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Direct Sound Radiation Testing on a Mounted Car Engine

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ABSTRACT

For (benchmark) tests it is not only useful to study the acoustic performance of the whole vehicle, but also to assess separate components such as the engine. Reflections inside the engine bay bias the acoustic radiation estimated with sound pressure based solutions. Consequently, most current methods require dismounting the engine from the car and installing it in an anechoic room to measure the sound emitted. However, this process is laborious and hard to perform.

In this paper, two particle velocity based methods are proposed to characterize the sound radiated from an engine while it is still installed in the car. Particle velocity sensors are much less affected by reflections than sound pressure microphones when the measurements are performed near a radiating surface due to the particle velocity's vector nature, intrinsic dependency upon surface displacement and directivity of the sensor. Therefore, the engine does not have to be disassembled, which saves time and money. An array of special high temperature particle velocity probes is used to measure the radiation simultaneously at many positions near the engine of a compact class car. The particularities of these probes, the mountings used, and the actions taken to cope with disturbances such as airflows are described in this paper.

The effective sound pressure is calculated with a particle velocity based transfer path analysis method and a novel sound power based method. To validate these techniques, the data obtained is compared to the results acquired in an anechoic room with a dismounted engine. It is shown that similar results can be obtained with both methods but that the sound power based methodology seems more practical. It is a straightforward and fast approach to characterize the average sound pressure level at certain distance.

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INTRODUCTION

The radiated sound level is one of the main characteristics that defines the acoustic performance of a car engine. The standard procedure for benchmark testing involves dismounting the engine from the car and installing it in an anechoic chamber in order to capture the noise produced solely by the engine in free-field conditions. This procedure is effective, but very laborious and time-consuming.

For these reasons, two innovative particle velocity based approaches are proposed to measure the sound radiation from a compact class car engine while it is still installed in the car. These methods enable the calculation of the average sound pressure level at a certain distance from the engine.

Many sound sources and reflections inside the engine bay contribute to the sound pressure at a particular position. Even close to the radiating surface it is hard to distinguish the direct contribution from other disturbances. However, the normal particle velocity near the vibrating surface is hardly affected by background noise and reflections for three reasons [1]:

1. The particle velocity level, due to vibration of the surface itself, is high because of near field effects.
2. The particle velocity level due to background noise is usually low because many objects have high surface impedance. The incoming and reflected sound waves are nearly equal of strength but opposite in phase thus they interfere destructively.
3. Particle velocity sensors are directional and can be pointed towards the vibrating surface, reducing the noise contributions from other directions.

For a long time, convenient particle velocity sensors were lacking. In 1994, the Microflown sensor was invented, which can directly measure acoustic particle velocity in a small spot compared to the wavelength of audible frequencies [2, 3]. These particle velocity sensors can measure the source strength even in reverberant environments. Wind shields and probes withstanding high temperatures have been developed for the tests described in this paper. Dedicated mountings have also been developed to install probes around the car engine. The properties of these novel probes are shown in the following section.

Two methods are introduced based on near field particle velocity measurements to calculate the free field sound pressure radiated, as if the engine was dismounted from the vehicle and installed in an anechoic room. Firstly, a transfer path analysis approach is described involving particle velocity measurements in the engine bay and acoustic propagation paths acquired in an anechoic chamber. Secondly, particle velocity measurements are combined with a radiation impedance model to estimate the engine's sound power, and ultimately the emitted sound pressure at a given distance. The theoretical foundations of both methodologies are described below. Furthermore, the test setup is described and the results obtained are presented and discussed.

HIGH TEMPERATURE AND AIRFLOW PROBES

Probes and mountings have been developed specially for the test near the engine. High temperatures, high airflow speeds, and difficulties with mounting the sensors around the engine were the main problems tackled.

Two probe types were used. The probe in the Figure 1 (left) can be used up to 250°C. The engine surface itself could be even hotter because the temperature at the probe position is often lower than the engine's temperature. The maximum airflow rating of these probes is 2 m/s. They were installed with special mountings that were glued on the engine surface. Figure 1 (right) shows the suspension that de-couples the probe from the surface. Vibrations are highly damped above 40 Hz due its low resonance frequency of about 15 Hz.

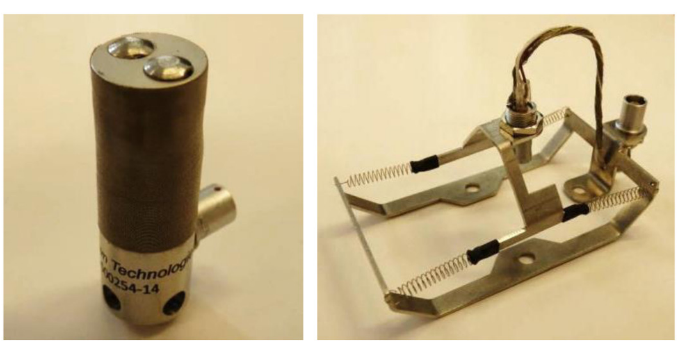

Figure 1. Left: high temperature particle velocity probe. Right: high temperature probe damper.

Probe dampers cannot be used for all measurement positions. Some locations have airflow speed exceeding 2 m/s or are difficult to reach because of rotating parts. Therefore, different high temperature probes with a straight plug at the bottom were developed, see Figure 2. The probe can withstand a temperature of 250°C and the gooseneck to mount the probe a temperature of 180°C. The larger, one inch diameter, wind cap can endure a maximum airflow speed of 7 m/s.

Figure 2. High temperature probe with one inch diameter wind cap (left) installed using a gooseneck (right).

Although the probes are equipped with wind shields, there are positions where the air flow is still too high. Accordingly, several additional measures were taken. The cooling fan of the car engine and the cooling fans positioned in front of the car were turned off during the tests. In addition, the bottom of the car was partly covered with a sheet of metal mesh reducing the (turbulent) airflow generated by the interaction of the roller bench rolls and the car tires. The type of mesh used hardly has an acoustic impact, thus the sound field can be assumed to remain unchanged. Furthermore, metal mesh wind shields have been placed around sensor positions with high airflows caused by the rotating engine parts.

FREE-FIELD SOUND PRESSURE ESTIMATION METHODS

Two methods involving particle velocity near the radiating object are proposed to calculate the sound pressure emitted by the engine. The fundamentals of a transfer path analysis approach (TPA) and a sound power method are described in the following subsections.

Particle Velocity based Transfer Path Analysis

When an object (in our case the engine) with a surface area *S* radiates sound under operating conditions, an infinitesimal small area M at a distance can be defined to determine how different sub-areas of *S* contribute to that point. Figure 3 shows a sketch of the scenario.

The theoretical derivation for calculating the sound pressure contribution follows Hald and Kinsler [4, 5]. Two test conditions are defined: 1) a reciprocal transfer function test with monopole source at *M* exciting the sound field, and 2) a noise radiation test with surface *S* producing noise while the monopole is turned off. Two sets of variables are used depending on the measurement conditions. p_{TF} and u_{TF} are the sound pressure and particle velocity during the reciprocal transfer function test, p_n and u_n are the sound pressure and particle velocity during the noise test. According to the principle of acoustic reciprocity,

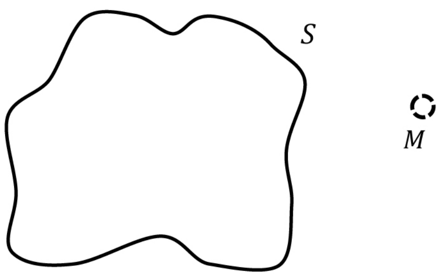

Figure 3. Schematic overview of the geometry involved in the derivation

$$\int_{\boldsymbol{M}} (p_{TF}u_n - p_n u_{TF}) d\boldsymbol{M} + \int_{\boldsymbol{S}} (p_{TF}u_n - p_n u_{TF}) d\boldsymbol{S} = 0 \tag{1}$$

For the noise test, the particle velocity integral u_n is zero across surface *M* because the net energy remains unchanged in *M*. Reference sound pressure p_r can be obtained by integrating sound pressure p_n over *M*. For the transfer function test, the integration of particle velocity over *M* gives the volume velocity of monopole source *Q*, leading to

$$-p_r Q + \int_{\mathbf{S}} (p_{TF}u_n - p_n u_{TF}) \mathrm{d}\mathbf{S} = 0 \tag{2}$$

In our case *S* can be assumed acoustically rigid, implying that the normal particle velocity very close to the engine is nearly zero during the transfer path test. Consequently, equation 2 can be simplified to

$$p_{r=} \int_{\mathbf{S}} \frac{p_{TF}}{Q} u_n \mathrm{d}\mathbf{S} \tag{3}$$

Equation 3 is the basic simplified equation of most velocity-based transfer path analysis and panel noise contribution methods [6]. It relates the sound pressure at the reference position p_r to the normal particle velocity u_n of the surface object *S* and the acoustic transfer function p_{TF}/Q. As shown in Equation 3, particle velocity characterizes the sound source excitation whereas the transfer function represents the attenuation of sound traveling from the emitting point to the receiver. Therefore, transfer path tests might also be used for other engines with a similar shape since the propagation in the free field merely depends on the geometry of the measurement scenario. This should enable the combination of particle velocity tests in the engine bay with “standard” free-field transfer functions for the estimation of the sound pressure emitted.

Sound Power Method for the Estimation of Free-Field Sound Pressure

A transfer path measurement is not always feasible due to hardware or time limitations. Therefore, an alternative method is suggested based on sound power estimations, avoiding traditional constrains of TPA and other complex testing methods. Sound power is commonly used as a quantitative description of the acoustic output of a device [7]. However, this quantity is fairly influenced by the test environment since it depends on sound pressure. Sound power ∏ is calculated by integrating the normal intensity over the radiating noise surface, i.e.

$$\Pi = \int_S I_n \, dS \tag{4}$$

where I_n is the active normal intensity given by [7, 8]:

$$I_n = \langle p \, u_n \rangle_t = \frac{1}{2} \mathrm{Re}\{p \, u_n{}^*\} \tag{5}$$

where *p* is sound pressure, u_n is the normal particle velocity and (.*) denotes complex conjugation. Sound pressure is affected by the measurement conditions, especially in a reverberant environment in the presence of background noise. As mentioned above, particle velocity in the vicinity of the test object is hardly affected by these disturbances. Alternatively, intensity can be expressed as a function of particle velocity and the specific acoustic impedance, i.e.

$$I_n = \frac{1}{2} |u_n|^2 \mathrm{Re}\{Z_n\} \tag{6}$$

where Z_n is the ratio between the sound pressure and normal particle velocity at the same point. To calculate the normal acoustic intensity, and thus the sound power of the engine, particle velocity measurements are combined with the free-field acoustic impedance. Two models are evaluated regarding plane (Z_{plane}) and spherical (Z_{sphere}) wave propagation

$$Z_{plane} = \rho_0 c \tag{7}$$

$$Z_{sphere} = \frac{ikr}{ikr + 1} \rho_0 c \tag{8}$$

where *k* is the wavenumber, ρ_0 is the density of air, *c* is the sound speed and *r* is the distance to the excitation point. Substituting Equation 7 and 8 into Equation 6 leads to

$$I_{plane} = \frac{1}{2}|u_n|^2 \rho_0 c \tag{9}$$

$$I_{sphere} = \frac{1}{2}|u_n|^2 \left(\frac{k^2 r^2}{k^2 r^2 + 1}\right) \rho_0 c \tag{10}$$

The ultimate objective is to predict the sound pressure at a given distance from the engine. Since sound power is an estimate of the source radiation, the spatially averaged sound pressure perceived at a certain distance can be obtained from their relationship. Provided that the energy attenuation caused by the air is insignificant, the sound power of a surface area S_1 surrounding the engine equals the sound power of a larger imaginary square box with a surface area S_2, thus

$$\Pi = \int_{S_1} I_n \, dS_1 = \int_{S_2} I_n \, dS_2 \tag{11}$$

Figure 4 shows a sketch of the situation. The above equation considers that the total acoustic energy remains constant but that the acoustic level at a position further away will decrease because the energy spreads over a larger area.

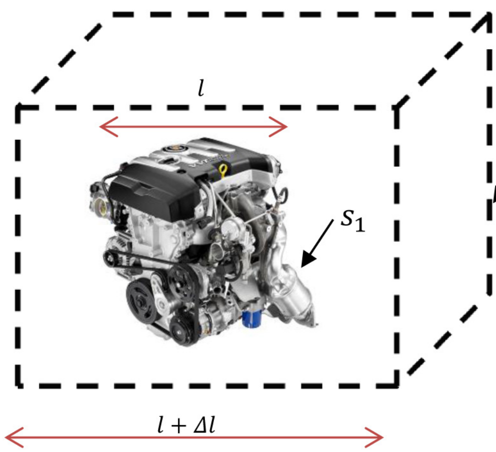

Figure 4. Areas involved in the calculation.

The spatially averaged sound intensity $\langle I_n \rangle_S$ can conveniently be expressed as a summation of *M* discrete positions,

$$\langle I_n \rangle_S = \frac{1}{M} \sum_{m=1}^{M} I_n(\vec{x}_m) \tag{12}$$

A discretized version of Equation 11 combined with Equation 12 yields an expression relating the spatially averaged intensity in the near-field $\langle I_n \rangle S_1$ and an area further away, i.e.

$$\Pi = A_m \sum_{m=1}^{M} I_n(\vec{x}_m) = S_1 \langle I_n \rangle_{S_1} = S_2 \langle I_n \rangle_{S_2} \tag{13}$$

where A_m is the average size of a patch in which the intensity is measured. Equation 13 enables the calculation of the spatially averaged intensity at certain distance from the source based on sound power. It is then required to establish the relationship between the averaged acoustic intensity and sound pressure. Since sound pressure level of a plane wave equals the sound intensity level in a free field, Equation 13 can be re-arranged and expressed in logarithmic form such as

$$\langle L_p \rangle_{S_2} = \langle L_i \rangle_{S_2} = 10 \log_{10}\left(\frac{S_1 \langle I_n \rangle_{S_1}}{S_2 I_{ref}}\right) = L_w + 10 \log_{10}\left(\frac{1}{S_2}\right) \tag{14}$$

where $\langle L_p \rangle S_2$ and $\langle L_i \rangle S_2$ are the sound pressure and intensity levels, respectively, spatially averaged around area S_2; and L_w is the sound power level of the source.

ROLLER BENCH TEST DESCRIPTION

Thirty eight particle velocity sensors were distributed at all sides of the engine to ensure that all sound radiated is captured, see Figure 5. Twenty-one transducers mounted in probe dampers (see Figure 1) were attached to the surface of the engine and the gearbox using glue and tape. The remaining seventeen sensors were mounted near the engine with goosenecks. These goosenecks were attached to a grid of steel pipes, which was supported on both sides of the car. The total time taken for the test, including set-up and data acquisition, was approximately one day.

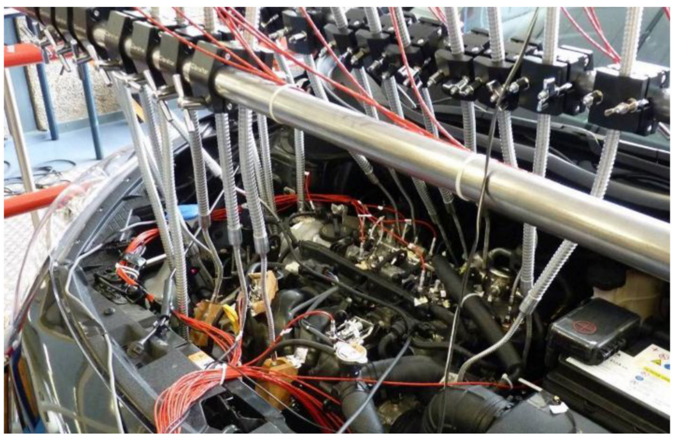

Figure 5. Test setup overview.

The tests have been performed on a roller bench in third gear. Different engine conditions have been used, i.e.: idle conditions, several fixed rotational speeds, and engine run-ups. Results presented in this paper will be focus on the test performed with an engine regime of 5000 RPM. Several details are omitted due to confidentiality agreements.

EXPERIMENTAL RESULTS

Results of two different calculation methods are evaluated in this section. But firstly the quality of the particle velocity measurements is assessed due to its crucial importance for both methodologies. The data is arranged into seven groups depending on the sensor position during the tests. Figure 6 shows the particle velocity signals acquired with a dynamic range of 100 dB. The sensor signals are only affected by airflow induced noise below 125Hz, which can be seen as an increase of the self-noise level especially at the front part of the engine. Consequently, the sound pressure is predicted only in the frequency band between 125 Hz and 7 kHz, where the particle velocity measurements achieved a good signal-to-noise ratio.

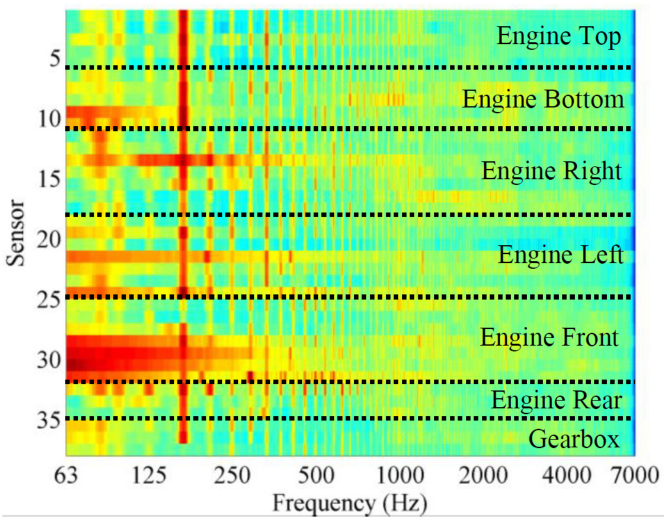

Figure 6. Particle velocity spectra in the vicinity of the engine surface

Transfer Path Analysis Test

The airborne TPA approach requires the propagation paths from the engine to a reference position. For this study, reciprocal measurements with a monopole source and the engine dismounted from the car body were performed in an anechoic chamber. Figure 7 shows the transfer function spectra acquired displaying the same color for signals belonging to the same group. A similar trend can be seen for all positions, indicating that probably theoretical functions of a simplified geometry could be used to simulate the acoustic transfer paths.

The validity and limitations of a sound pressure estimation method can be evaluated straightforwardly, by comparing the measured and estimated sound pressure spectra. Figure 8 shows the results obtained with the transfer path analysis procedure using two different approaches: direct summation of the individual contributions in the time domain (phase-matched), and superposition of individual sound pressure contribution spectra, disregarding the phase relationship between signals. Both approaches should yield the same answer if the engine noise can be regarded as a set of uncorrelated noise sources. For the case studied, this assumption is met for most frequencies. The measured and synthesized sound pressures are in good agreement between 200 Hz up to 4 kHz. Errors are found beyond this frequency range. For instance, there is a discrepancy of almost 10 dB between the results of both models at 165 Hz, and, the sound pressure is overestimated above 4 kHz upwards limiting the usable range of the TPA method at high frequencies.

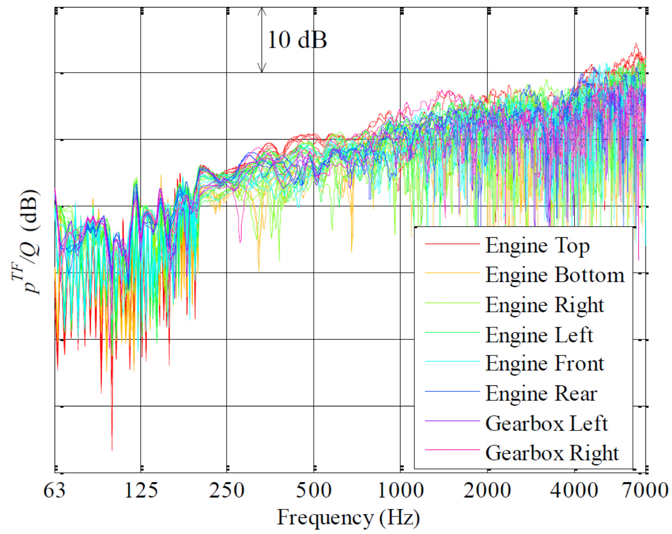

Figure 7. Transfer paths from a monopole source to different measurement positions.

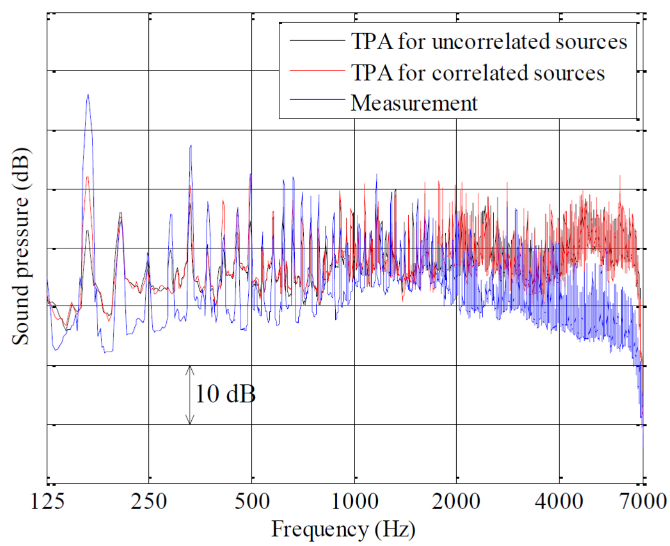

Figure 8. Comparison between the directly measured and the estimated sound pressure using transfer path analysis methods

Sound Power Method

The free-field acoustic impedance at the measurement spot is required to estimate the sound intensity and, ultimately, the sound power. Previously, two models have been described for plane and spherical waves. Figure 9 shows the results obtained using Equation 9 and Equation 10 for a surface distance of 0.02 meters. At high frequencies the intensity estimations are similar, but discrepancies occur at low frequencies because only the spherical wave model involves a frequency dependent correction, which affects longer wavelengths and larger surface distances more strongly.

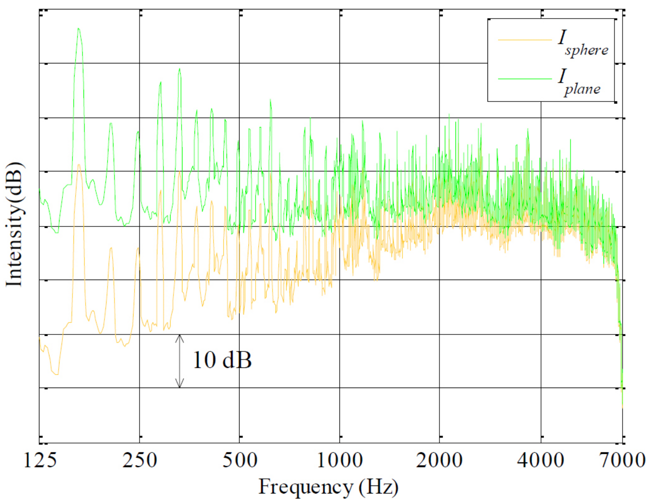

Figure 9. Sound intensity calculated using different impedance models

The summed intensity spectra around the engine multiplied by the average surface area covered by each sensor position yields the estimated acoustic sound power. The spatially averaged sound pressure emitted can be then computed using Equation 13. In the case studied, sound pressure was measured in an anechoic chamber at four positions located one meter away from the engine. Figure 10 presents a comparison of the two sound power estimations with the average pressure measured. As can be seen, the plane wave impedance model overestimates the sound pressure radiated below 300 Hz, but accurately describes the noise emitted at mid and high frequencies. In contrast, there are significant discrepancies with the spherical wave impedance model. Hence, the plane wave impedance model seems to describe best the free-field acoustic impedance at 0.02 meters from the engine surface.

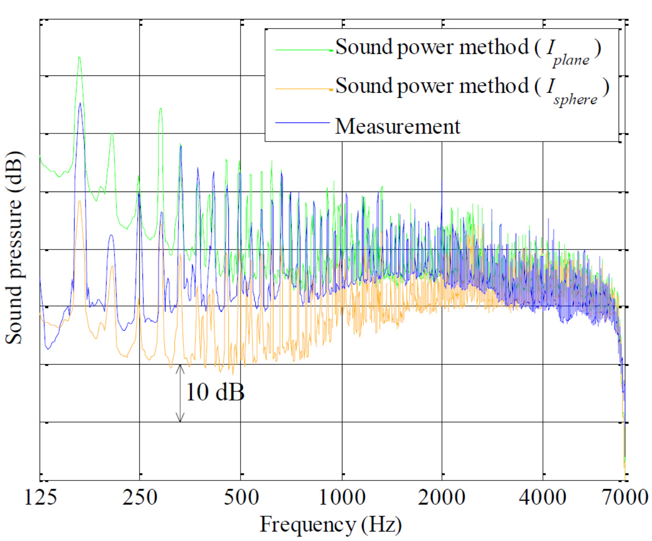

Figure 10. Comparison between measured and estimated sound pressure using sound power methods

Comparison of TPA and Sound Power Methods

Two different approaches have been proposed to predict the sound pressure at a certain distance from a car engine: a transfer path and a sound power method. The first technique is used to estimate the sound pressure at a particular spot whilst the latter is used to calculate the spatially averaged sound pressure received at certain distance from the sound source. The outcomes of both methods can be similar if results of TPA measurements at several positions equidistant from the source are averaged, obtaining an estimation of the overall noise emitted by the car's engine.

The TPA method has been evaluated with a superposition of individual sound pressure contribution in the time domain, and with a summation of the power spectra. Although the latter approach is suitable only for uncorrelated signals, the good agreement with the time domain calculation method shows that asynchronous data acquisition can yield acceptable estimations in the mid and high frequency range. Since the superposition of sound pressure contributions in the time domain is not limited by the nature of the source, results of this approach are presented in this section.

The sound power method introduced combines particle velocity measurements performed in the engine bay with an impedance model to estimate the sound power as if it were measured in free-field conditions. For the experimental case evaluated the outcome of the plane wave impedance model has been found more suitable, implying that proper results are obtained when the particle velocity and sound intensity spectral level are assumed equal. Notably, this assumption does not seem to hold for frequencies below 300 Hz. Free-field impedance measurements could be performed in future tests to improve the accuracy of the particle-velocity based sound power method, especially at lower frequencies.

Figure 11 shows a comparison of the sound pressure estimated with a TPA method averaged at four positions at one meter from the engine, and the sound power method with a plane wave impedance model. The results of the two methods are similar between 300 Hz and 4 kHz, and both are in good agreement with the measured sound pressure performed in the anechoic chamber with the dismounted engine.

The discrepancy of the TPA method at high frequencies could be caused by errors in the volume velocity estimation from the monopole sound source, a low spatial density of particle velocity measurements or the assumption that the engine surface is rigid no longer hold at such high frequencies.

Although satisfactory results are obtained with both approaches, the sound power method has several advantages over the TPA method. A transfer path analysis methodology is more involved in practice: it is a two steps measurement process, it requires measuring the source under operational conditions and measuring the acoustic transfer path using a monopole source, which should be performed for several locations to compute the overall acoustic performance of the engine. In addition, the assumption that the engine can be regarded as a rigid surface might not hold for high frequencies.

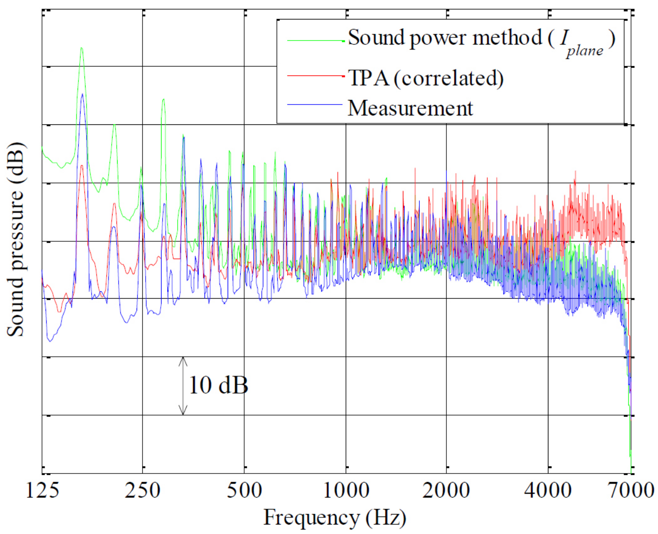

Figure 11. Comparison between measured and estimated sound pressure using both sound power and TPA methods

In summary, results of both measurement methods evaluated in this paper are in good agreement with the measured sound pressure; however, the sound power method seems most convenient since reciprocal transfer function tests are not required.

CONCLUSIONS

Currently, engines have to be disassembled from the car for benchmark tests, which is a rather complicated procedure. A novel measurement methodology has been introduced to characterize the sound radiated from the engine while it is still mounted in the car. The particle velocity has been measured at 38 locations near the engine. Special mountings and probes suitable for high temperatures and airflows have been used. Additional measures were taken to reduce the airflow; windshields have been placed at the bottom of the car and at dedicated positions, and the cooling fans of the car were turned off during the test. Low frequency limitations were imposed by the wind-induced noise of the particle velocity data acquired.

Two methods based on particle velocity have been introduced to calculate the sound radiated from an engine. The first method combines particle velocity measurements with acoustic transfer paths previously acquired in an anechoic room. The second method estimates sound power using the measured particle velocity together with an acoustic impedance model. Both methods have been compared to a direct sound pressure measurement in an anechoic room. The results achieved are fairly similar, therefore validating both *in-situ* measurement procedures to estimate the noise emission of a complex noise source.

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