

Noise Optimization of Diesel Engines with New Combustion Systems

Shanjin Wang, Christophe Chalu, Nicolas Duclaux and Marc Paquien

Renault SAS Powertrain Engineering

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ABSTRACT

The Euro 6 emission standard requires a strong reduction of NO_x and soot emissions for future Diesel engines. One of the ways to reach the Euro 6 standard for new Diesel engines is to adopt the low NO_x combustion concept with new injection strategy, but this kind of combustion can give higher combustion noise and worse stability in transient conditions. This paper describes some of the new methodologies developed by Renault for controlling and optimizing Diesel combustion noise, particularly for engines with low NO_x combustion modes.

In steady working conditions, it was found that a homogeneous combustion mode gave high level of combustion excitation particularly in the 1000 Hz octave band. Thus improvement should be made in the engine structural attenuation (SA) in this frequency range in order to limit engine noise deterioration. This requires not only the technical solutions for improving the structural attenuation, but also reliable methods for measuring engine SA. By using an innovative method based on synchronous Wiener filtering, engine SA can be measured in a fast and accurate way for frequency ranges above 800 Hz, but in the low frequency range, the accuracy is not always satisfactory. For this reason, a new technique has been implemented in the measurement process which enables good results to be obtained even in the low frequency range. Technical solutions for improving engine SA can then be evaluated accurately.

For Diesel combustion noise in transient driving conditions, a new measurement method and a new indicator have been developed which enable engine

transient combustion noise to be quantified in repeatable conditions. Firstly, several representative driving conditions have been standardized, and secondly a new criterion has been defined which takes into account the effect of mechanical noise on the perception of combustion noise. In this way, transient combustion noise of any new engine can be controlled and compared to existing engines, and therefore improvement can be achieved during the early stages of the engine development process.

INTRODUCTION

The further evolution of Diesel engines for passenger cars in the next decade will be dominated by the necessity to meet drastically reduced emission limits, while maintaining and even enhancing the specific benefits of Diesel engines in terms of fuel economy and fun-to-drive. For example, with respect to the current Euro4 emission standard, the future Euro 6 standard imposes a reduction of particulates of nearly 80% and a reduction in NO_x of nearly 70%. Such engine-out emission levels can be obtained either by reducing the emissions directly at source, or by using expensive after-treatment systems such as NO_x traps. In this context, alternative combustion modes as solutions to reduce engine-out emissions have been widely investigated by a large number of car manufacturers, in parallel with exhaust gas after-treatment system development. For example, the homogeneous combustion mode, by generating a lean and fairly homogeneous mixture, is a very promising way for obtaining low levels of NO_x directly in the combustion chambers [1, 2].

However, one of the risks with these alternative combustion modes is the deterioration of combustion noise. Therefore, in addition to the necessary efforts for reducing and optimizing combustion excitation in the chambers, it is important to develop suitable solutions and methodologies for improving the engine structural attenuation, such that the radiated combustion noise of the engine can be maintained at an acceptable level even with an increased combustion excitation.

ANALYSIS OF COMBUSTION EXCITATION

A major difference between a low NO_x combustion mode and a conventional one is in the injection strategy. For example, as shown in Fig.1, with the homogeneous combustion mode, the time lag between two adjacent injections is much shorter than that in the conventional combustion between the pilot injection and the main injection. This difference in the injection time lag can significantly change the cylinder pressure spectrum.

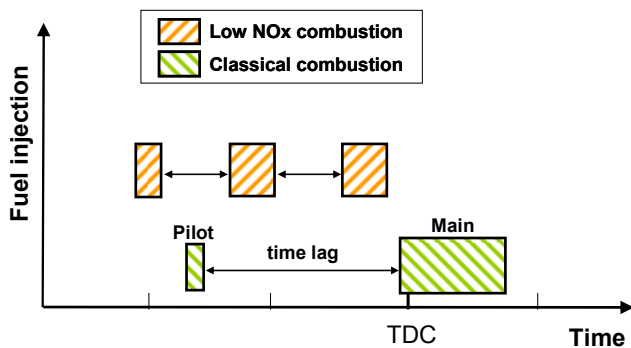


Fig.1 – Comparison of new injection mode with classical one

In general, in a classical combustion system with 2 injections (pilot injection and main injection), the cylinder pressure spectrum is composed of a series of equally spaced lobes, as shown in Fig.2. It has been found that the distance between the lobes is inversely proportional to the time lag between the two injections. Fig.2 shows two examples of cylinder pressure spectrum obtained with a time lag of 1000 μ s and of 2000 μ s respectively. The separation of the spectrum lobes obtained with a time lag of 1000 μ s is doubled, and their amplitudes are more pronounced. Therefore, with respect to a conventional combustion mode, a homogeneous combustion mode is expected to generate cylinder pressure spectra with more separated and more pronounced lobes. Of course, the cylinder pressure spectrum depends not only on the time lag but also on other injection and combustion parameters.

Fig.3 shows two realistic spectrum results, obtained with a homogeneous combustion mode and with a conventional mode respectively. With the homogeneous

combustion mode, the combustion excitation is increased in the medium frequency range between 600 Hz and 1300 Hz, but in the high frequency range the excitation level is lower.

This phenomenon can have a high impact on the vehicle interior noise, because most of the current isolation solutions in a vehicle body are efficient only in the high frequency range (> 1000 Hz). For engine noise in the low and medium frequency ranges (<1000 Hz), it is more difficult and more expensive to implement isolation solutions in the vehicle body.

Consequently, for Diesel engines with a homogeneous combustion mode, more effort should be dedicated to the improvement of engine structural attenuation, particularly in the mid frequency range, so that the radiated combustion noise of the engine is maintained at an acceptable level.

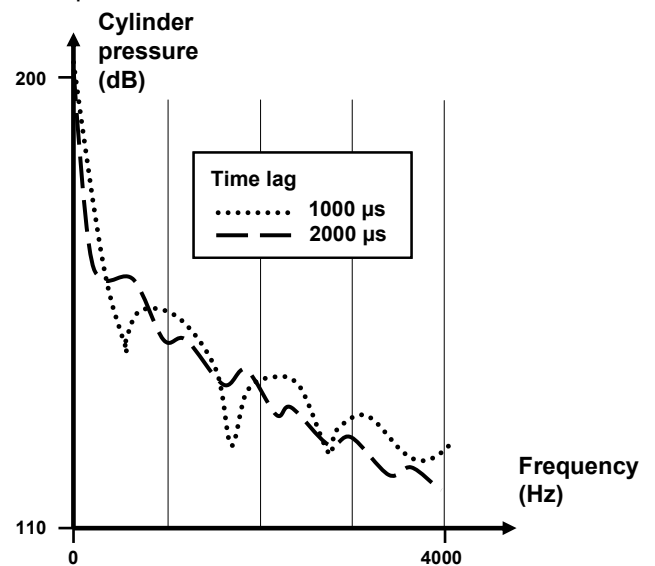


Fig. 2 – Influence of the injection time lag on the cylinder pressure spectrum

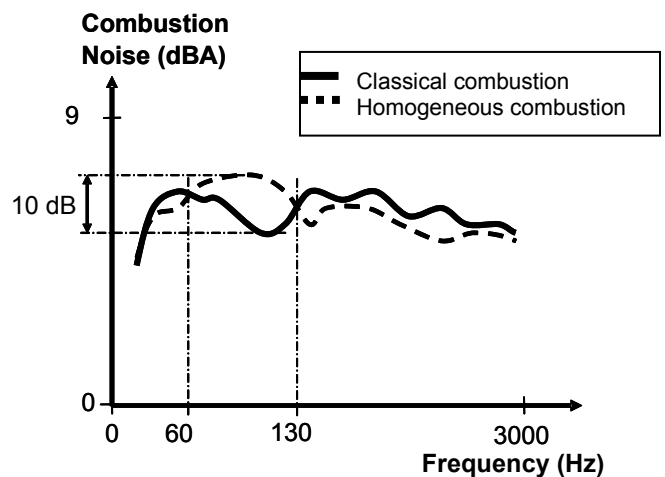


Fig.3 - Comparison of combustion noise spectrum between classical combustion and homogeneous combustion

ENGINE STRUCTURAL ATTENUATION

Engine structural attenuation (SA) is defined as the difference expressed in decibels between the combustion pressure in the combustion chambers and the combustion noise measured by a microphone placed at 1 m from the surface of the engine. The combustion excitation in a chamber can be obtained directly by measuring the cylinder pressure with a piezo-electrical transducer, but the radiated combustion noise cannot be directly measured by a microphone, as the microphone captures not only the combustion noise, but also noise from other sources. For simplicity, noise from sources other than combustion is called mechanical noise in the following parts of this article. If we can separate combustion noise from mechanical noise, then the engine structural attenuation can be calculated. This can be done by means of the synchronous Wiener filtering method [3, 4, 5].

PRINCIPLE OF THE WIENER FILTER

The technique has been developed by N. Wiener in the 1950's for stationary processes.

Let $x(n)$ be the desired (digital) signal of which only a corrupted measurement

$$y(n) = x(n) + b(n) \quad (1)$$

is available, where $b(n)$ is some unknown additive noise.

Let also $r(n)$ be a reference signal that is fully coherent with $\hat{x}(n)$, but completely uncorrelated with $b(n)$. As shown in Fig.4, the aim is to find the best linear filter $h(n)$ which, when applied to $r(n)$, yields an estimate of $x(n)$, i.e.

$$\hat{x}(n) = \sum_t h(t) r(n-t) \quad (2)$$

It can be shown that such a filter should minimize the quadratic distance between $\hat{x}(n)$ and $x(n)$, or equivalently between $y(n)$ and $x(n)$. In other words, $h(n)$ is a prediction filter of $x(n)$ from $r(n)$. In the frequency domain, the solution for $h(n)$ is:

$$H(f) = S_{yr}(f) / S_{rr}(f) \quad (3)$$

with $H(f)$ the Fourier transform of $h(n)$ (i.e. the frequency response function), $S_{yr}(f)$ the cross spectral density between $y(n)$ and $r(n)$, and $S_{rr}(f)$ the auto power spectrum of $r(n)$.

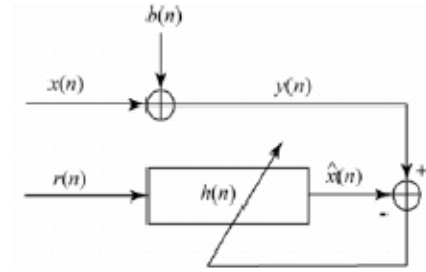


Fig.4 – Principle of Wiener filter

The so-defined Wiener filter is particularly well-adapted to the issue of separating combustion noise on engines, since the reference signal $r(n)$ is naturally provided by the cylinder pressure and the cylinder pressure fulfils the requirement of being uncorrelated with the mechanical noise. However, a major difficulty lies in estimating the Wiener filter from equation (3) since it strictly applies to stationary signals only. This problem has been alleviated by recasting the formulation of the Wiener in the cyclostationary framework, because the waveforms measured on a Diesel engine, although possibly random, repeat themselves with periodic statistics from cycle to cycle. Cyclostationarity allows the definition of a cyclic estimator of the Wiener filter, the so-called synchronous Wiener filter, by considering averages on a cycle to cycle basis instead of time averages as done with stationary signals. This is briefly illustrated in Fig.5, where it is shown how the signals are cut into short portions phase locked on the TDC of each cylinder. These portions are then Fourier transformed for calculating the synchronous Wiener filter $H(f)$. This filter in conjunction with the reference pressure signal provides the contribution of combustion noise as shown in Fig.6.

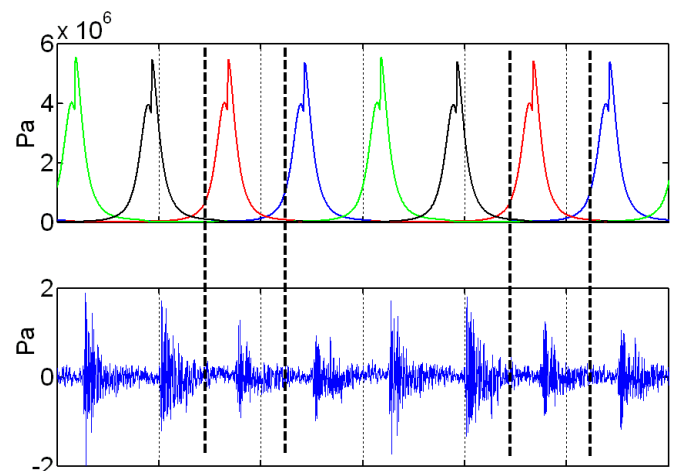


Fig.5 – Re-arrangement of the signals in short portions synchronized on the cylinder pressure