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EXTRACTION AND ANALYSIS OF DIESEL ENGINE COMBUSTION NOISE

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Abstract: *This paper deals with the study of the structural response of an operating diesel engine. The linear relationship assessed in operating conditions between the cylinder pressures and the noise is called spectrofilter, and is commonly used to separate combustion and mechanical contributions from the overall engine noise, using Wiener filtering. A first part of the paper proves the necessity to remove the deterministic part of signals in the identification of spectrofilters, using indirect criterions as the causality of identified filters and their stability to changes in operating conditions. A second part concerns the comparison between identified filters and results of impact hammer tests, where it is shown that spectrofilters can provide useful information about the dynamic behaviour of the engine block.*

Keywords: diesel engine, combustion noise, spectrofilters;

INTRODUCTION

Combustion noise is the contribution of the combustion process inside the cylinders in the overall engine noise. Diesel engine combustions generates a high and rapid rise of the cylinder pressure, acting as an impact on the internal surface of the combustion chamber [1]. The pressure inside the cylinders is easily measured using a pressure sensor through the cylinder head. This information is commonly used by engineers to assess the combustion noise level [2], using a structural attenuation (standard frequency dependent ratio between the in-cylinder pressure and combustion noise). The aim of this work is to present a method allowing the determination of the frequency response functions between the cylinder pressures and output signals (acceleration, acoustic pressure), in order to determine the combustion noise using Wiener filtering approaches [3][4]. This method allows the time domain separation between combustion noise and mechanical noise (defined as the difference between the whole signal and the combustion noise). The difficulty of these approaches is the operational determination of the frequency response functions, also called spectrofilters. This determination is described in the first section of the paper. The validity of a method based on random parts of signals is then presented in a second part, and a third section is dedicated to the comparison of spectrofilters with impact hammer tests, allowing some understandings of structural paths involved in the combustion noise propagation.

ESTIMATION OF THE SPECTROFILTER : THEORY

The spectrofilter of the diesel engine combustion noise is the MISO (Multiple Input Single Output) linear system relating the in-cylinder pressures (inputs) to a vibration or acoustic signal (output), cf. figure 1. It is generally assessed in operation, from the acquisition of all cylinder pressures and the output signal.

The transfer functions between each cylinder and the output are assessed using a H1 estimate. A difficulty is that cylinder pressures are highly coherent, so the identification of transfer functions have to be independently processed, using a synchronized windowing operation capturing in the output signal the contribution of each cylinder. This window generally truncates the output signal for the duration between two consecutive combustions, and its shape has to be optimized with respect to its frequency response [5].

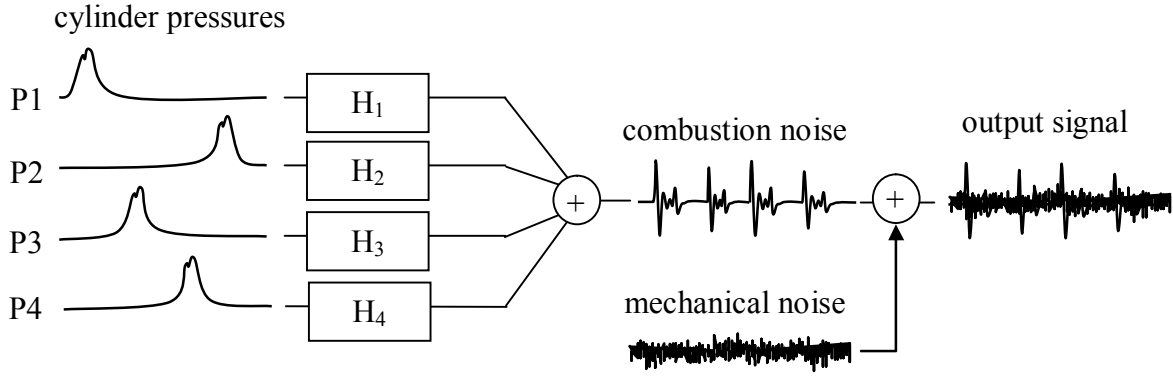


Figure 1. Synoptic diagram of the diesel engine combustion noise definition. Particular case of the four cylinders engine.

This windowing operation is the more critical aspect of the method, because the duration of the window is generally too short to capture the entire vibroacoustic response of the engine (a typical 4 cylinders diesel engine operates from 800 to 4000rpm, resulting in windows length between 8 and 40 ms).

Another difficulty is that some other sources are contributing to the overall noise, and sources that are contributing at the same time than the considered combustion are not cancelled by the windowing operation (for example the piston slap impact occurring in the same cylinder). Moreover, mechanical sources are known to be highly coherent with combustions [6], because of their periodical components. Unfortunately, the H1 estimate is known to be biased in case of correlated noise [7]. A solution to reduce the coherency between the combustion process (the input signal) and the mechanical sources (noise) is to remove the deterministic part of signals using synchronous averaging [8,9]. The H1 estimate of the transfer function is finally assessed from residual parts of input and output signals.

VALIDATION OF THE SPECTROFILTER ESTIMATION BASED ON RANDOM PARTS OF THE SIGNALS

Signals recorded on diesel engines (cylinder pressures and vibration or acoustic output) are known to be highly deterministic [6,9], so removing the deterministic part of such signals can be thought to be hazardous, and a strong decrease of SNR can be expected.

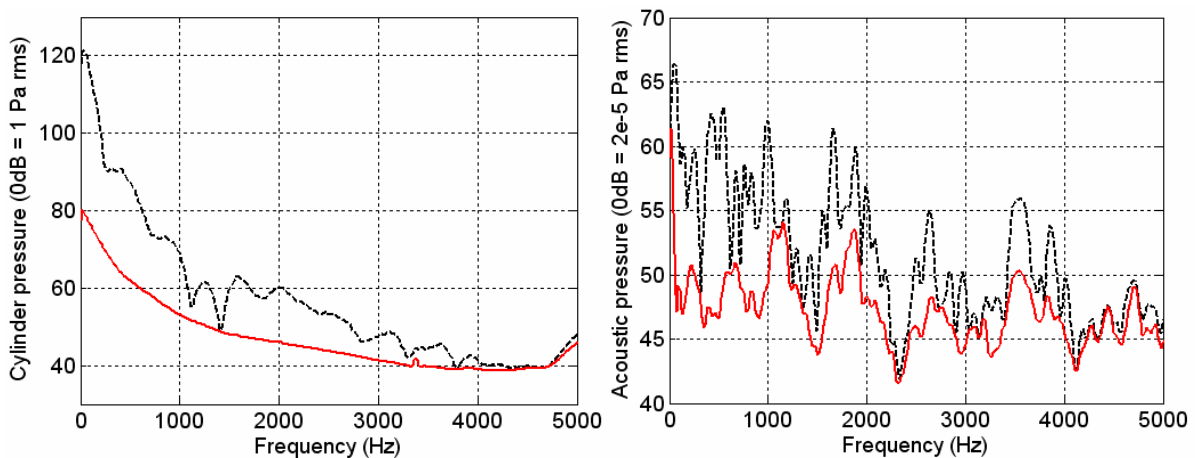


Figure 2. Operating point 1800rpm, 50% load. Left : cylinder #1 pressure spectra. Right : acoustic pressure spectra. Dashed black : whole signal. Red : random parts.

The deterministic-random part decomposition of signals recorded on a diesel engine are drawn in figure 2, and the coherence function and H1 estimate between the cylinder pressure and the output signal in figure 3.

The true spectrofilter is not known, so that indirect validations have to be implemented. The validity of spectrofilters has been assessed using two indicators :

- the stability of the spectrofilter to changes in operating conditions
- the causality of their corresponding impulse responses

As the spectrofilter represents the vibro-acoustic response of the engine block, it is expected to be relatively unchanged in different operating conditions (speed and load). For the same reason, the corresponding impulse response has to be causal.

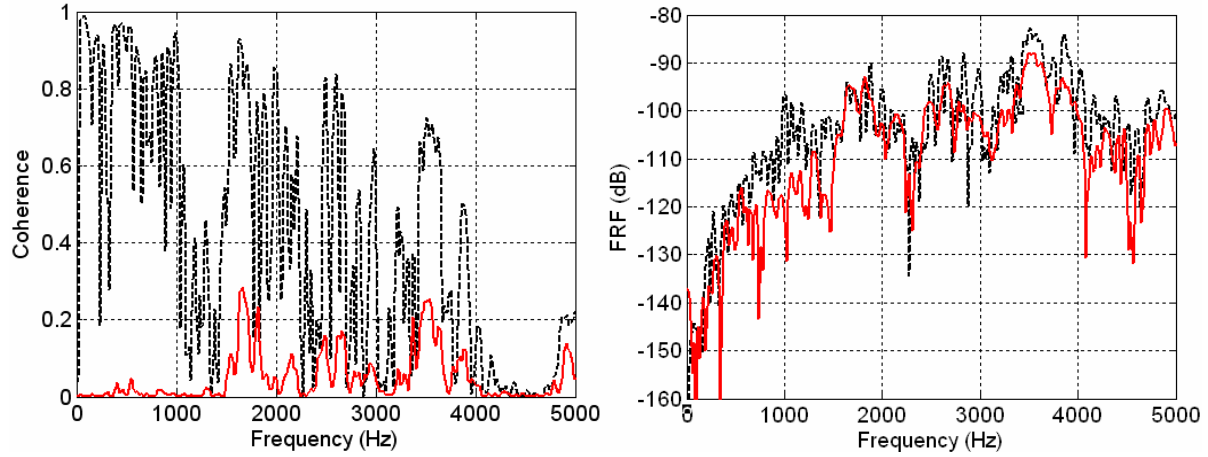


Figure 3. Operating point 1800rpm, 50% load. Left : coherence function. Right : H1 estimate of the FRF. Dashed black : from whole signals. Red : from random parts.

The uncertainty of the estimation of a transfer function is given by [7]

$$\varepsilon = \sqrt{\frac{1 - \gamma^2}{2n\gamma^2}} \quad , \quad (1)$$

where ε is the normalized standard deviation of the modulus of the estimator, γ^2 the coherence function between the input and the output, and n the number of averages used to estimate auto and cross spectra (equal to the number of engine cycles during the acquisition). As shown in figure 3, that the coherency is much lower between random parts than between whole signals. The uncertainty is thus higher for the H-estimate computed using random parts. This uncertainty is compared with the variability of different estimations of the spectrofilter with different loading conditions (160 operating points with different speed, torque and fuel types, [5]). The normalized standard deviation of estimated spectrofilters are drawn in figure 4 (left), together with the averaged theoretical uncertainty ε .

The difference between the observed variability and the theoretical uncertainty is very high for the spectrofilters estimated from the entire signals. It can be explained by the presence in the observed variability of a strong bias error induced by the coherency between cylinder pressures and mechanical sources. Concerning the spectrofilters estimated from random parts, the observed variability is much more in concordance with the theoretical uncertainty, especially between 400 and 1400 Hz. It means that in this frequency band the observed variability can be explained entirely by the uncertainty of the estimation, and consequently, that the removal of the deterministic part is efficient to unbiased the spectrofilter. Moreover, the variability of the spectrofilter using random parts is much smaller than the one from the whole signal. However, above 1500Hz, a significant variability is still observed even on the spectrofilter from random parts, that cannot be totally explained by the theoretical uncertainty.

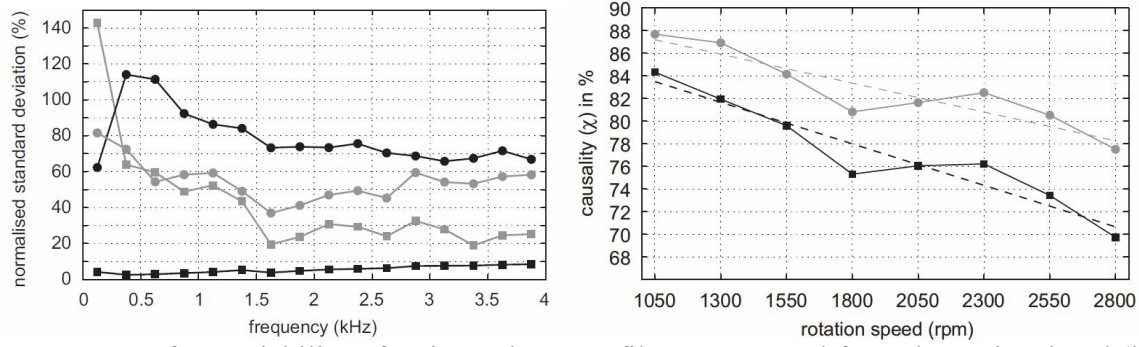


Figure 4. Left : Variability of estimated spectrofilters computed from the entire signal (black) and removing the deterministic parts (gray). Circular markers : observed variability. Square markers : theoretical uncertainty (ϵ). Right : causality of estimated spectrofilters with the entire signal (black) and removing the deterministic parts (gray). Dashed lines : linear regressions. *From [5]*

The causality of an impulse response will be defined here as the following ratio:

$$\kappa = \frac{\int_0^{+\infty} h(t)^2 dt}{\int_{-\infty}^{+\infty} h(t)^2 dt} \quad (2)$$

Where $h(t)$ is the inverse Fourier Transform of the estimated transfer function. This ratio is 100% if the impulse response is causal, and 50% if there is the same energy before and after $t=0$. The causality averaged over the operating points is drawn in figure 4, as a function of the rotation speed. The causality of spectrofilters estimated from random parts is higher than the one from the whole signal. Another observation is that the causality decreases with the rotation speed. That can be explained by the windowing operation that is worsened when the speed increases (mainly because of its shortening).

COMPARISON BETWEEN ESTIMATED SPECTROFILTERS AND IMPACT HAMMER TESTS

The use of operating conditions to measure transfer functions on the engine can be seen as some impact hammer tests : the combustion in the cylinder generates a high pressure rise that can really be compared to a hammer impact. The difference is that the window length is really short for operating measurements, inducing a spectral smoothing which can reduce the quality of results. It can thus be interesting to compare operating measurements to real impact hammer tests. Some FRF measurements have been carried out on the stopped (hot) engine using an impact hammer. The impact is vertically located on the cylinder head, very close to the axis of cylinder #1. Three transducers are considered, an accelerometer just next to the impact on the cylinder head (vertical), a second one on one of the bearing caps close to cylinder #1 (vertical), and a microphone above the engine. The 2 FRFs obtained for accelerometers using the impact hammer are drawn in figure 5, together with the spectrofilters estimated from random parts of signals for three different operating conditions. Spectrofilters are divided by the section of the cylinder, to get the same unit than FRF from impact hammer tests (i.e. $(m/s^2)/N$ for accelerometers, and Pa/N for the microphone).

A good agreement is observed for the accelerometer placed on the cylinder head, between 1 and 3kHz. For the bearing cap accelerometer, a strong peak is observed at 1.5kHz for the spectrofilters, that is not visible on impact hammer tests. It can be explained by the fact that the impact hammer excitation can not fully represent the combustion. In operation, the combustion process applies a force on the cylinder head (gas excitation path) but the same force in opposite direction to the piston, that is mainly transmitted to the bearings via the connecting rod and crankshaft (piston-crankshaft path). The strong peak at 1500Hz can thus be attributed to a modal behaviour of this path, as widely addressed in the literature [11-13].

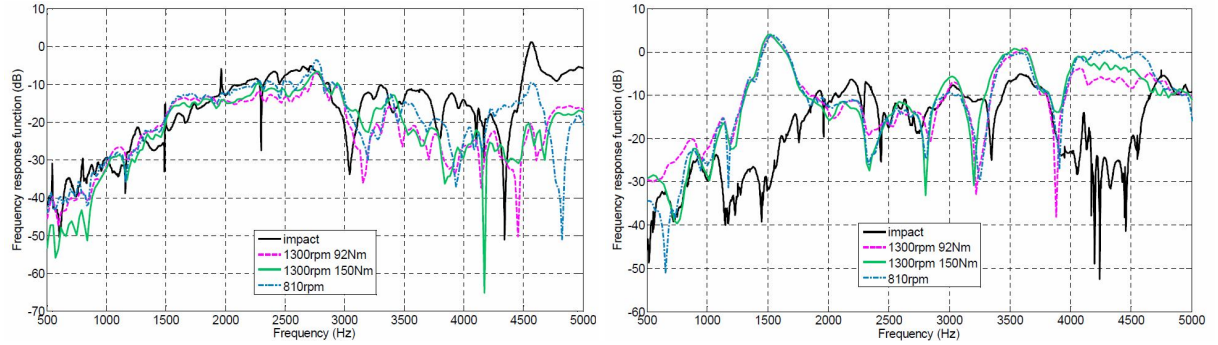


Figure 5. Left : FRF at the cylinder head accelerometer. Right : FRF at the bearing cap accelerometer. Black : FRF from impact hammer tests. Colours : spectrofilters obtained in operation. *From [10]*

The FRF at the microphone are drawn in figure 6. The FRF resulting from the impact hammer excitation is significantly higher (more than 10dB) than the operating spectrofilter in low and mid frequency (<1200Hz). This can be explained by the cancelling effect of gas excitation and piston-crankshaft paths of the combustion excitation in operation. Indeed, these two paths are exciting the engine block with the same magnitude, but in opposite direction. In higher frequency, the modal behaviour of the moving parts assembly and of the engine block eliminates this cancelling effect. A strong peak is observed between 1500 and 1800Hz in operation, that is not seen on impact hammer tests. This peak can be attributed to the modal behaviour of the piston-crankshaft path. Above 2kHz, it is interesting to note that both FRF have approximately the same magnitude.

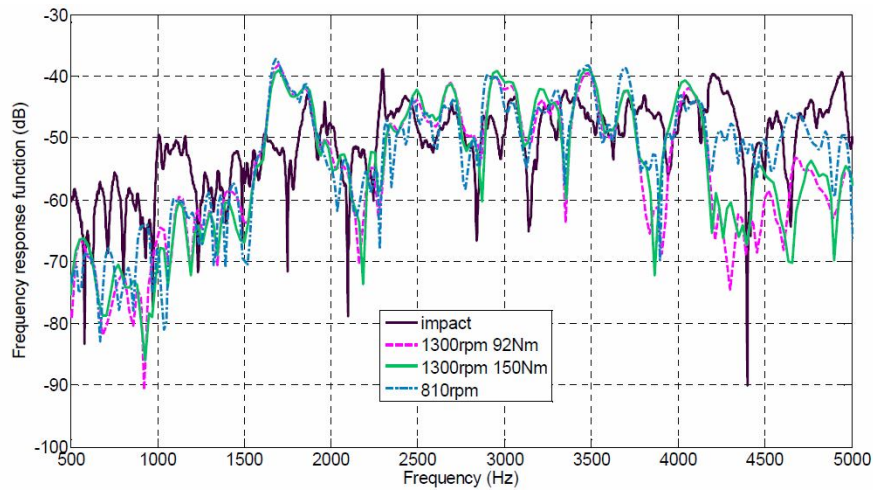


Figure 6. FRF for the microphone output signal. Black : FRF from impact hammer tests. Colours : spectrofilters obtained in operation. *From [10]*.

CONCLUSION

Extracting the combustion noise from overall noise is not an easy challenge, even if it is related to supervised source separation. The combustion noise can only be assessed by indirect methods, and consequently its definition is generally linked to the selected approach. In this work, several indirect validations have been carried out, based on the physical nature of identified filters, i.e. vibro-acoustic transfer functions depending mainly on the dynamic behaviour of the engine block. Future work will concern the application of experimental modal analysis to spectrofilters, applied to the synthesis of the combustion noise. A potential application concerns the implementation of listening test experiments, aiming to relate the combustion settings to some indicators of annoyance.

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