

Comparison of Compression Ignition Engine Noise Metrics in Low-Temperature Combustion Regimes

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ABSTRACT

Many combustion researchers use peak pressure rise rate or ringing intensity to indicate combustion noise in lieu of microphone data or using a combustion noise meter that simulates the attenuation characteristics of the engine structure. In this paper, peak pressure rise rate and ringing intensity are compared to combustion noise using a fully documented algorithm similar to the ones used by combustion noise meters. Data from multiple engines operating under several low-temperature combustion strategies were analyzed. The results suggest that neither peak pressure rise rate nor ringing intensity provides a direct correlation to engine noise over a wide range of operating conditions. Moreover, the estimation of both metrics is often accompanied by the filtering of the pressure data, which changes the absolute value of the results. Thus, all ringing intensity and peak pressure rise rate data should be provided with the filter characteristics to allow an independent assessment of the noise potential. The major difference between ringing intensity and peak pressure rise rate on the one hand and noise on the other appears to be in the speed scaling relationship.

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INTRODUCTION

The development of advanced modes of combustion has received considerable attention recently due, in part, to the need to improve fuel economy and reduce pollutant emissions. Low temperature combustion strategies such as Homogeneous-Charge Compression Ignition (HCCI), Premixed-Charge Compression Ignition (PCCI), Gasoline Compression Ignition (GCI), and Reactivity Controlled Compression Ignition (RCCI) offer lower emissions and / or higher fuel efficiency than conventional diesel or spark-ignition engines. There are, however, many issues that have prevented these strategies from widespread commercialization. Among the challenges that researchers must address, especially in the case of HCCI, is the high heat release rate, which in turn causes very rapid pressure rise rates. The high pressure rise rate is considered to be a major contributor to engine noise because of the associated impulsive loading on the engine structure.

The problem of engine noise has been studied extensively since the 1950s[1]. The noise radiated from the engine structure originates from the mechanical parts of the engine that are subjected to the non-steady loading from the combustion event. The total radiated noise from the engine is a combination of combustion noise (the energy associated with the dynamically varying cylinder pressure) and mechanical noise[1]. Combustion noise, *i.e.*, the energy associated with the cylinder pressure alone, is in the range of 150 to 200 dB[2], which would cause severe damage to hearing tissue. The engine structure, however, attenuates the combustion noise and reduces it by many orders of magnitude (at least by 90 dB)[3].

Investigators have studied combustion parameters that exacerbate or alleviate the combustion-generated noise [1, 2]. For example, advancing ignition timing, using low cetane fuel, and operating at high speed and high load all increase the combustion noise[1, 2]. A wide range of conditions for which combustion-generated noise dominates were tested,

and the noise intensity of the cylinder pressure was compared to the radiated noise from the engine. The transfer function relating the frequency spectra of these two signals has been called the “structure attenuation” curve[2, 3]. To be a useful comparative tool, a smooth curve representing the average of attenuation curves of seven truck and tractor engines was initially determined and called the standard attenuation curve, also known as SA1[2, 3]. Engine structure attenuation was found to increase with improved design, and because the increased attenuation was almost consistent at all frequencies, a new attenuation curve was proposed called SA1-7 [3] (SA1 being the original curve and -7 representing an extra 7 dB attenuation).

Analog and digital devices have been constructed and commercialized to apply this single attenuation curve to the cylinder pressure signal. Examples of such device are the Lucas combustion noise meter and the AVL model 450 combustion noise meter. The human auditory system also acts as a filter. In order to calculate noise levels as they are perceived by humans, an A-weighting is applied to get results in dBA[4]. The attenuation caused by the engine structure is significantly more important in modulating the intensity of the combustion noise than the A-weighting.

COMBUSTION NOISE METRICS

The measurement of engine noise requires an anechoic chamber and sound measurement equipment, and is beyond the scope of standard combustion-system development. As such, the combustion community has long relied on the peak pressure rise rate, *PPRR*, as an indicator of engine noise[5]. Citing discrepancies between observed sound levels and *PPRR*, Eng [6] introduced the ringing intensity, *RI*, (defined fully below) as an indicator of engine noise. Yet another combustion noise metric that is used is the noise level evaluated by directly applying the structure attenuation to the digitized cylinder pressure. The latter two methods are described more fully below.

Ringing Intensity

Eng [6] studied the physics of sound waves and suggested that the power flux or intensity, *I*, transmitted by a pressure wave is indicative of the overall noise generated by an engine. The intensity for a wave of amplitude ΔP is given by

$$I = \frac{1}{2\gamma} \frac{\Delta P^2}{P} \sqrt{\gamma R T} \quad (1)$$

where γ is the specific heat ratio, *R* is the specific gas constant, *P* is pressure, and *T* is temperature. It is important to note that Eq. 1 is only valid for a single wave with a particular frequency. When waves of more than one frequency are present, like in an internal combustion engine, the intensities of the waves are superimposed (as compared to ΔP). The total acoustic energy transfer rate is then

$$I_{tot} = \frac{1}{2\gamma} \frac{\sqrt{\gamma R T}}{P} \sum_n (\Delta P_n)^2 \quad (2)$$

where ΔP_n is the amplitude of the n^{th} mode. The value of ΔP_n for each frequency is acquired by performing a Fourier analysis on the pressure signal. Assigning the correct pressure and speed of sound is challenging in an environment where the temperature is not uniform, and both temperature and pressure are changing in time.

Eng suggested that in HCCI combustion the rapid pressure rise rate is the main cause of the pressure oscillations; therefore he suggested that ΔP scales linearly with maximum pressure rise rate through a dimensional scaling factor β (with dimensions of time), which should be determined experimentally. Further, the temperature and pressure can be approximated as the maximum temperature and pressure, respectively, because the pressure waves exist mainly near the peak pressure timing. Ringing intensity, *RI*, in the form that is most commonly used, is given in Eq. 3, which is derived from Eq. 1 with the assumptions discussed above

$$RI \approx \frac{1}{2\gamma} \frac{\left(\beta \frac{dP}{dt_{max}}\right)^2}{P_{max}} \sqrt{\gamma R T_{max}} \quad (3)$$

The value β is commonly taken as 0.05 ms as this was the value Eng determined for his experimental setup[6].

There are three main assumptions in the derivation of Eq. 3. First is the fact that Eq. 1 was used instead of Eq. 2 to derive the equation for ringing intensity; this ignores the fact that the power flux of a compound wave is the sum of the power fluxes of the underlying harmonic waves. The assumption of a single value of β to relate how the pressure pulsations scale with the maximum pressure rise rate may be reasonable for a specific set of combustion conditions; however, different values of β should be expected for different combustion modes and operating parameters as noted by Eng. The use of the maximum pressure and temperature is reasonable because the pressure pulsations usually occur close to the peak pressure timing. The advantage of applying the first two assumptions is that Eq. 3 can be used in a predictive manner with computational results instead of only being an experimental post-processing tool. However, if these assumptions are relaxed, the equation can more accurately represent the physics by summing over all modes as

$$I_{tot} = \frac{1}{2\gamma} \frac{\sqrt{\gamma R T_{max}}}{P_{max}} \sum_n (\Delta P_n)^2 \quad (4)$$

Even though a Fourier analysis is required to get ΔP_n at each frequency; the sum of ΔP_n^2 for all frequencies can be calculated in the time domain. According to Parseval's identity the energy content in the time domain is the same as in the frequency domain.

$$\int_{-\infty}^{+\infty} |x(t)|^2 dt = \int_{-\infty}^{+\infty} |X(f)|^2 df \quad (5)$$

where $x(t)$ is a waveform in the time domain and $X(f)$ is its Fourier transform pair.

Combustion Noise Meter

Combustion noise meters are typically devices that provide a noise level in dB or dBA by taking cylinder pressure as the input signal. Filters are applied to the pressure signal to first account for the attenuation due to engine structure, and then (optionally) to account for the frequency response of the human hearing system.

The structure attenuation function (also known as the mean free-field response) that was implemented¹ is

$$SA(dB) = \begin{cases} \sum_{i=0}^6 a_i f^i & 100 \leq f \leq 2300 \\ \sum_{i=0}^6 b_i f^i & 2300 < f \leq 10000 \end{cases} \quad (6)$$

where f is the frequency in [Hz], SA is in units of dB, and the coefficients a_i and b_i are given in Table 1. Outside of the frequency range given, the $SA(dB)$ is considered to be a large and negative value, i.e., fully attenuated.

The equation for the A-weighting filter is [7]

$$A(f) = 2 + 20 * \log_{10}(R_A(f))$$

$$R_A(f) = \frac{12200^2 * f^4}{(f^2 + 20.6^2) * \sqrt{(f^2 + 107.7^2) * (f^2 + 737.9^2) * (f^2 + 12200^2)}} \quad (7)$$

and A is in units of dB.

Because the two filters are given in units of dB, the total attenuation of the combined filters is just the sum of the two values, and the total transmission filter, $T(f)$, is found as

$$T(f) = 10^{(SA(f)+A(f))/20} \quad (8)$$

Note that both SA and A are power (amplitude squared) attenuation transfer functions, while T is written to be applied to the amplitude spectrum by virtue of the 20 in the exponent denominator. Figure 1 shows the SA and A transfer functions (note that A is plotted on the right axis) along with the combined effect. It is apparent that the A-weighting is not as significant in comparison to the structure attenuation.

Table 1. Coefficient for the determination of the structure attenuation function.

i	a_i	b_i
0	-1.594243E+02	-1.065899E+02
1	2.029175415E-01	1.896910E-02
2	-2.981767797E-04	-7.393291000E-06
3	2.494291193E-07	1.266005000E-09
4	-1.166026273E-10	-1.278282000E-13
5	2.820300100E-14	7.033316000E-18
6	-2.747693353E-18	-1.621458000E-22

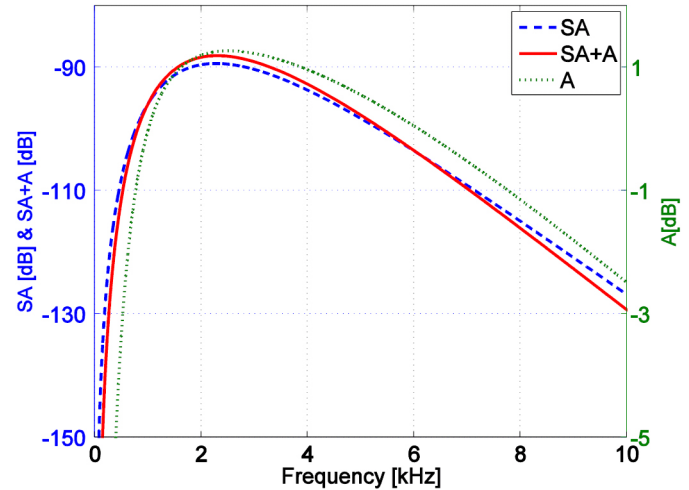


Figure 1. Structure and A Weighting Attenuation Curves

The algorithm for a typical combustion noise meter is pictured in Figure 2. The cylinder pressure is Fourier transformed, and then filtered in the frequency domain. The root mean square (RMS) value of the filtered pressure, P_{RMS} , is calculated, and the final noise level is obtained by comparing P_{RMS} to a reference sound level, which is taken as 20 μ Pa (the human hearing threshold [4])

$$Noise (dB) = 20 * \log_{10} \left(\frac{P_{RMS}}{20 \mu Pa} \right) \quad (9)$$

Although conventional, stand-alone noise meters, like the Lucas noise meter, are analog devices, there is no barrier to performing all of the steps in Figure 2 digitally. A MATLAB code was developed to perform the above calculation and is provided in Appendix A. In order to validate this program, noise levels provided by an AVL noise meter were compared against the noise levels calculated by the MATLAB program for a large number of cycles from two different engines. The comparison showed excellent agreement between the two calculation methods, with a maximum difference of 0.4 dBA.

¹A curvefit was performed to data provided in [7]. There have been several other attenuation functions proposed in the literature (all having a similar shape), but it is beyond the scope of this paper to assess which of these functions is the 'best'. The comparison between the different noise metrics explored here, i.e. the calculate noise versus PPRR and RI, however, is not expected to change with a slightly different attenuation function.

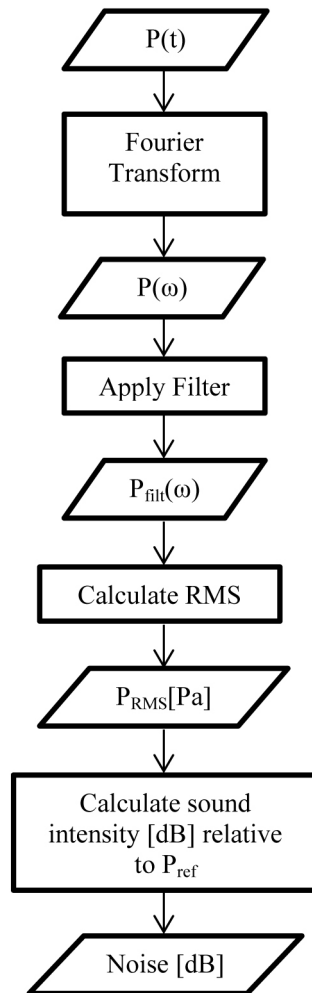


Figure 2. Noise Meter Algorithm

There are three differences between this algorithm and I_{tot} calculated in Eq. 4. First, and most significantly, the pressure is passed through an attenuation filter that mimics the noise reduction behavior of the engine structure and the human hearing system combined. Second, the leading terms in I_{tot} (the speed of sound, maximum pressure, and specific heat ratio) are not considered in calculating noise. Third, the noise level is calculated in decibels relative to a reference sound pressure while I_{tot} is calculated in power flux units (W/m^2). Comparing Eqs. 4 and 9 it is evident that if a flat filter is used in the noise meter, a one-to-one logarithmic relationship between I_{tot} and the noise would exist for a condition where the maximum pressure and speed of sound at maximum temperature do not vary much from cycle to cycle. Because of this, subsequent comparisons between RI and noise (the name that will be used for the output of Eq. 9) will be presented with semi-log axes.

RESULTS

Data from three different laboratories have been analyzed to compare the engine noise metrics. The three engines will be referred to as E1, E2, and E3. Data from E1 included conventional diesel combustion (CDC), HCCI and RCCI operation; data from E2 included low temperature combustion with diesel fuel (LTCD) and GCI operation; and data from E3 was acquired only with LTCD operation. The data represent speeds from 1000 to 3000 RPM, and loads from 4.28 to 14.13 bar gross IMEP.

Individual cycle results

In order to calculate ringing intensity, the specific heat ratio, γ , needs to be identified; a constant value of 1.34 was used for this study. Additionally, β was taken as constant at 0.05 ms.

Figure 3 shows the correlation between RI and $PPRR$ for LTCD operation at a given speed-load point; each symbol represents a different cycle. The data are highly correlated, and a quadratic fit, as suggested by Eq. 3, gives a very high correlation coefficient. This indicates that within a given operating condition (with modest cyclic variation), the RI is

dominated by the $PPRR$ and the $\frac{1}{2\gamma} \frac{\sqrt{\gamma R T_{max}}}{P_{max}}$ factor does not vary significantly from cycle to cycle.

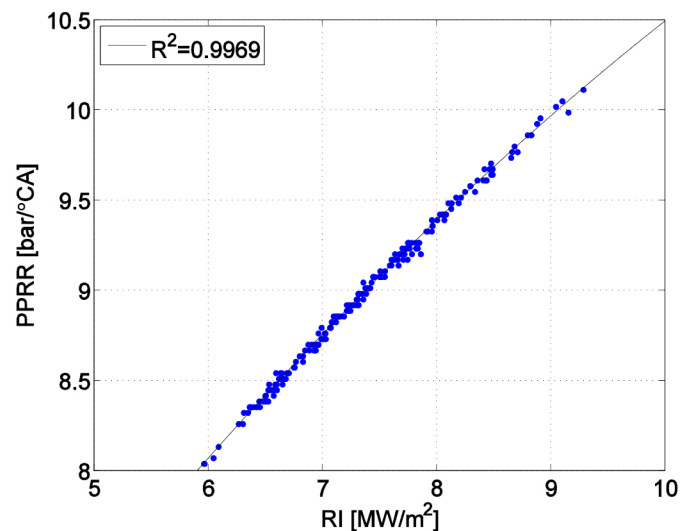


Figure 3. $PPRR$ vs. RI for 200 cycles at a specific operating condition. A second order polynomial fit is provided.