (first part of the results), the aim of this work not being to validate quantitatively the accuracy of the RI correlation over a wide range of conditions.

A second critical assumption concerns the use of the peak values for the cylinder pressure and the mass-average temperature. This is originally based on experimental observation that the pressure oscillations begin to appear just after the peak cylinder pressure, as reported by Eng [12]. This assumption is in accordance with the results of Dahl et al. [18] using high speed imaging. They found that the first mode acoustic oscillations almost always appear to develop from rapidly combusting regions late in the cycle.

Finally, it has been argued that a drawback to the RI is that it does not consider the frequency distribution of the resonant modes and their attenuation by the engine structure [22, 27]. Because of this, RI is not expected to correlate well with the absolute noise value of the ringing/knocking phenomenon. Also, it should be noted that the RI limit for avoiding knock and the β term in [Equation 2](#) are engine specific. For instance, the RI limit documented by Eng [12] with $\beta = 0.05$, is 2 MW/m² for a small displacement single-cylinder engine (0.55 liter) but 6 MW/m² for a larger-displacement (1.6 liters) engine. Our selected limit of 5 MW/m² is consistent, considering the size of our engine (1.0 liter) and its diesel engine structure. In any case, the RI limit of 5 MW/m² used in the present study was not intended as an absolute ringing/knock metric to be applied to other engine platforms. Rather, it is a metric that provides a means of setting a threshold to avoid knock in our engine over a wide range of operating conditions.

Combustion Noise Level (CNL)

In this study, the Combustion Noise Level (CNL) was estimated through the analysis of the in-cylinder pressure according to the algorithm described by Shahlari et al. [22], who also showed that the output of this algorithm matches well with the AVL Combustion Noise Meter, which is widely applied by industry. This algorithm is based on a technique originally published by Austen and Priede [35], which was subsequently extended by Russell et al. of Lucas in the 70s-80s [23] and used for an instrument sold commercially by Lucas as a "Combustion Noise Meter" at least in the latter part of this time period [36].

The algorithm consists of converting the complete cylinder pressure trace (whole cycle) into the frequency domain by applying a Fast Fourier Transform (FFT) to derive the Power Spectral Density (PSD) in bar²/Hz. Two filters are applied, one to simulate the noise attenuation by the engine structure and the other to account to the response of the human ear (known as the A-filter). [Figure 2](#) illustrates the two filters along with a typical PSD. Notice that the two filters are plotted using two different y-axis scales shown on the right-hand side of the figure. Note from these scales that the effect of the A-filter is much smaller than that of the engine structure filter. Therefore, the PSD of the in-cylinder pressure trace is mainly attenuated by the engine-structure, and in the following plots, only the structure-attenuation filter will be presented. To compute the CNL, the resulting attenuated signal is then integrated over the whole frequency range f and converted to an RMS value to

obtain the Sound Pressure: $SP [Pa] = \sqrt{\sum_{fmin}^{fmax} PSD_{att}(f) \cdot df}$.

Finally, it is converted to decibels (logarithmic scale) by comparing the SP to the human hearing threshold of 20 μPa : $CNL = 20 \times \log(SP/20\mu Pa)$. The CNL calculation is performed for each individual cycle and then ensemble-averaged over the 100 cycles acquired at each condition.

7. A comparison with the RI calculated based on the P_{max} , T_{max} & $PPRR$ from the individual cycles showed negligible differences.

According to Figure 2, a large portion of the “zero” mode and the first resonant mode fall within the 800 to 5000 Hz band where the structure attenuation is ~ 10 dB or less.

The noise can also be expressed in terms of power emitted per unit of surface (linear scale), as given by the Intensity of Sound (I_{sound} in W/m^2):

$$I_{\text{sound}} = (SP/K)^2 \quad (3)$$

where $K = \sqrt{\rho \cdot c}$, with ρ and c are the density of the medium (room air density) and the velocity of sound in the medium $c = \sqrt{\gamma \cdot R \cdot T_{\text{room}}}$.

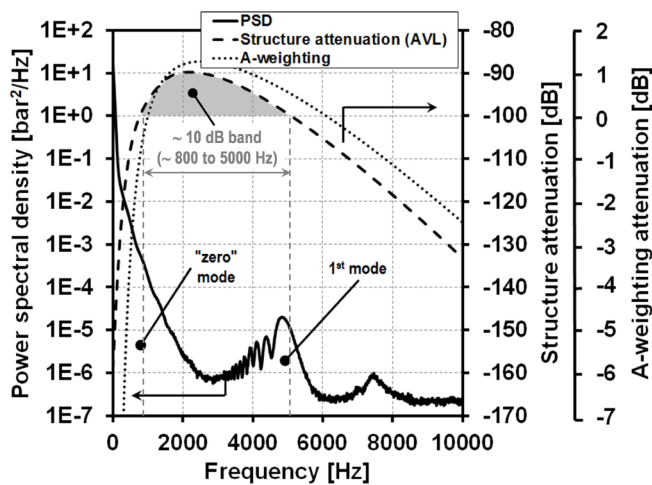


Figure 2. Example of a frequency spectrum of a cylinder pressure trace with the “generic” engine structure attenuation curve and A-weighting filter from Ref. [22], $RI \approx 5 \text{ MW/m}^2$ (onset of knock).

This approach to estimate the *CNL*, assumes a “generic” structure attenuation filter that behaves as a linear system regardless of the operating conditions [23, 37]. It does not consider among other factors, the complex patterns of acoustic energy propagation through the engine structure and its non-linearity with time (acceleration of the block) [37]. The structural attenuation spectrum is a smoothed average of several mid-80s indirect and direct injection diesel engines from various manufacturers (1.0 to 1.6 liter/cylinder) [23, 36, 38]. Nevertheless, this structural attenuation was found to be fairly representative of our engine (1.0 liter/cylinder, diesel structure) [39]. This attenuation filter was developed to adjust the in-cylinder PSD so that the *CNL* correlates with the overall noise measured at $\sim 1 \text{ m}$ from the engine surface [23]. Since the structural attenuation is engine specific, a *CNL* based on this average attenuation filter should not be considered as an absolute measure of the combustion noise [38, 40]. Despite these assumptions, the use of this attenuation filter is considered sufficient for providing relative *CNL* values for a given engine or to compare different combustion systems [23].

The *CNL* is derived by integrating the PSD of the pressure trace. The highest frequency (*i.e.* the Nyquist frequency) contained in the PSD depends on the shaft-encoder resolution and the engine speed. If the sampling rate is too low (not fine enough encoder resolution or engine speed is too low), the aliasing phenomenon might bias the resulting PSD and then the *CNL*. It was found that with the current encoder resolution ($1/4^\circ \text{CA}$ increments) for the engine speed range investigated in this study (600 to 2400 RPM), the aliasing phenomenon does not significantly bias the results. The maximum induced error was about $+ 0.4 \text{ dB}$ (at 600 RPM).

Knock Index (KI) and Resonant Modes

As part of this work, the Knock Index (*KI*) is introduced as an additional means of characterizing the intensity of the ringing/knock phenomenon. This metric is used in the first section of the results to support the selection of $RI \approx 5 \text{ MW/m}^2$ as an appropriate threshold to detect the onset of knock. Moreover, the robustness of the *KI* compared to the *RI* is discussed in greater depth in Appendix 2.

The *KI* is defined as the knock energy obtained by integrating the amplitude of the PSD (in bar^2/Hz) over first resonant mode. As illustrated in Figure 3 an appropriate frequency range of 3.0 to 6.2 kHz was used. The exact values for the limits of integration are not critical. This mode has been chosen because it contains the majority of the acoustic wave energy [12, 13, 14, 18, 27], as it can be deduced from Figure 3. This metric correlates directly with the information contained in the largest pressure oscillations seen on the pressure trace while acoustic oscillations (ringing/knock) are present.

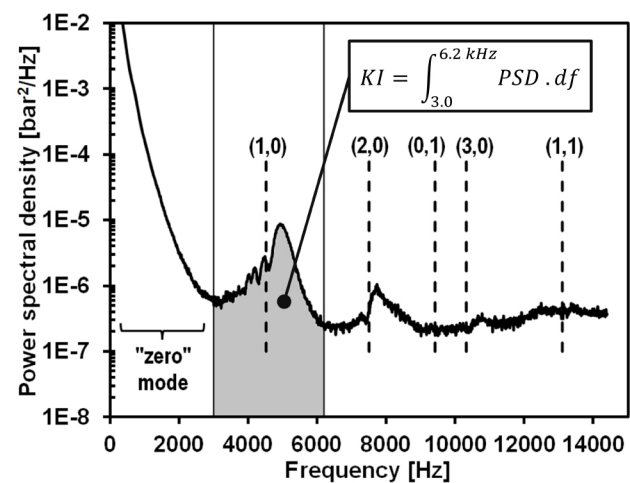


Figure 3. Definition of the Knock Index (*KI*) used in this study. The vertical dashed lines represent the frequency modes (m,n) estimated according to Equation 4. $RI \approx 5 \text{ MW/m}^2$ (onset of knock).

From the plot in Figure 3, it can also be seen the estimated first five resonant modes (m,n) represented by the straight vertical dashed lines, calculated from solution of the wave equation for a simplified circular cylinder (Equation 4 and Table 3) [19]. In Equation 4, $f_{m,n}$ is the resonant frequency for the m (circumferential) and n (radial) pressure node, $\alpha_{m,n}$ is the wave number given in Table 3, $\sqrt{\gamma \cdot R \cdot T_{cyl}}$ is the speed of sound approximated in this study at the mass-average peak temperature and B is the cylinder bore diameter. As seen in Figure 3, the first (1,0) and second (2,0) circumferential modes could be clearly identified from the derived PSD, while the third circumferential mode (3,0) seems to hardly appear. The radial modes are not present on this PSD. Perhaps this is due to the location of the pressure sensor (42 mm off center) which might correlate with the corresponding acoustic radial nodes (compare with schematic in Table 3), or perhaps the radial modes are just very weak or non-existent.

$$f_{m,n} = \alpha_{m,n} \cdot \frac{\sqrt{\gamma \cdot R \cdot T_{cyl}}}{\pi \cdot B} \quad (4)$$

Table 3. Resonant modes for the right circular cylinder [19].

(m,n)	(1,0)	(2,0)	(0,1)	(3,0)	(1,1)
$\alpha_{m,n}$	1.841	3.054	3.832	4.201	5.332

In this study, the contribution to the PSD of the normal combustion process will be defined as the “zero” mode as indicated on Figure 3 (0 to ~ 3000 Hz range). This mode is not resonant and it does not include any significant contribution from the acoustic pressure oscillations. Instead, it is induced by the uniform pressure rise due to the normal combustion process.

Finally, it must be mentioned that the information contained in the resonating modes (including the KI), needs to be interpreted with caution. As illustrated and discussed in depth in Appendix 2, the KI does not always provide consistent trends, depending on the operating conditions.

Three factors might contribute to uncertainty and inconsistency in obtaining quantitative information from analysis of the first-mode integral, *i.e.* the KI :

1. It has been demonstrated that monitoring these acoustic oscillations is sensitive to the location of the pressure sensor relative to the wave direction ([14, 18] and Appendix 2). Sufficient ensemble-averaging should mitigate this phenomenon, due to the assumed randomness of the zone locations generating the acoustic oscillations. However, even for a given engine geometry, results can be inconsistent, depending on whether or not there is a preferred direction for the resonant wave oscillation at the

condition of interest (evidence supporting this statement are provided in Appendix 2). For example, for the first circumferential mode (1,0), the pressure transducer will see much larger oscillations if it is in-line with direction of the pressure oscillation rather than perpendicular to it.

2. Second, there is uncertainty as to the cycle-phasing relative to the combustion event at which these oscillations are developing. This could affect the gas temperature, and therefore the resonant frequency of the oscillations (Equation 4) [27]. Also, the timing of the oscillations relative to the expansion process, could affect the amplitude of these pressure waves and the number of periods for the waves to oscillate in the cylinder.
3. Third, in regard to the KI calculation, the first mode power content might overlap the zero mode and the second mode under some conditions affecting the accuracy of the computed KI .

RESULTS & Discussion

Comparison between the Combustion Noise Level and the Ringing Intensity

As discussed in the introduction, the noise generated by engine knock is perceived by the human ear as more irritating, than the base noise produced by the normal combustion process. Moreover, once knock occurs, it is often perceived by the human hearing system, as dominating the overall combustion-generated noise of an engine. It is therefore of primary importance to ensure that the engine is not knocking. Once knocking is controlled, the loudness of the base-combustion noise is also of concern. In the recent literature [2, 22, 26, 27], and within the engine industry, there has been a rise in interest in the CNL as a method for evaluating the combustion noise of LTC engines, and it has been suggested that the CNL , be used as an alternative constraint parameter to the RI or as an additional constraint for engine optimization. For these reasons, it is of interest to investigate how the CNL responds to the subjectively perceived onset of the knock, even though the CNL is not a metric originally intended to measure the onset and/or intensity of knock, but rather, the overall loudness of the combustion event. Accordingly, the main objectives of this section are to compare the CNL with the RI (which was designed to be a ringing/knock metric [12]), and to investigate whether the CNL is sensitive to, or correlates with, the onset of knock. To accomplish this, our selected RI threshold of 5 MW/m² for the onset of HCCI knock is first compared with other methods of detecting the onset of knock at selected operating conditions. Then, the CNL and RI techniques are applied to these data and their performance compared.

To investigate the relationships between RI , CNL , and engine knock, previously published datasets [9] dealing with the effects of fueling rate on the thermal efficiency, have been analyzed. They consist of two ϕ_m sweeps shown in Figure 4, first without and then with EGR addition to control the RI at ~ 5

MW/m². The intake pressure and temperature were held constant ($P_{in} = 2.0$ bar, $T_{in} = 30^\circ\text{C}$). Some selected low-pass filtered and ensemble averaged cylinder-pressure traces and heat release rates are displayed in Figure 5, corresponding to the four highlighted operating points, labeled from 1 to 4 (on the RI curves) in Figure 4-a. They show the effects on the combustion process of increasing ϕ_m while keeping the control variables constant (operating points #1, 2 & 4), and holding the RI constant with EGR addition as ϕ_m increases (operating points #2 & 3). These datasets have been selected for the current study because they represent a particular case for which the knock onset is not only characterized by a RI of ~ 5 MW/m², but also by a decrease in the thermal efficiency due to higher heat transfer caused by the knock phenomenon, as they will be described later in this subsection. Hence, they include a measurable parameter confirming the presence of knock other than the subjective onset in audible knocking characteristic noise for $RI > \sim 5$ MW/m². Moreover, as stated previously, beyond just looking at the appearance of significant ripples on the pressure trace, the knocking frequency components, particularly the first mode (which contains most of the knocking power), are analyzed by the mean of the KI (Figure 3).

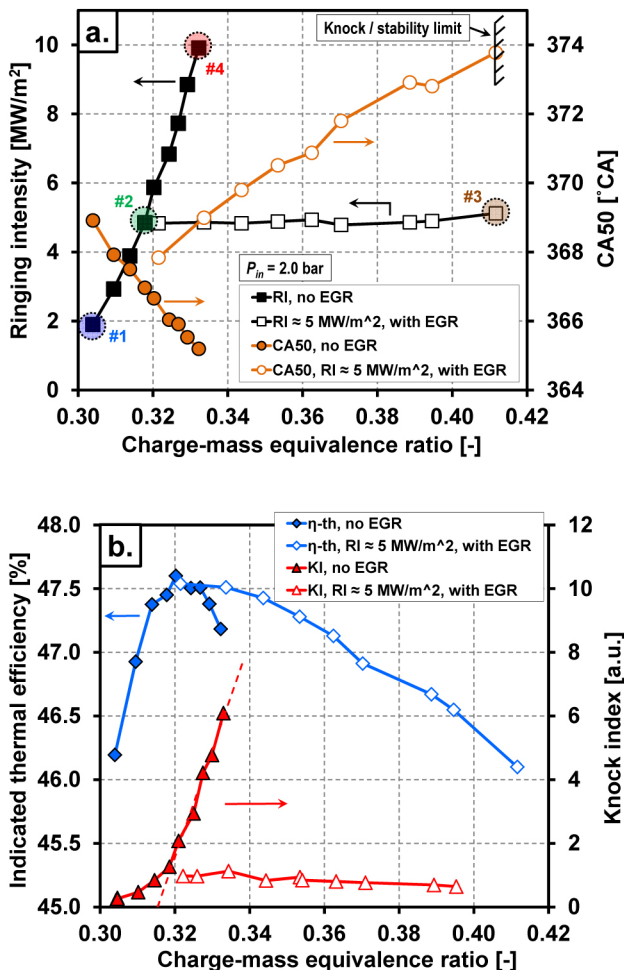


Figure 4. RI & CA50 (a.) and thermal efficiency & KI (b.) for ϕ_m sweeps for zero EGR (filled symbols) and using EGR to maintain a RI of ~ 5 MW/m² (empty symbols). See Table 4 for operating conditions.

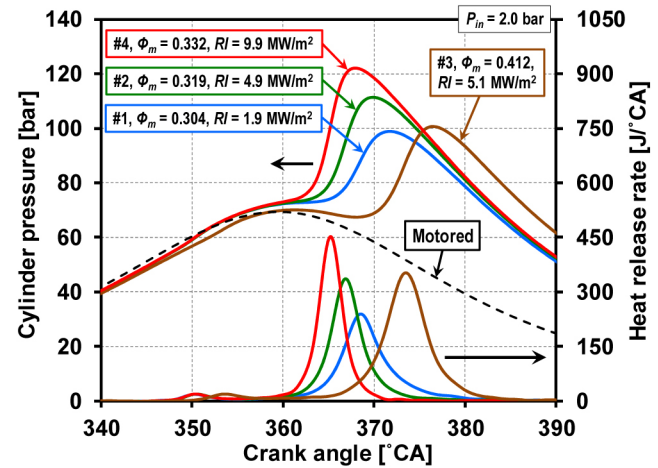


Figure 5. Low-pass filtered and ensemble averaged cylinder pressure traces and heat release rates for the four selected operating points circled (numbered from 1 to 4) in Figure 4-a.

The first sweep, shown by the filled-symbol data points in Figure 4, represents the case for which the fueling rate, quantified by ϕ_m , was increased from 0.304 to 0.332. Because no EGR was used and the intake temperature was held constant, the CA50 advanced from ~ 369 to 365°CA with this fueling increase. As can be seen on Figure 4-a, this results in an increase in the RI from 1.9 to 9.9 MW/m². Referring to the RI equation (Equation 2), despite an increase in the peak pressure (as seen in Figure 5), this increase in the RI is a consequence of a rapid increase in the $PPRR$ from ~ 7 to 18 bar/°CA (or ~ 50 to 127 bar/ms), as it can be noticed in Figure 5, along with a moderated increase in the mass-averaged peak temperature (by ~ 180 K). These effects are caused by a combination of the increased pressure rise resulting from the higher ϕ_m and a more advanced CA50. The latter occurs because the higher- ϕ_m mixture is more reactive at this boost level, and because the higher ϕ_m increases the wall temperatures and the residual-gas temperatures which increases the initial charge temperature (see ref. [9] for further explanations).

For the second sweep, shown by the empty symbols in Figure 4, CA50 was retarded (from ~ 368 to 374°CA) by adding progressively more EGR as ϕ_m was increased from 0.321 (no EGR) to 0.412. Retarding CA50 in this manner controlled the $PPRR$ to hold the RI constant at ~ 5 MW/m². According to Equation 2, the $PPRR$ needs to be slightly decreased as CA50 is more retarded due to lower peak pressures (as seen in Figure 5 for operating points #2 & 3) and higher peak temperatures which increase with ϕ_m . However, these changes in peak pressure and temperature are not very large so the $PPRR$ varies only slightly over the sweep. Eventually, CA50 becomes so retarded that even a small increase in timing retard would cause the combustion to become unstable, but any less CA50 retard or increase in fueling would cause knocking, and the so-called "knock-stability" limit is reached [30, 41, 42]. Furthermore, it is of interest to note that the results in ref. [9] show that, for these same two datasets, the combustion efficiency increases from 93.5 to 98.5% over the