# A Critical Review of Combustion Noise in Combustion Engines

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***Abstract:*** *Combustion noise generated mainly depends upon the rate of in cylinder pressure developed during ignition delay period. Overall design of combustion chamber as well as variations in various fuel injection parameters e.g., injection pressure, amount of fuel injected and its timings also play a crucial role in contributions of combustion noise emissions from engines. The present work issues various factors effecting combustion base noise and analysis its generation using a mathematical model.*

***Keywords:*** *Piston secondary motion, liner system*

## 1. Introduction

Depending upon the type of engine as well as various operational parameters, overall noise emissions from a typical diesel engine may be in range 80-110dBA [1,2, 3]. Split injection using electronic control unit (E.C.U.) may help to shortens the period of premixed phase of combustion and hence help to reduce the overall noise emissions by about 5-8dBA [4]. Head and Wakes have shown that during transient operational conditions, overall noise levels are about 4-7dBA higher as compared to steady state operations [5]. Cold starting conditions may lead to higher ignition delay period which in turn causes increase in the premixed period of combustion [6]. Quality of fuel injected inside combustion chamber also affects the magnitude of combustion noise. It has been observed that a reduction of Cetane number of fuel from 50 to 40 causes a rise of up to 3 dBA in combustion based noise emissions [7]. For a naturally aspirated engine, the combustion-based noise depends upon the quantity of fuel that mixes with air charge during the course of delay period and hence the compression ratio of engines also plays a vital role [7]. In case of gasoline engines, the delay period is longer due to lower compression ratio which may lead to lower temperature of charge and hence more noise emissions [7].

## 2. Background

Due to high efficiency, diesel engines have been a favourite choice in case of heavy-duty automobiles including trucks [8]. However, they suffer from major drawbacks of high noise, weight and vibrations. These engines may be further classified into following two major types:

1. Direct injection (D.I.) engines

2. Indirect injection (I.D.I.) engines

In case of D.I. engines, the fuel is injected directly inside the combustion chamber and as a consequence of it, lesser time is available for formation of fuel and air mixture. Hence a heterogeneous mixture consisting of both rich as well as lean parts is formed inside the chamber.

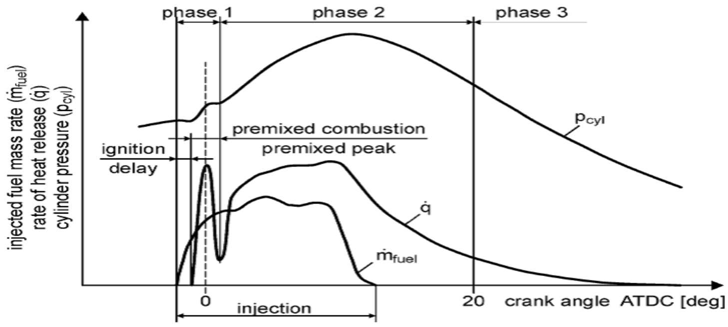


**Fig. 1.** Various phases of Diesel engine combustion [8]

Figure no 1 shows various phases of combustion as observed during course of operation of a typical diesel engine. The delay phase starts with onset of injection process and ends with beginning of premixed phase of combustion. The injection of fuel inside combustion chamber begins a few degrees before TDC position depending on the various injection conditions of engine. As soon as the cold jet of fuel penetrates the chamber, it mixes up with hot compressed air already present inside. The droplets thus formed vaporize, forming layers of fuel-air mixture around the periphery of jet. As the temperature rises to about 750K, the first break down of Cetane fuel takes place. Further propagation of various chemical reactions produces C2H2, C3H3, C2H4, CO2 as well as water vapours [9]. Resulting rise in temperatures causes a complete combustion of fuel-air mixture formed. This sudden period of combustion further leads to rise in the heat release rate as well as high pressure gradient (dP/dθ). This further enhances temperatures in the pre-mixed zone leading to conditions favourable for production of NOx. Once the premixed phase consumes all mixture formed, oxygen available for combustion is consumed around the inner regions wherein the temperatures in ranges of 1600-1700K are reached [8]. Now various partially burnt particles diffuse towards outer layers and begin to burn within a thin region of reaction formed around the periphery of spray leading to formation of a diffusion flame. This phase of combustion is known as diffusion-controlled combustion and is depicted by region 2 and 3 in figure no 1. Higher temperatures along with lack of oxygen provides an ideal condition for the formation of soot.

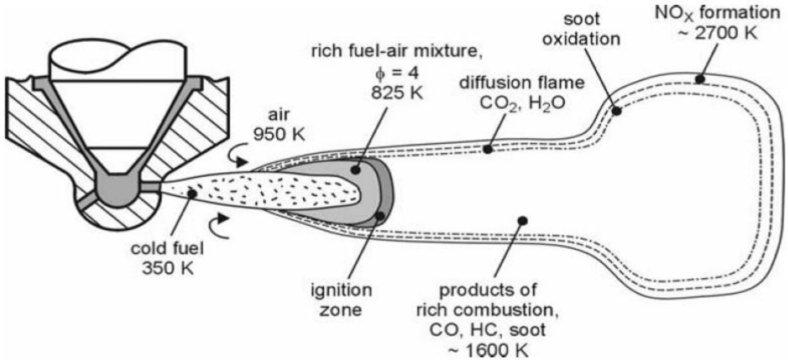


**Fig. 2.** Conventional diesel engine spray formation [8]

The diffusion flame thus formed then uses rest of oxygen available from surrounding environment resulting in high temperatures of order 2700K which consumes all the soot formed. At outer zone of flame there is enough oxygen content for formation of NOx. Figure no 3 shows the rate of soot formation as a function of crank angle. Most of soot that is formed during earlier stages is later consumed and hence final exhaust emissions may have only a fraction of initial soot emissions. As seen from figure no 1, the diffusion-controlled combustion can be divided into further three phases. During the second phase, the burning rate is dependent on rate of mixing of fuel fragments formed and air and hence rate of reaction is faster. During the third phase, oxidation of remaining unburnt particles and soot takes place, however due to decreased temperature of end gas formed during the expansion stroke as well as lesser oxygen content available, slower reaction rates are observed.

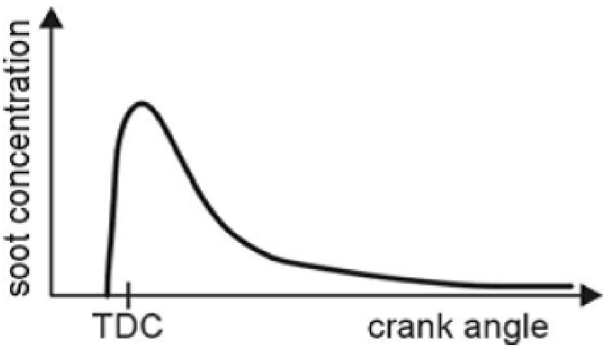


**Fig. 3.** Rate of soot formation [8]

Process of NOX and soot formation in combustion engines has shown an opposite trend as shown in figure no 4. In order to reduce the NOx formation rate it is necessary that local temperatures must not rise beyond 2000K [8]. A possible way to do so is to inject fuel late inside combustion chamber which further shifts the combustion phase towards expansion phase resulting in significant reduction of chamber temperatures. However, rate of consumption of fuel and soot formation increases due to late combustion.

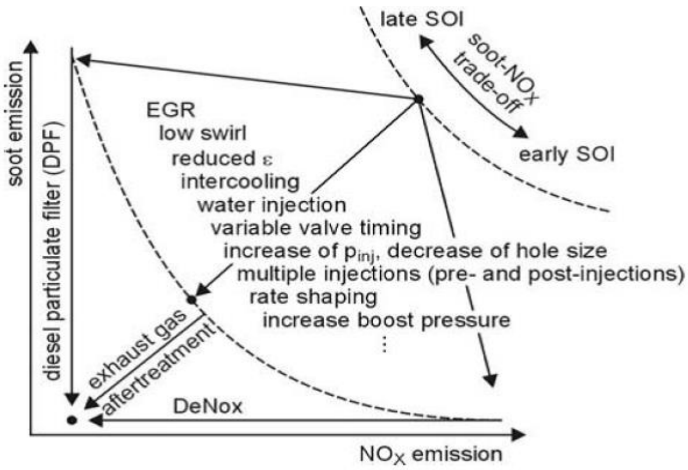


**Fig. 4.** Soot & NOx trade off [8]

Hence modern systems utilize multiple injection techniques in order to control both NOx as well as soot formation rate [8, 9, 10, 11].

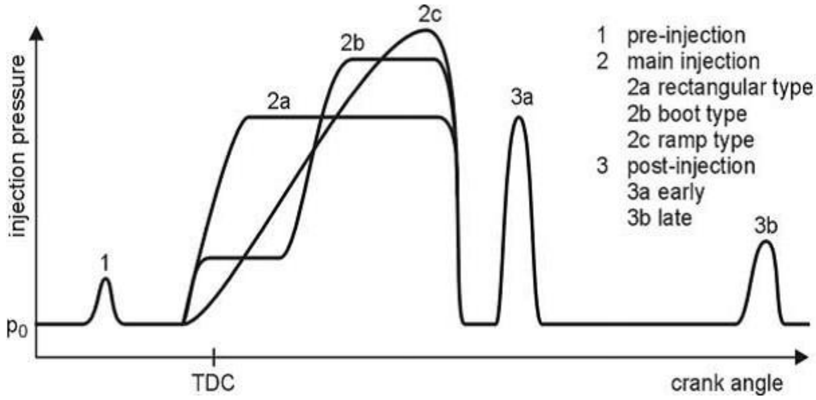


**Fig. 5.** Multiple injection methods adopted for modern diesel engines [8]

There are generally three phases of injection process used, namely pre-injection period, main-injection period & post injection period. There is a delay period between instant at which fuel is injected inside the combustion chamber and actual start of ignition process. Greater this delay period, more is the temperature achieved during course of combustion and hence better conditions exist for NOx formation. In order to shorten this delay period, a small amount of fuel is pre-injected before main injection occurs during the phase of pre-mixed combustion. Torque and power produced in engine mainly depends on the duration of main injection period. It is advantageous to vary the injected fuel mass with time in order to reduce the specific consumption of fuel. This is achieved by rate shaping as seen in figure no 5. Rate shaping curve may be rectangular, step or boot type in shape. Post-injection of fuel is done in order to reduce the soot emissions and in some cases may be useful for exhaust gas recirculation treatment [12]. It has been reported that post injection may reduce the rate of soot formation by about 70% without increasing the fuel consumption [13].

## 3. Factors effecting combustion noise [20]

The rate of pressure (which mainly depends on the ignition delay period) and quantities of combustion gas formed during this period is a key parameter to analyze combustion-based noise. A shorter delay period means lesser amount of combustible gas formed and hence lesser combustion noise. Hence delay period must be reduced as much as possible for effective reduction of combustion noise. Structure and layout of engine also plays a significant role [14-20]. Between 300Hz-2000Hz they were related to first derivative of in cylinder pressure, whereas above 2000Hz were related to second derivative of cylinder pressure. Increase in the compression ratio and chamber temperature may shorten this delay period. However, an increase in compression ratio can cause a rise in noise due to slapping motion of skirt. Various parameters of fuel injection system like instance of fuel injection, injection pressure, number of nozzles and fuel supply rate also effects the combustion noise. Increasing the pressure of injection or engine speed leads to an increase in the amount of fuel accumulated during the delay period resulting in rise of combustion noise.

There are many approaches to control combustion noise. One of these includes reducing cylinder pressure spectrum typically in middle and high frequency ranges. Other include reducing ignition delay period or amount of combustible gases formed during this period. Increasing the stiffness of parts, use of turbo charging process and use of split injection methods have also proved to be other effective methods.

## 4. Effects of heat release rate

Previous works have shown a relationship between peak of combustion noise and overall heat release rate [4,5]. It was observed that higher slopes of rate of heat release curves led to higher combustion noise irrespective of fuel injection timings [2]. There is a tradeoff between combustion efficiency and the noise generated due to combustion [7]. Efficient combustion leads to higher heat release rate near top dead center position which further gives rise to high frequency components in noise spectrum. Late release of heat release leads to lower pressures and subsequent lower frequencies in spectrum.

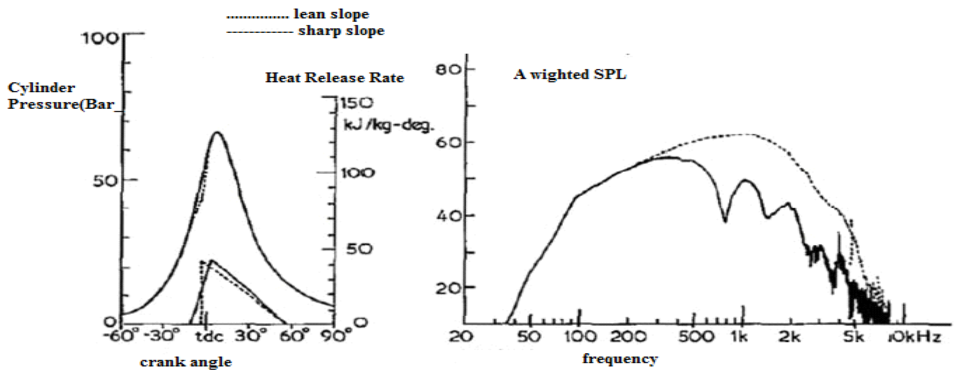


**Fig. 6.** Effects of heat release rate on combustion noise

It has been observed that up to a 10dB reduction in sound pressure levels were possible without change in fuel consumption as shown in figure no 4.66. However, there is a fall in efficiency of cycle and smoke emissions increase if ROHR is not terminated 50° after TDC position (ATDC) [7].

## 5. Effects of cyclic variations

Combustion process in diesel engines varies from cycle to cycle which may lead to variations in noise emissions from engines [28]. These variations may be attributed due to different fuel injection rates, compression ratios as well as difference in fuel spray process, mixture formation and flame propagation. Torregrosa has studied these cyclic variations as shown in figure 7 [9]. These variations were attributed to resonance phenomenon occurring in combustion chamber.

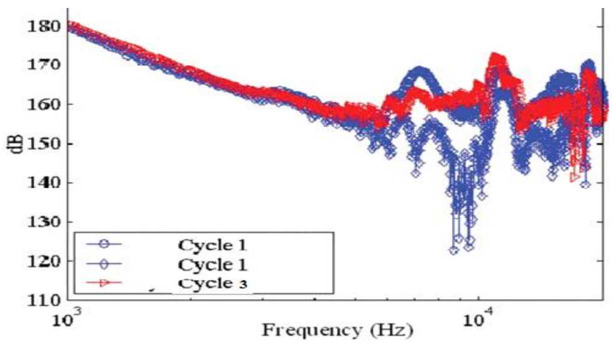


**Fig. 7.** Cyclic variations in combustion noise

## 6. Resonance phenomenon

Resonance taking place inside combustion chamber effects the noise emissions. Grover observed high peaks in the noise spectrum which may be attributed to this phenomenon [3]. Hickling found peaks of higher amplitude by filtering data in range 20Hz-1500Hz which showed increased with engine load [2]. The amplitude of oscillations due to resonance is dependent on the geometry of bowl as well as temperature of gas [3]. Resonance phenomenon can be considered as an unsteady process as it changes continuously during course of combustion process [4].
The frequency of resonance ($f_r$) may be defined in terms of cylinder bore(B), axial length(L) and speed of sound(C) as:

$$f_r=\sqrt{(\frac{C}{2L})^2+(\frac{q_{m,n}}{B})^2} \qquad (1)$$

Where m,m,k determines the circumferential, axial and radial modes as shown in figure no 8.

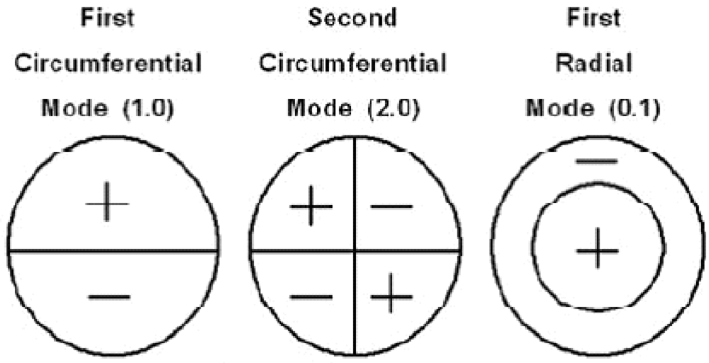


**Fig. 8.** Various modes of combustion chamber cavity

## 7. In cylinder pressure decomposition method

A methodology to decompose the total in cylinder pressure signals was first proposed by Payri [5]. In this method the cylinder pressure was decomposed into three parts namely: combustion pressure, resonance pressure and compression pressure. The combustion part was dependent on injection process and hence resulting ROHR. Using suitable cutoff frequencies, the motored part of pressure can be isolated from excessive part (which is sum of compression and resonance part. Figure no 9 shows results of this methodology as applied to in cylinder pressure [5].

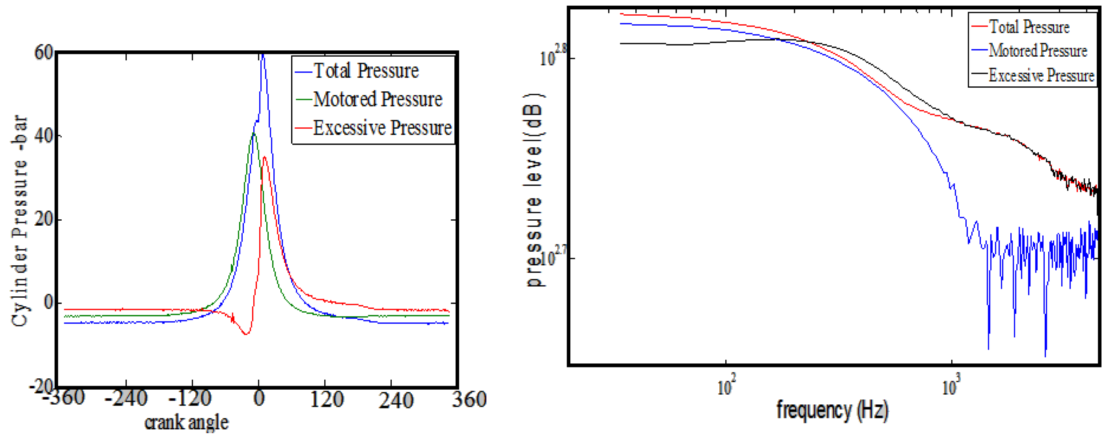


**Fig. 9.** Decomposition of cylinder pressure signal

As evident from these figures, motored part dominated at low frequency ranges. Resonance portion is clearly visible with fluctuating peaks. This portion can be isolated by filtering excessive pressure part using a high pass filter. The contribution due to combustion process can be obtained by subtracting resonance portion from excessive portion. These contributions dominated in mid frequency ranges, whereas the resonance phenomenon dominated at high frequency ranges [5].
The decomposed signals thus obtained can be further used to calculate various indices defined in terms of ideal engine speed ($N_{ideal}$) as:

$$I_n = \log\left[\frac{N}{N_{ideal}}\right] \quad (2)$$

$$I_1 = \left[\frac{N}{N_{ideal}}\right]\left[\frac{(\frac{dp}{dt})_{pilot} + (\frac{dp}{dt})_{main}}{(\frac{dp}{dt})_{motored}}\right] \quad (3)$$

$$I_2 = 10*\log(10^6 \int (\frac{P_{residual}^2}{P_{motored}^2})dt \quad (4)$$

Where $(\frac{dP}{dt})_{pilot}$ is maximum pressure gradient during pilot injection period, $(\frac{dP}{dt})_{main}$ is maximum pressure gradient during main injection, $P_{residual}$ is residual pressure and $(\frac{dP}{dt})_{motored}$ is maximum pressure gradient in motored pressure signal. These indices can be further used to express overall noise (ON) emitted from engine given by:

$$ON = C_0 + C_1I_1 + C_2I_2 + C_nI_n \quad (5)$$

Where constants $C_0$,$C_1$,$C_2$ and $C_n$ depend upon size of engine.

## 9. Mathematical Model of Generation of Combustion Noise [7]

In this part of work a transient model of generation process of combustion noise has been discussed. For this purpose, aMorlet wavelet was defined in terms of its central frequency $f_c$ and bandwidth fb as:

$$\psi(t) = \frac{1}{\sqrt{\pi f_b}} e^{i2\pi f_c t} e^{\frac{-t^2}{f_b}} \quad (6)$$

Figure no 10 shows real and imaginary parts of this wavelet having $f_b$ = 1.5 and $f_c$ = 1

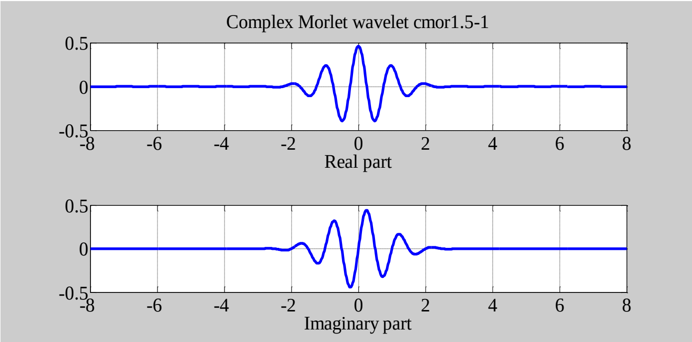


**Fig. 10.** Complex Morlet Wavelet

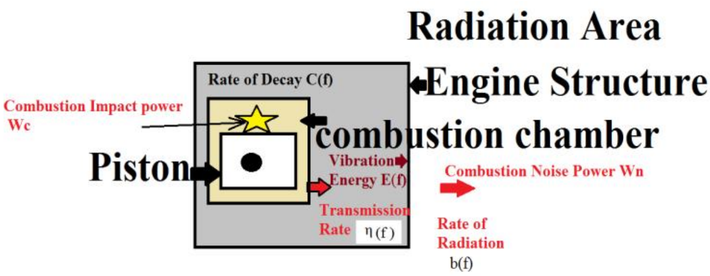


**Fig. 11.** Noise generation model

Further a transient model of combustion noise generation is discussed in Figure no 11. This model can be analyzed by following three processes: generation of vibrational energy inside chamber due to combustion process, transmission and decay of this energy and finally its radiation around the engine surface.
The combustion process inside engines generates combustion impact power ($W_c$) which is related to in cylinder pressure(p), impedance of medium (ρc) and cylinder surface area(A) as:

$$W_c = \frac{P^2}{\rho c} A \quad (7)$$

The available energy at engine surface can be written in terms of transmission rate coefficient η(f) as:

$$E(f) = \eta(f)\int_0^t W_c dt \quad (8)$$

Differentiating both sides of this equation we have:

$$\frac{d}{dt}\big(E(f)\big) = \eta(f)W_c \quad (9)$$

Taking into account the decay rate C(f) this equation gets modified as:

$$\frac{d}{dt}(E(f)) = \eta(f)W_c - C(f)E(f) \quad (10)$$

Where the decay constant C(f) may be defined as:

$$C(f) = -\frac{d[\log(Wn))]}{dt} \quad (11)$$

$$\frac{d}{dt}(W_n) = b(f)\eta(f)W_c - C(f)W_n(f) \quad (12)$$

## 10. Conclusions

This part of work investigated use of various experimental data for diagnosis of combustion-based noise. Further a transient model of combustion noise generation was also analyzed. High levels of correlations observed between various indices, show the applicability of vibration signals from engine as an input feedback parameter in case of a closed loop control system as depicted in figure no 12 [6]. This process can help to achieve an effective control over combustion noise by controlling optimal MBF50/MBF100/ fuel injection timings [7].

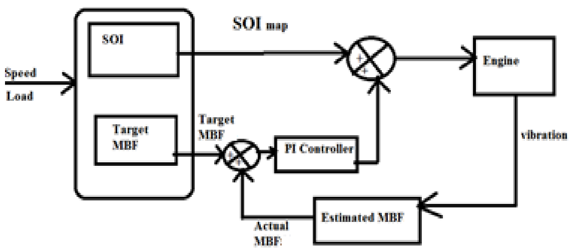


**Fig. 12.** Use of vibration signals as a feedback

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