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Inverse transfer path analysis, a different approach to shorten time in NVH assessments

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ABSTRACT

This paper presents the design and implementation of a simplified method, based on the transmissibility concept, for a noise path assessment, which allows a rapid and accurate analysis. The Inverse Transfer Path Analysis aims to assess, and determine, the contribution of the critical paths, which are transmitting structure-borne noises and vibrations, from the vehicle's vibration sources to the driver's ear.

The cabin noise transfer function, from the involved attachment points and directions, can be simultaneously obtained by applying an impulsive noise source inside the cabin. This approach avoids the use of other time consuming classic procedures.

The proposed methodology includes two types of tests, static condition tests in a semi-anechoic chamber and operational tests on a roller bench. The results assessment comprises the analysis of the noise contribution of each path, depending on the frequency and the vehicle speed range.

This publication introduces a novel NVH method proposed to study and identify noise transfer paths in a car structure. The theoretical approach of the methodology, practical implementation, and obtained results, are described in this work, as well as a methodology validation, to evidence the suitability of the proposed method.

1. Introduction

The increasing comfort expectations of customers, in the automotive sector, together with the need to meet the noise regulations, are a continuous challenge for car manufacturers. Noise, Vibration and Harshness (NVH) is one of the main areas related to the customer perception of quality in vehicles [1,2]. Therefore, the interest in the field of noise and vibrations improvement has been intensified over time.

Short development times, and noise problems, which often arise late in the vehicle development schedule, are becoming a constant challenge in terms of acoustic assessment for vehicle manufacturers [3–7]. In the highly competitive automotive market, the ability to perform a fast NVH assessment on a new design has become essential for any manufacturer. Therefore, a fast and effective noise analysis tool would be very convenient to analyse the noise and vibrational behaviour of the vehicles.

The sound radiated inside a vehicle is always produced by diverse sources, which are the active part of the system. Structure-borne and Airborne are the two noise canals transmitting noise to the receiver, or passive part. There are some different paths through which noise is spread from sources to receivers inside the cabin. Interior noise received for the car occupants is the total of all paths' contributions, see Fig. 1, and this noise is the reference to determine the comfort level offered to the passengers of a vehicle [8,9].

The method presented in this paper is focused on the body car structural noise transmission. Therefore, only the structure-borne noise, introduced to the cabin by the motor mounts and the suspension points, has been taking into account in this work.

Low-middle frequency noise is the main factor that contributes to the comfort on cars [2]. Not only it is referred to the noise, but also to vibrations around 20 Hz, which cannot be heard but can be felt as well. These low frequencies are responsible for most of the discomfort, or harshness, felt by the occupants.

Interior noise assessment allows identifying the amplitude of the partial noise contribution, from each path, within the range of required frequencies. This knowledge is essential for the improvement of the acoustic and vibrational behaviour for each specific component, and indeed for the whole vehicle.

It is possible to find in the literature several approaches to define passive transfer paths by which vibrations, or acoustic signals, propagate through the vehicle [5]. The most recognised

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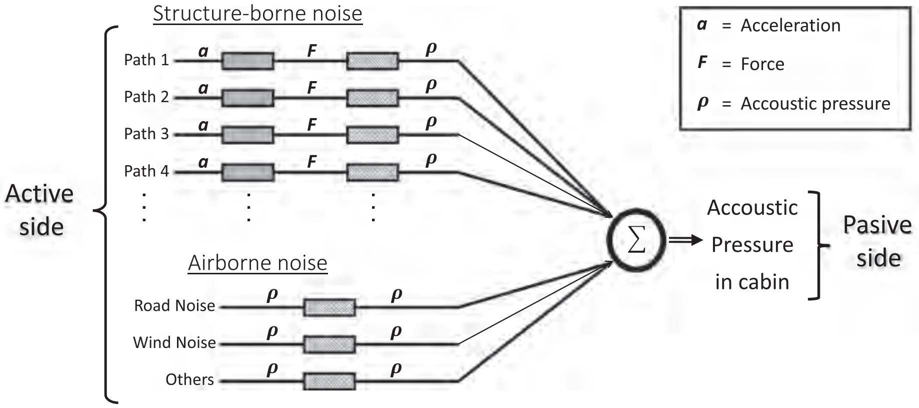


Fig. 1. Noise transfer behaviour in a car.

method to determine the relationship between the noise generated by the sources, and how is perceived by the receiver, is the Transfer Path Analysis (TPA), which was developed in the early '80s [9]. TPA was seen as a tool to improve the NVH performance of systems with complex structures like cars, aircrafts or ships.

Since the beginning of the use of TPA, different methods have been developed, with the aim of offering an alternative to reduce its complexity and long-time procedure. This is a widely used and very accurate method, but due to its complexity and long execution time, it is not always the most suitable one.

The application of transmissibility concept, in Operational Paths Analysis (OPA) methods, offers a different solution by a significant reduction in complexity and measurement time. Nonetheless, the current approach of this methodology has some requirements which limit its applicability, like the mount stiffness's identification [10], or the necessity of a number of known forces acting at the connection points during operation [11]. There is extremely complicated to correctly excite, or to measure, these connection points during the test, if not impossible.

1.1. Transfer paths analysis

In TPA, the received acoustic pressure ($\rho^{\mathrm{TPA}}_{(\boldsymbol{\omega})}$) is considered as a superposition of individual contributions from each transfer path ($\rho_{\mathrm{i}\,(\boldsymbol{\omega})}$). Each individual path pressure is calculated as the product of the frequency response function ($\mathrm{FRF}_{\mathrm{i}\,(\boldsymbol{\omega})}$) by the operational force ($f_{\mathrm{i}\,(\boldsymbol{\omega})}$) applied to that path, as can be seen in Eq. (1). Therefore, it is necessary to determine the FRF and the operational force to assess each noise path contribution [12].

$$\rho^{TPA}(\omega) = \sum \rho_i(\omega) = \sum FRF_i(\omega) \cdot f_i(\omega) \tag{1}$$

TPA test procedure requires two basic steps; Static and dynamic tests. Static test consists in the estimation of the frequency response function ($\mathrm{FRF_i}$) from experimental static tests (e.g. impact hammer test, shaker test, etc.). Dynamic test, or Operational tests, requires to drive the vehicle during an acceleration, or deceleration, (e.g. run-up, run-down, etc.) on the road or on a chassis dynamometer, to introduce operational loads (f_{i}) and record the corresponding acoustic pressure (ρ).

Despite the TPA is a widely used and a very accurate method, its complexity and time-consuming procedure has motivated the study of variations in the method [13,14], or the development of simpler and faster noise path analysis methods or techniques. Difficult access to crucial measurement locations, to obtain FRFs, causes either high instrumentation and measurement effort, and that means more consumed time.

1.2. Operational paths analysis

The application of transmissibility concept in OPA based methods is the main alternative to the TPA, and it offers a different solution by a significant reduction in complexity and measurement time.

OPA methods apply transmissibility in a direct way, making an in-situ estimation of the frequency response function of the system. Through a transmissibility matrix ($\mathrm{T}_{i\,(\boldsymbol{\omega})}$) between sound pressure ($\rho^{\mathrm{OPA}}_{(\boldsymbol{\omega})}$) and operational accelerations ($a_{\mathrm{i}\,(\boldsymbol{\omega})}$), eliminates the need for time consuming FRF measurements as only operational data is needed for the analysis [15–17].

$$\rho^{OPA}(\omega) = \sum \mathrm{T}_i(\omega) \cdot a_i(\omega) \tag{2}$$

The validation of the results is considered one of the limitations of this methodology [15,9]. Data processing flow in OPA methods actually corresponds to a "backward–forward" calculation because the same velocity data are used twice, thus the validation of the results is not so direct here like in TPA methodology [10,9,18]. This means that, on contrary to the TPA approach, where the comparison of the calculated and the measured target is a widely accepted tool for assessing the validity of the results, validation by synthesis does not make sense in the OPA method [11,12]. Therefore, here the assessment usually comes from the comparison of results with other methods [15,19].

1.3. Limitations of current NVH approaches

Despite that many diverse approaches, which can be taken to solve the matter, the existence of a method which combines speed and accuracy continues being a necessity in most cases. The current situation implies that, when an NVH assessment is required, noise engineers must decide between accuracy or execution speed.

If the implementation of the TPA is decided, it must be presumed that an exhaustive work, and a considerable time, will be required to perform it correctly. This option is usually preferred in academic studies where the time is not a crucial factor. Implementation of TPA requires disassembly of components at some stages and a meticulous performing.

TPA bigger handicap is that a long implementation time is considered a disadvantage nowadays, as developing times in the industry are becoming shorter each time. Only the TPA impact hammer test stage, as a reference for a car assessment, takes weeks of work for several highly skilled researchers, bringing the TPA time preforming between one to several months, when the whole OPA procedure do not use to take more than several days.

Where the OPA solution is preferred the results come very quickly, as it can be performed without removing parts, and not special training is required. The problem here is that sometimes this way does not prove to be conclusive enough [15,9]. Finally, the noise assessment can take even more time, as additional tests are necessary using different methods, before being in appropriate position to deliver a suitable conclusion.

There is a catalogue of methods following both approaches [20–23], but the problem of having to choose between speed or accuracy is still present. This gap between the methods that seek efficiency, and those that seek speed of implementation, is the reason for developing the work described here. This paper aims to define a new method that can provide some improvement in the NVH field for industrial applications, where a commitment between good accuracy and high speed of implementation is required.

The main target of this paper is to describe a new simplified method to provide a basis for rapid NVH assessments. The methodology aims to study the noise transfers paths contributions without disassembling the system. This avoids using time consuming procedures to determine paths contributions without affecting the accuracy of the results.

2. Methodology

The main goal of this method, called Inverse Transfer Path Analysis (ITPA), is to quantify the contribution of each one of the transfer paths, through which noise and vibrations are transmitted to the receiver. This is achieved by determining the relationship between the inputs, generated by the sources, and the output received by the receptor.

In this research, only the structure borne noise, introduced to the cabin from the motor mounts and the rear suspension points, has been considered. Thus, a variation between measured and computed pressure levels will be present in the results, due to the omitted air borne contribution.

The absence of the air borne contribution can be assumed as it does not modify the behaviour of the structural paths. It means that the partial contribution of each structural path still can be determined, main goal of ITPA. Calculated, or computed, acoustic pressure will contain the same orders, or harmonics, with the same shape than measured acoustic pressure.

Two main parts can be found in the system to be analysed, an active and a passive part. Active part generates the vibration and noise (source), while passive part contains the means that transmit these vibrations (paths) and the objects/people that absorb these fluctuations (receiver). According to [9], these three elements can be defined, as observed in Fig. 2:

- Source: internal DoFs (degree of freedom) belonging to the active components that cause the operational excitation but are unmeasurable in practice (electric engine and shaft rolling effects).
- Interface/passive paths: coupling DoFs residing on the interface between the active and the passive components (chassis and bodywork).
- Receiver: response DoFs at locations of interest on the passive component, possibly including acoustic pressures and other physical quantities (driver).

Since this method does not require removing the engine, or any other element of the car, the cross-coupling effects introduced when the engine is mounted are always considered.

ITPA implementation methodology is composed of three main stages: Stage 1- Static Test, to determine the noise transfer function; Stage 2- Operational Test, to acquire accelerations and sound pressure in dynamic conditions; and Stage 3- Calculations and results.

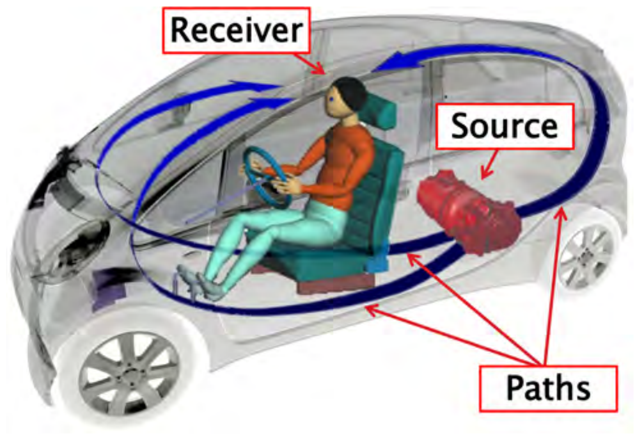


Fig. 2. ITPA source-path-receiver system example.

During static test in Stage 1, transmissibility concept is applied in an inverse way, to avoid the FRF estimation. It is based in the concept of mechanical-acoustic reciprocal transmissibility, which has been proved accurate enough in low and medium frequencies [24–26,19]. This approach of vibro-acoustic reciprocity has been often utilized in the measurement of Noise Transfer Functions (NTF), and widely demonstrated over time in numerous publications [27,28].

In the operational test, Stage 2, inputs will be accelerations and the output will be the acoustic pressure at the driver's head position, as only the influence on the cabin from structure-borne paths will be considered. Since static tests are required to characterise the behaviour of each path, the model performance is preserving the TPA structure. Therefore, the validation of the model can be determined by the comparison between synthesized and measured target acoustic pressure.

Calculation of the acting forces over the system are not required with this method. As static and operational tests both work with accelerations and acoustic pressure, it is not only unnecessary, but also redundant, to calculate forces from accelerations in each case. As ITPA does not require to know the operational forces, acting during the test, main limitations of indirect OPA methods, or the TPA itself, have been left behind. This is a relevant difference because in practice, knowing the operating forces with accuracy enough, is a really difficult task, if not impossible.

2.1. Stage 1: static test

Noise transfer functions are calculated here in an inverse way to the natural working mode of the dynamic physical system. Transfer functions are determined by applying an impulsive noise source inside the cabin, at the receiver point, and registering the produced accelerations at the end of main transfer noise paths, at the engine and suspension points (Fig. 3). Therefore, an impulsive sound source with adequate omni-directional characteristics at near field, and powerful enough to excite the body car structure, will be required [29].

Thus, this stage consists in performing static tests, introducing an acoustic pressure ($\rho_{st\_j(t)}$) inside the car, near to the driver position. Then, this acoustic pressure, and the corresponding produced accelerations ($a_{st\_ji(t)}$) at the defined points, are registered.

$$\rho_{st}(t) = \begin{bmatrix} \rho_{st\_1(t)} \\ \vdots \\ \rho_{st\_j(t)} \end{bmatrix} \tag{3}$$

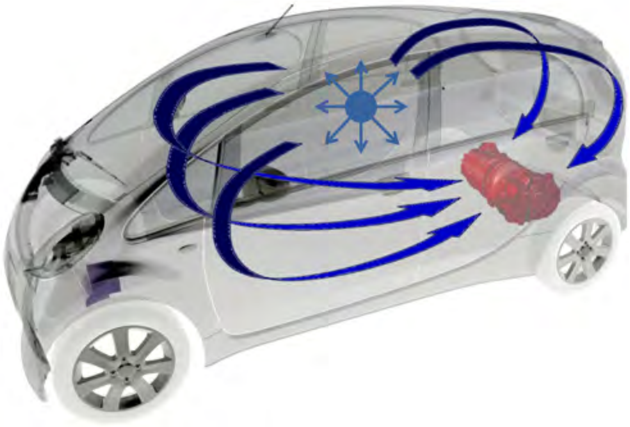


Fig. 3. Static tests system.

$$A_{st}(t) = \begin{bmatrix} a_{st_{11(t)}} & \cdots & a_{st_{1i(t)}} \\ \vdots & \ddots & \vdots \\ a_{st_{j1(t)}} & \cdots & a_{st_{ji(t)}} \end{bmatrix} \tag{4}$$

where the subscript “*st*” refers to a static test, “*j*” denotes the impulsive noise test number, “*i*” is the path number and (t) indicates time domain. As the inverse matrix method has been used, in order to define noise transfer functions, a number of tests equal or greater than the number of paths is necessary to define a square accelerations matrix $(A_{st(t)})$.

By using the Short-Time Fourier Transform (STFT), accelerations and acoustic pressures data are transformed to the frequency domain, $A_{st}\ (\omega)$ and $\rho_{st}\ (\omega)$. Then the acoustic pressure-to-accelerations relationship can be defined as the noise transfer function $(H_{st(\omega)})$ as the result of the Eq. (5).

$$H_{st}(\omega) = \begin{bmatrix} h_1(\omega) \\ \vdots \\ h_n(\omega) \end{bmatrix} = A_{st}(\omega)^{-1} \cdot \rho_{st}(\omega) \tag{5}$$

This transfer function contains the acoustic behaviour information of each noise transfer path, depending of the frequency, where n is the total number of paths involved in the test. It will be used later to calculate the estimated sound pressure of each path, from the recorded acceleration in the dynamic test.

2.2. Stage 2: operational test

The test performing, and data acquisition, in an ITPA operational test does not differ from the rest of methods. But the post processing to obtain the final results, is more direct here as operational forces do not need to be calculated. This direct mathematical approach can be a limitation of this method if any of the relevant path is missing, as the transfer function will try to minimize any frequencies non defined in the physical system of the static test.

Operational measures consist in a run-up test, at a range of speeds of interest, on a chassis dynamometer. An important advantage of ITPA method comes from the fact that testing instrumentation is shared for static and operational test, which leads to a considerable saving of implementation time.

During the run-up, on the roller bench, accelerations $(A_{op(t)})$, and the acoustic pressure inside the cabin $(\rho_{meas(t)})$, are recorded using the microphone and accelerometers, previously installed for the static test. Where the subscript “*meas*” refers to the actual measured sound pressure inside the cabin, during the run-up. An “*op*” denotes it comes from an operational test, and an “*i*”, as before, is the path number.

Then, all data are transformed to the frequency domain by an STFT.

2.3. Stage3: calculations and results

The calculated acoustic pressure $\rho^{ITPA}(\omega)$ is considered in this method, like in TPA, as the sum of each noise path contribution $(\rho_{path\_i(\omega)})$. This is shown in the Eq. (6), where $h_i(\omega)$ is the noise transfer function of each path from the static tests, $a_{op\_i}\ (\omega)$ is the acceleration corresponding to each path, from the operational test, and n is the total number of paths involved in the test. Thus, the acoustic pressure corresponding to each path $(\rho_{path\_i(\omega)})$ can be calculated, and the summation of all of them leads to the global ITPA acoustic pressure $(\rho^{ITPA}(\omega))$.

$$\rho^{ITPA}(\omega) = \sum\nolimits_{i=1,n} \rho_{path\_i}(\omega) = \sum\nolimits_{i=1,n} a_{op\_i}(\omega) \cdot h_i(\omega) \tag{6}$$

Finally, the validation of the results, and of the method itself, comes by comparison of measured (ρ_{meas}) and calculated (ρ^{ITPA}) acoustic pressure (Fig. 18). From the NVH assessment point of view, the identification of the acting orders is essential to deduce the correct conclusions. Thus, a good agreement between both pressures is the proof that ensures the goodness of the method.

As a summary, the ITPA procedure is presented in the flowchart of Fig. 4. It shows two different kind of tests: static test, to determine the noise transfer function, and dynamic, or operational test, to register system data to assess.

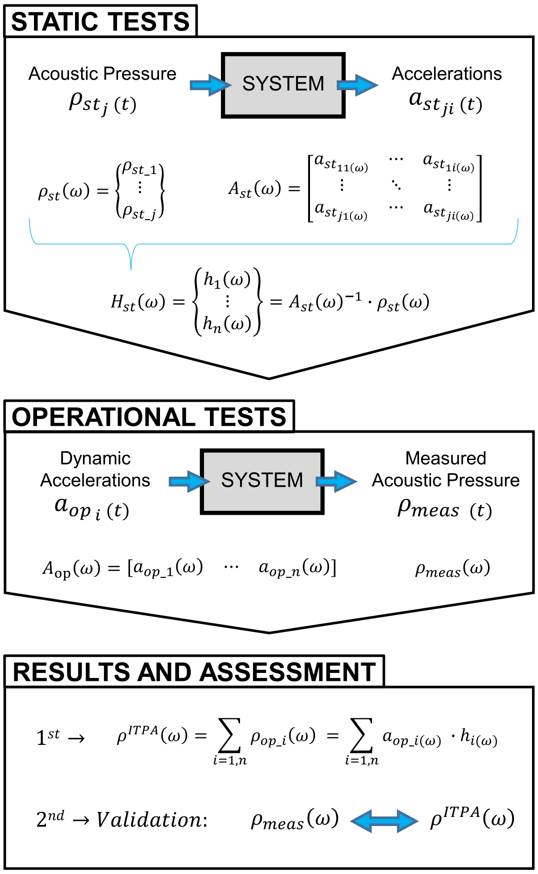


Fig. 4. ITPA procedure flowchart.

3. Experimental validation

In order to evaluate the applicability of the methodology, and to have the empirical validation of the ITPA, an experimental test was performed. An exhaustive results assessment was developed, and also, the comparison of the results with the expected ones allowed to present the validation of the method.

3.1. Experimental test

The experimental part of this work was performed on an electric vehicle. The car used for testing was a Peugeot model iON. Due to the urban characteristics of the testing car, a speed range from 0 to 100 km/h was defined for the dynamic test of the stage 2.

To collet accelerations, a set of accelerometers have been installed in the car, six in the three main powertrain mounts (Axial and vertical directions), and two in the rear suspensions supports points in vertical direction (engine location is on the rear axle in this car). Fig. 5 presents the accelerometers placed in their locations.

Due to the equipment limitations, only the eight paths presumed to be radiating the most structure-borne noise were instrumented. This channel limitation can be assumed because a missing path will not change the structural behaviour of the system. Thus the method still can be proved.

Details about the different positions within the car, and corresponding paths, can be found in the Table 1. Accelerations will be measured at the eight directions shown in the table.

The perceived noise inside the car is recorded as a sound pressure by a microphone located near the driverś ear position, according to the standard ISO 5128:1980 [8] (Fig. 6).

A wooden clapper specifically designed for working in small spaces was used as a noise source for introducing the acoustic pressure in the system. This device is powerful enough to excite the car structure, as well as portable, very easy to work with, and provides good repeatability during the tests. Clapper characteristics can be found in reference [29].

Table 1
Accelerometers locations and measurement directions on the vehicle.

Path	Location	Position	Direction
1	Engine mount (Front left)	1	Vertical (y)
2	Engine mount (Front right)	2	Vertical (y)
3	Engine mount (Rear)	3	Vertical (y)
4	Left suspension	4	Vertical (y)
5	Right suspension	5	Vertical (y)
6	Engine mount (Front left)	1	Axial (x)
7	Engine mount (Front right)	2	Axial (x)
8	Engine mount (Rear)	3	Axial (x)

To perform this test, the clapper (Fig. 7) is placed over the driver seat, thus ensuring the noise source origin is as close as possible to the driver position. The tester should operate the clapper from the rear seat, shielded behind the driver seat back rest, to minimise, as much as possible, interfering with the sound propagation inside the car.

To avoid the influence of any acoustic bouncing effect through the windows, the car was placed inside a semi-anechoic chamber (Fig. 8). This way, the influence of any source from the exterior of the vehicle is also avoided.

A series of static tests was performed, using the clapper to introduce an acoustic pressure (ρ_{st_j}) inside the car, near to the driver position. Then, the acoustic pressure, and the corresponding produced accelerations (A_{st_ji}) at the defined points, were registered. An example of recorded data from a clapper test can be seen in Fig. 9.

Eq. (5) requires the use of the inverse matrix method to calculate the noise transfer function *Hi*. In order to obtain the square accelerations matrix, at least eight clapper tests are required, given that eight paths have been considered. However, to avoid possible errors due to manual manipulation of the clapper, instead of eight, a total of 40 clapper test were performed in this work. These tests were split into five groups, in order to calculate the mean values. Then the obtained eight test mean values were introduced in the Eq. (5) to obtain the noise transfer function *Hi*.

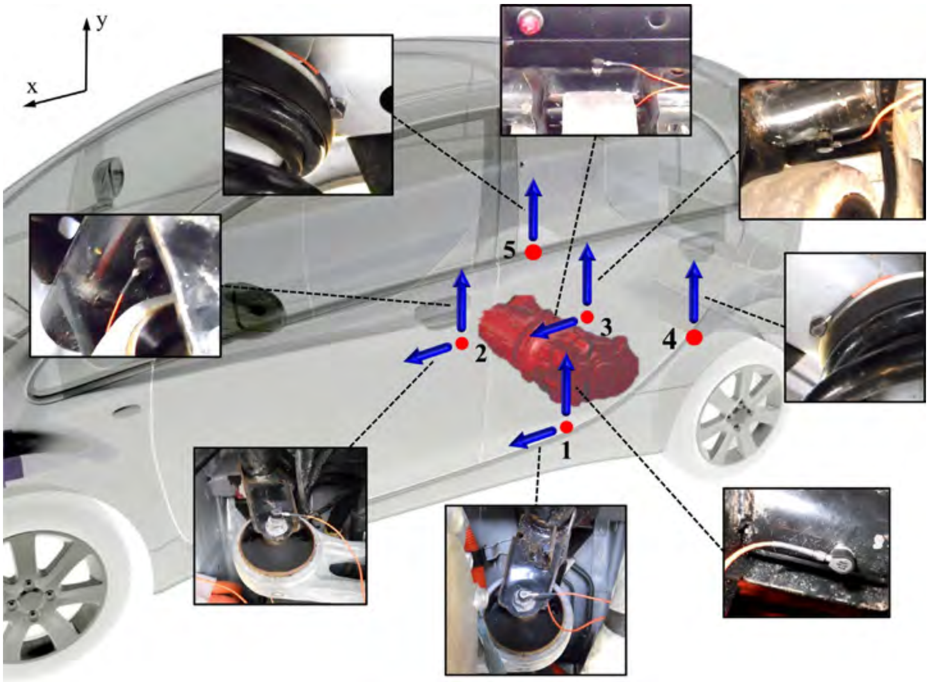


Fig. 5. Installed accelerometers and their positions on the vehicle.

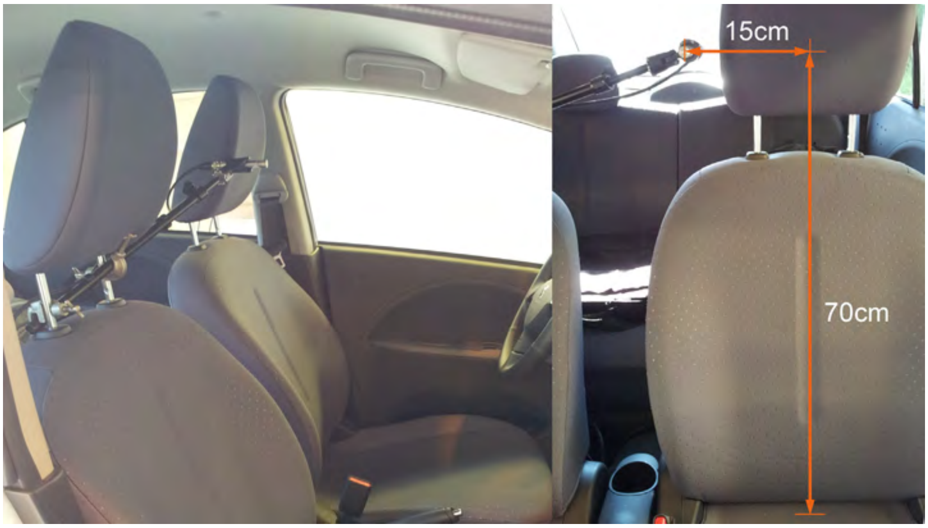


Fig. 6. Microphone location inside the car.

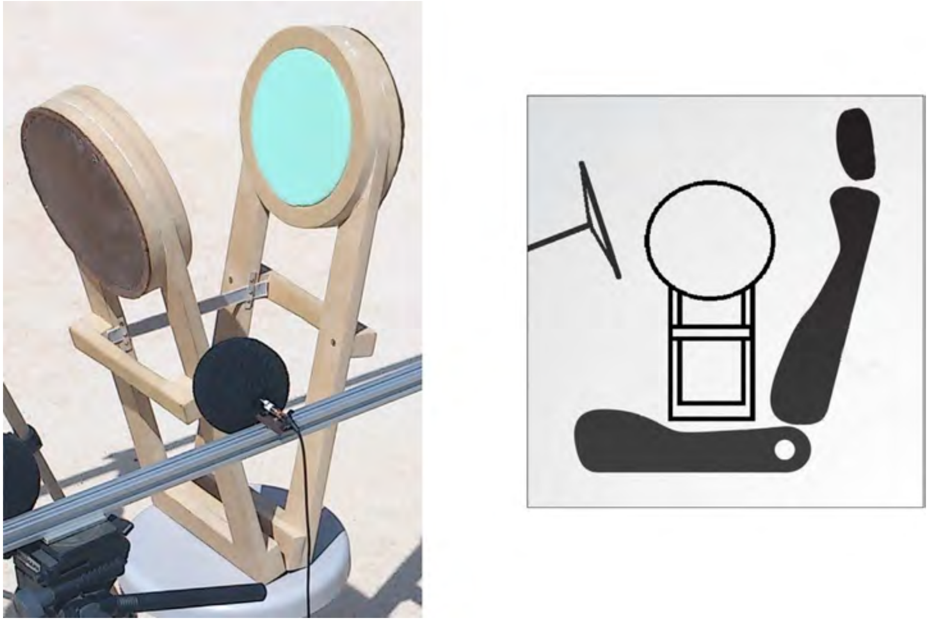


Fig. 7. Noise source (Clapper) and position inside the car.

Fig. 8. Vehicle inside the semi-anechoic chamber.

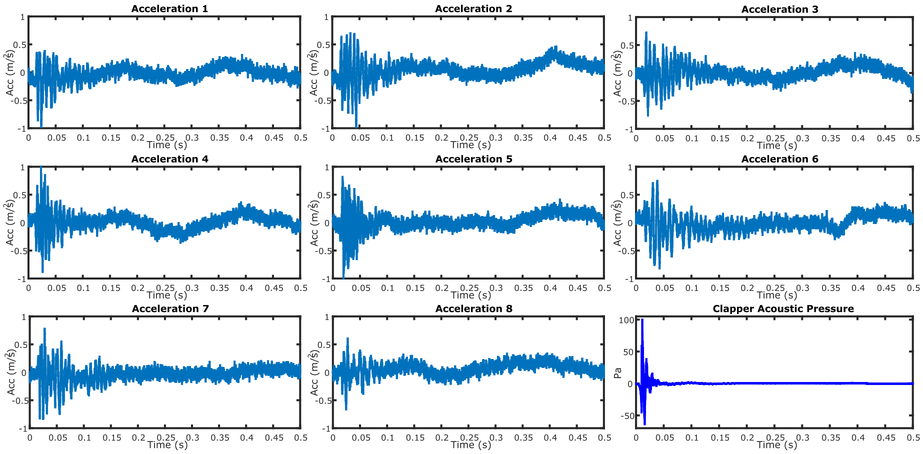


Fig. 9. Recorded data from a clapper test in the time domain.

Fig. 10. Car installed on the roller bench.

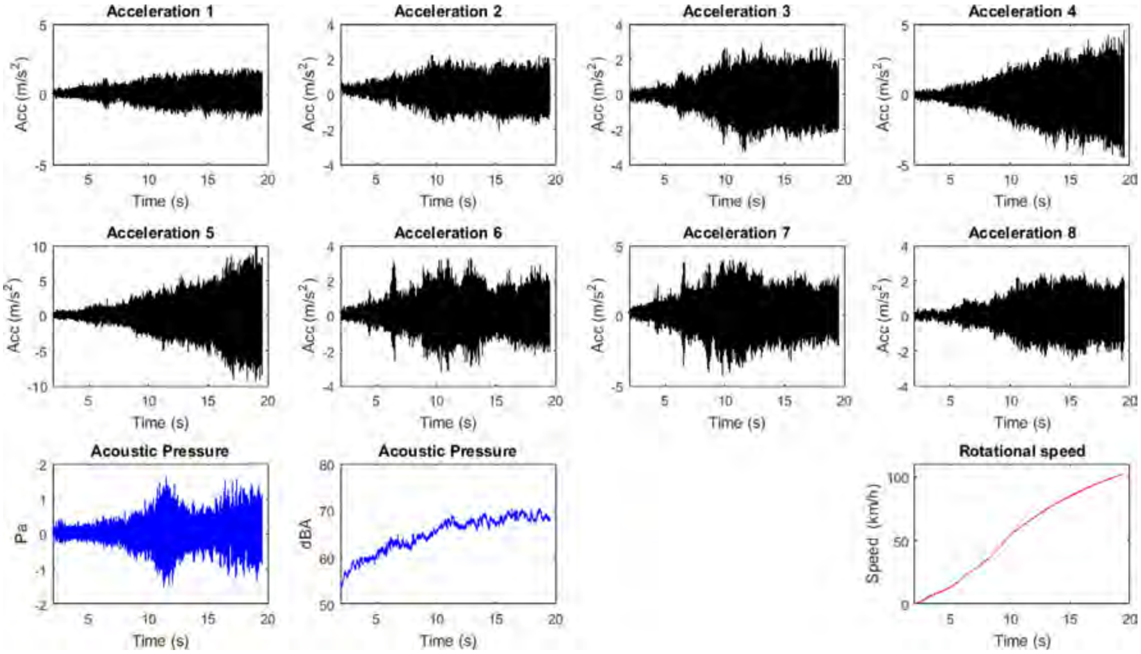


Fig. 11. Example of the data gathered during the dynamic test.

For the dynamic test, the car was secured on a chassis dynamometer (Fig. 10) and a run-up test from 0 to 100 km/h was performed. Accelerations and the acoustic pressure inside the cabin were recorded, using the already installed instrumentation in the car. The different signals recorded during this stage are presented in Fig. 11.

Once the dynamic part of the test has been performed, the ITPA calculated acoustic pressure (ρ^{ITPA}) can be obtained by applying the Eq. (6). Thus, it will allow to determine, not only the global noise perceived in the cabin, but also to identify the particular behaviour of each considered path, and its partial contribution to the cabin noise, depending on the frequency.

3.2. Evaluation of the results

Fig. 12 shows the obtained sound pressure level for the different transmission paths (Fig. 5), presented in a 3D plot of Frequency, Speed and Acoustic Pressure.

Looking at the Fig. 12, it is noticeable that the different noise contribution, from each path, depends on the frequency region. Low (<500 Hz) and medium (500–1200 Hz) frequencies are more impacted by the 1st to 3rd paths and the 6th to 8th paths, which corresponds to the engine mounts. High frequency region is dominated by the influence of the 4th and 5th paths, which are the suspension points.

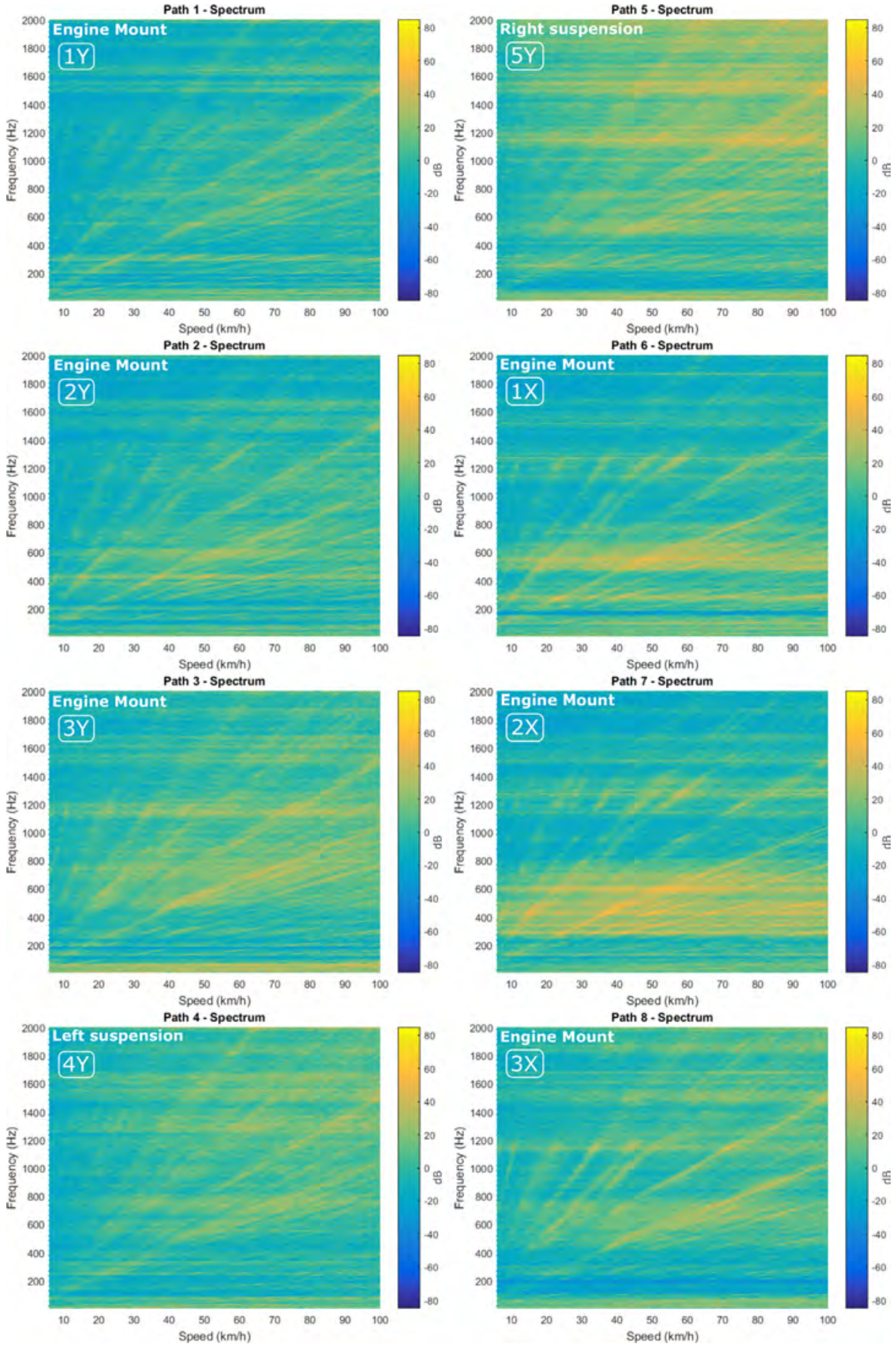


Fig. 12. Sound pressure level in dB(A) for each transfer path against frequency and vehicle speed.

Table 2
Main harmonics of the system.

Harmonic	Rationale	Order number
Transmission: Wheel (two helical gears pairs)	(25/42) × (18/65)	0.16
Transmission: Intermediate shaft (first gear pair)	(25/42)	0.6
Engine: Rotation (reference)	–	1
Engine: Number of pair of poles (4)	4 × 1	4
Engine: Number of poles (8)	8 × 1	8
Transmission: Second pair pinion	(25/42) × 18	10.71
Engine: Number of stator coils (16)	16 × 1	16
Transmission: First pair pinion	25 × 1	25
Engine: Number of stator dents (48)	48 × 1	48

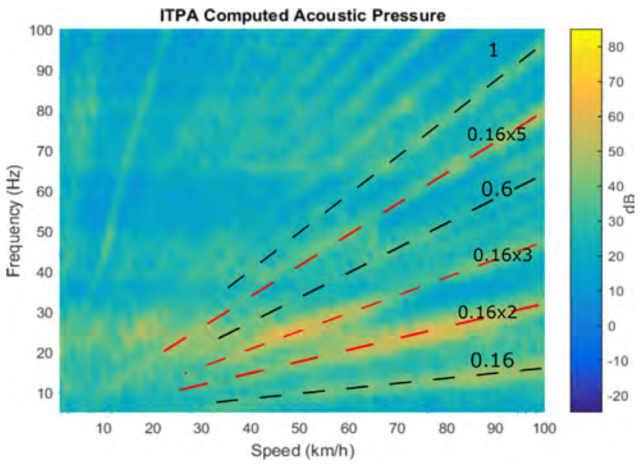


Fig. 13. Detail of calculated acoustic pressure (ρ^{ITPA}) lower frequencies, showing the lower engine orders.

Thus the initial conclusion is that the electric motor has a major influence in low and medium frequency ranges, whilst the suspensions are the critical elements at the higher frequencies. But in a system like this, with having so many different elements in movement inside a complex car structure, initial conclusions are not enough, and a deeper study of the different harmonics, and paths contributions, is necessary.

There are several harmonics signals that are present, to a greater or lesser extent, in all of the noise path contribution charts. The identification of these main harmonics is crucial to identify relevant noise transfer paths, and possible solutions in the assessment.

The physical system in this work is a powertrain composed of an electric motor and a transmission. The electric motor has 4 pair of poles, and 16 coils distributed along 48 stator dents. The transmission system consists of two gear pairs, with an intermediate shaft. The first pair of gears is composed by a 25 teeth pinion rotating in conjunction with the motor, and a 42 teeth gear wheel which rotates with the intermediate shaft. The second pair of gears includes an 18 teeth pinion in the intermediate shaft and a 65 teeth gear wheel rotating in conjunction with the wheels' shaft.

Several harmonics are expected in this system. The first harmonic, or reference order, in the system will be the electric motor rotation. Then harmonics due to the rotor pair of poles, rotor poles, stator coils and stator dents should be present as orders 4, 8, 16 and 48 respectively. Regarding the harmonics due to the transmission, orders 0.16, 0.6, 10.71 and 25 should be identifiable too. This is due to the physical characteristics of two gear pairs, with an intermediate shaft (see Table 2 for more details).

Fig. 13 shows a calculated acoustic pressure (ρ^{ITPA}) chart detail, between 5 and 100 Hertz, with the first three harmonics identified. Several harmonics of the wheel rotation order (0.16) can be found together with the main orders.

The rest of the main orders, or harmonics, are presented in Fig. 14, where the ρ^{ITPA} shows a similar shape to the individual paths, but with higher levels of acoustic pressure. In the lower part of this chart, the wheel harmonics are clearly noticeable as well.

Identification of predominant paths is crucial for a noise assessment to become a useful tool. This allows to define the relevant elements within the system, which is necessary to work for improving the noise and vibrational response of a system. To determine the critical, or more relevant path, or paths, a deeper study of the relevant orders in the system is required. Thus, a new serie of charts in the frequency domain, following closely the lines of the main harmonics, was elaborated.

Figs. 15–17 present the calculated sound pressure levels (SPL) from each path, for every identified main harmonic in the system. Calculated total SPL ("Sum") and the total measured SPL ("Meas") were included at the bottom of the charts, as reference. There is a clear difference in the behaviour of the paths depending on the frequency. Hence SPL order charts have been divided in three groups, lower, medium and higher frequencies.

From lower orders in Fig. 15, it can be concluded that dominant paths correspond to the engine supports on the Y axis. Such a conclusion makes sense since the engine mounts must support the torque when the power is applied to the wheels.

As the medium frequency range is reached, a change in the behaviour is observed (Fig. 16). The 6th and 7th path contributions stand out above the rest. Thus, it means that predominant direction changed here to X axis. The physic sense of such a conclusion is that the behaviour of left and right engine mounts, at these frequencies, need to be improved to offer better support.

In the high frequency range (Fig. 17), 6th and 7th paths are still present, but 4th and 5th path contributions are increasing as the frequency goes up. This means that the main effect changes, from X axis of engine mounts, to the suspension supports. The suspension paths contribution will be more relevant the higher the frequency.

As a summary, it can be concluded that the 1st, 2nd and mainly the 3rd path, bring the more significant contribution in lower frequencies. The medium frequency range is dominated by the 6th and 7th paths. At the higher frequencies 4th and 5th paths are the dominant ones, with some influence from the 6th and 7th paths at the beginning.

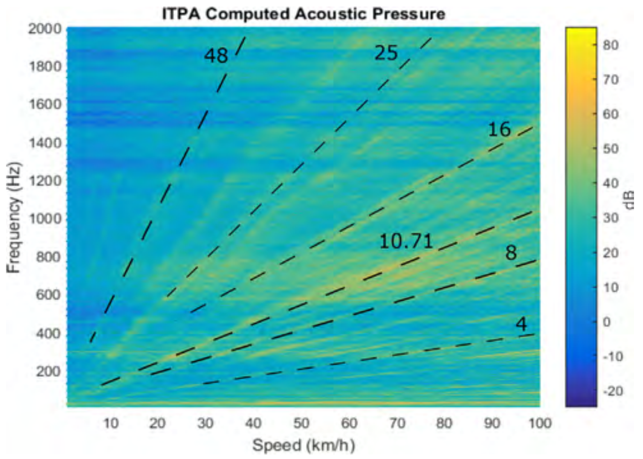


Fig. 14. STFT of calculated acoustic pressure (ρ^{ITPA}), illustrating the higher engine orders.

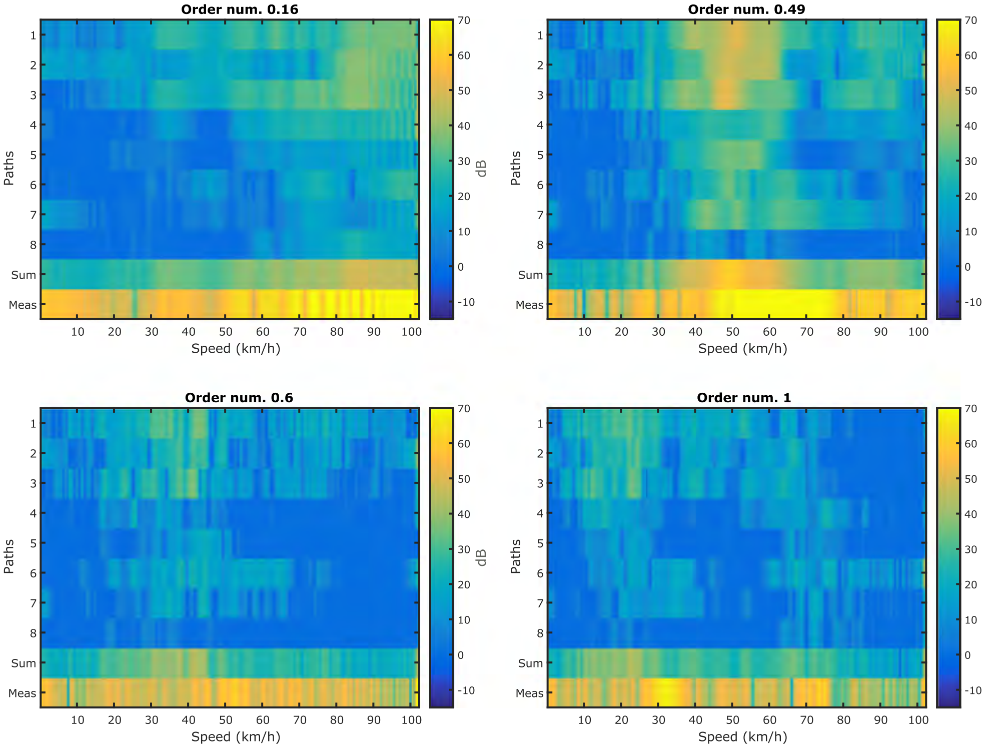


Fig. 15. Sound pressure levels (SPL) due to paths (1–8) in orders 0.16, 0.49, 0.6 and 1.

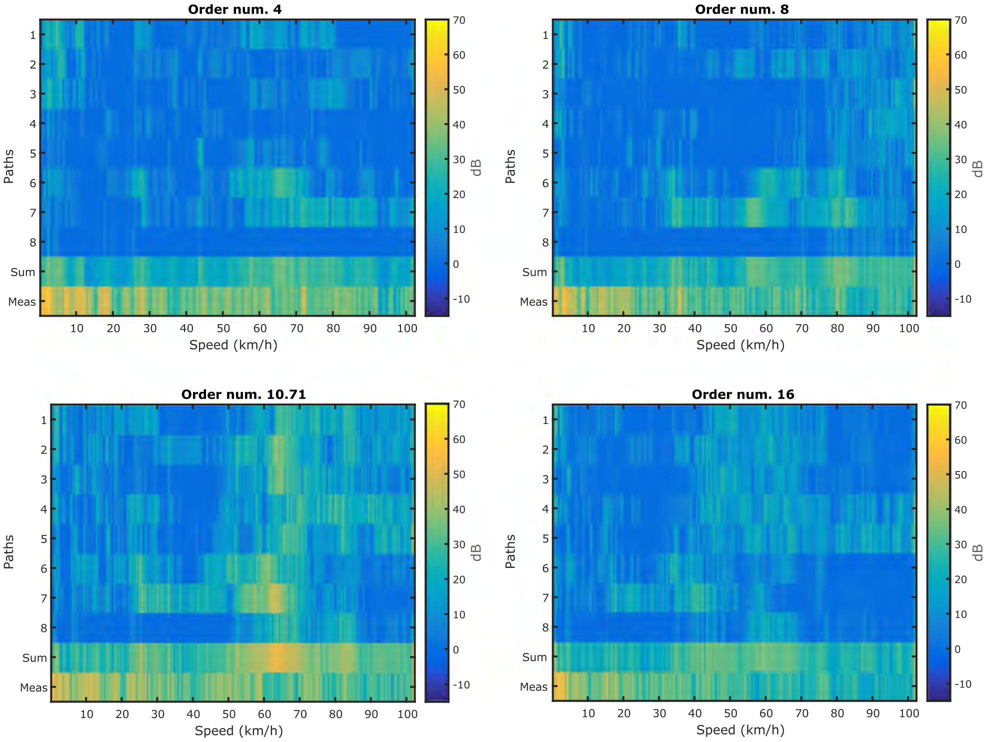


Fig. 16. Sound pressure levels (SPL) due to paths (1–8) in orders 4, 8, 10.71 and 16.

3.3. Results assessment and model validation

The independence between static and operational data is the main advantage of this approach over other methods. This avoids the inherent limitations of the "backward–forward" procedures. Here, the noise transfer function comes from static tests, where the system is isolated from external influences.

As the ITPA incorporates both static and operational tests, the comparison between the calculated (ρ^{ITPA}) and the measured (ρ_{meas}) acoustic pressure is conclusive enough to assess the validity

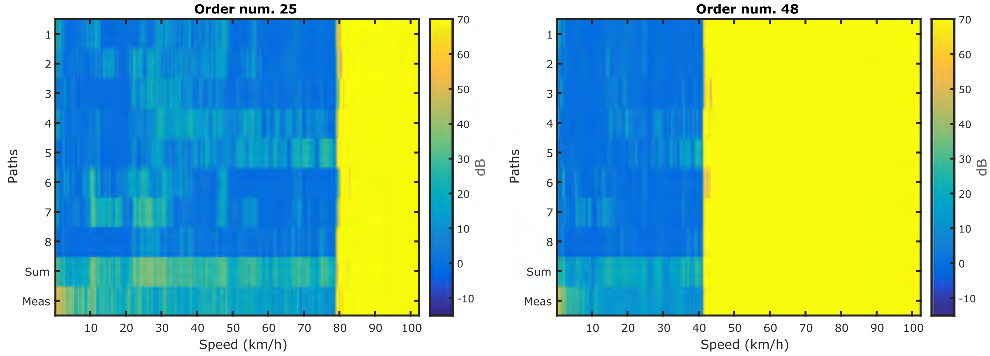


Fig. 17. Sound pressure levels (SPL) due to paths (1–8) in orders 25 and 48.

of the results, and thus, the method itself. Fig. 18 shows two sound pressure level/frequency/speed charts, corresponding to the comparison between, calculated or computed $\rho^{ITPA}(\omega)$ and measured $\rho_{meas}(\omega)$ signals. As can be seen, both spectrum show an adequate agreement, and all the main harmonics can be clearly identified in both charts.

In the lowest frequency range shown on the Fig. 18, below 200 Hz, there is a frequency band with a noticeable difference in the sound pressure levels, this behaviour is within expectations. It indicates that it may be missing sources or missing paths in the system. The air borne contribution, which is present in the measured signal but not in the calculated one, is clearly a source of variations here. In addition to that, eight channels can be enough to prove a method, but not to exhaustively define a complex structure, like a body car, so due to experimental limitations, some paths could be missed.

To show the differences between calculated and measured pressure levels, a comparison at several constant speeds has been carried over (Fig. 19). The fact that some punctual bands are higher in the calculated signal than in the measured one is suggesting that a missing path may be overloading some of the existing ones. Also, this effect moves to higher frequencies as speed increases, which perfectly matches with the behaviour of a structural path.

But even considering the existing differences between some bands, the value of average, minimum and maximum difference between measured and calculated sound levels are 1.92 dB, −5.25 dB and 9.41 dB respectively, which is an indication of the good agreement between both spectra.

Fig. 20, an enlargement of the Fig. 18, is showing a detail of the region below 350 Hz of both spectra. Harmonic signals are clearly identified, not only matching in both charts but also more defined in the calculated one. It makes easier to find harmonics on the computed signal, which is the goal of the method described in this work.

Moreover, there is an interesting detail in Fig. 20, just below 300 Hz. Here, there is a static noise due to the fan installed, in front of the car, for the dynamic test on the roller bench. It can be seen how ITPA is mathematically trying to avoid the effect of this fan, as this is an external noise, and therefore, no included in the noise transfer function of the system. This behaviour demonstrates the suitability of the ITPA methodology, as it is highlighting the noises transmitted by the studied system and reducing those that are considered to be external to the structural paths.

A detailed assessment of the test results was exposed to determine which are the relevant paths transmitting vibrations and noise in function of the frequency. It can be concluded that the electric motor paths have a major influence in the low and medium frequency range, whilst the dominant elements in the higher frequency range are the suspensions supports. It is also important to note that the detailed analysis of the main harmonics of the system has brought same conclusions than the work of Diez-Ibarbia et al. [15], as it can be seen in the Fig. 19 of their work, where a summary of the partial paths contributions using TPA on a similar electric car is presented.

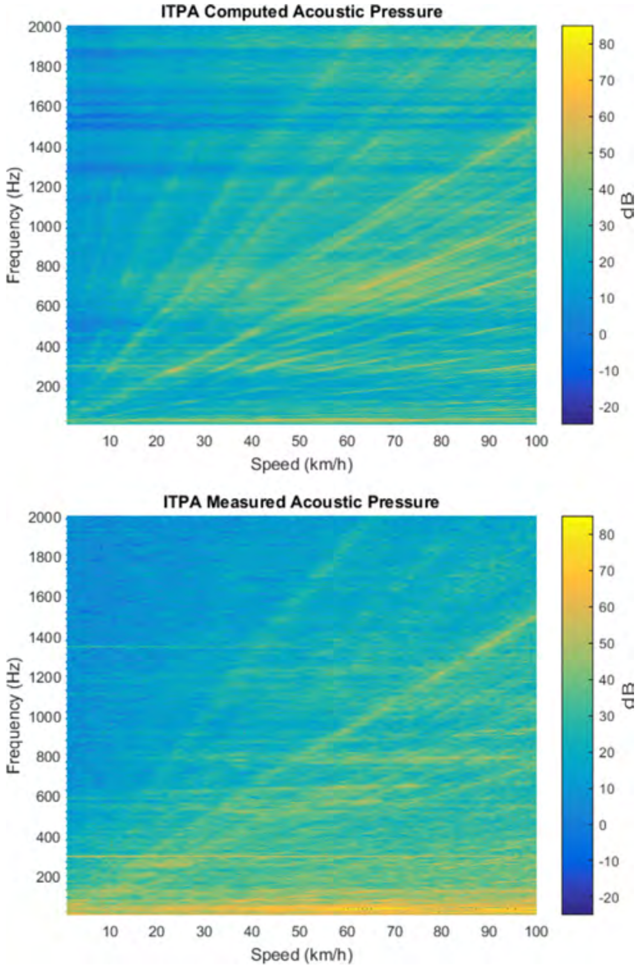
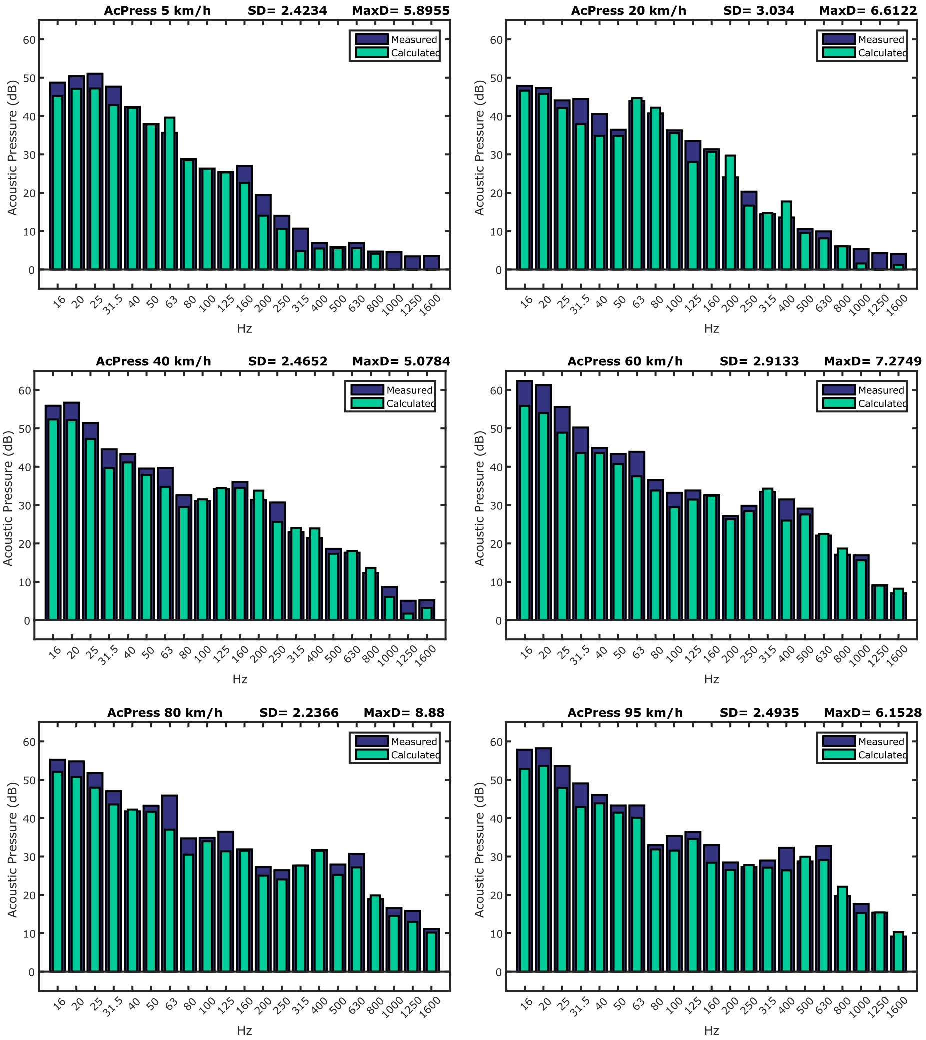


Fig. 18. Calculated (ρ^{ITPA}) Vs the measured (ρ_{meas}).

ITPA results were validated by comparing the measured and calculated response signals of the vehicle interior noise. The validation showed that the ITPA leads to results in good agreement with reality. ITPA has proved to be accurate enough to analyse and determine the partial paths contributions to a sound, from data in operating conditions,


AcPress 5 km/h SD= 2.4234 MaxD= 5.8955

AcPress 20 km/h SD= 3.034 MaxD= 6.6122

AcPress 40 km/h SD= 2.4652 MaxD= 5.0784

AcPress 60 km/h SD= 2.9133 MaxD= 7.2749

AcPress 80 km/h SD= 2.2366 MaxD= 8.88

AcPress 95 km/h SD= 2.4935 MaxD= 6.1528

Fig. 19. Calculated (ρ^{ITPA}) Vs Measured (ρ_{meas}) one-third octave bands for representative speeds.

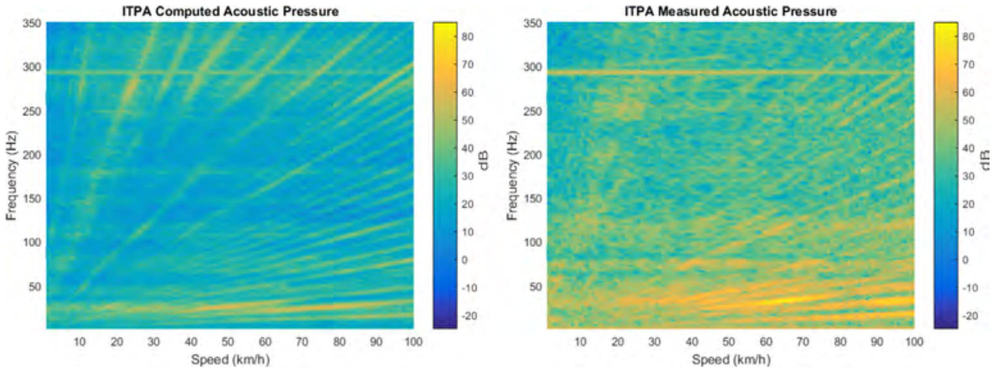


Fig. 20. Lower frequencies detail between Calculated (ρ^{ITPA}) and the measured (ρ_{meas}).

Thus the suitability of the method has been validated in two ways: Firstly, due to its agreement between calculated and measured sound pressure signals inside the cabin, and secondly by the concordance with the results of other researchers through different methods.

Even considering the differences between the calculated and measured acoustic pressures, the method has been proved. Since a larger number of sources, or paths, can be included for improving the accuracy, without varying the methodology.

4. Conclusions

A novel NVH method was developed in this work. Inverse Transfer Path Analysis basic principles and methodology were presented. An experimental test was conducted on an electric car to prove and validate the model applicability. Conclusions from this research can be summarized as:

1- Adequate agreement between measured and predicted noise allows assessing the critical paths on the system. Even without considering the air-borne effect, and with the limited paths considered, the Standard Deviation average is below ±3 dB, and the harmonics match in both signals.
2- The model can provide accurate conclusions to determine the dominant paths involved in the noise transmission. Based on the empiric experience [2], the overall testing time could be reduced up to one tenth of the testing time needed with other methodologies.
3- The omission of the air borne contribution is a limitation in this work, as sound pressure levels between measured and calculated interior noise will always present differences.
4- The limited number of paths considered in this test may cause that some path can be neglected. This can be an inconvenience, especially if the model is sensible to the effect of missing paths.

Possible future research topics would mainly concentrate on expanding the number of paths in the test, checking the sensibility of the system when one of the paths is missing, or even testing the model including the airborne paths. It can also be expected that this approach could be introduced for noise assessment tests in the industrial field, where time is a relevant factor.

Thus it can be concluded ITPA is a useful tool to perform NVH assessments on cars. The method is leading to the identification of path contributions to the cabin noise and delivering correct conclusions in car noise assessments. ITPA can be used as a rapid alternative to the TPA method to obtain qualitative results within a short period of time.

CRediT authorship contribution statement

Ginés Cervantes-Madrid: Conceptualization, Methodology, Software, Validation, Formal analysis, Investigation, Resources, Data curation, Writing - original draft, Writing - review & editing, Visualization, Project administration, Funding acquisition. **Ramón Peral-Orts:** Methodology, Formal analysis, Supervision, Writing - review & editing. **Nuria Campillo-Davo:** Visualization, Resources, Supervision, Writing - review & editing. **Héctor Campello-Vicente:** Resources, Writing - review & editing.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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