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Experimental approaches for NVH study of Electric and Hybrid Electric Vehicles, Annex 3A: Transfer Path Analysis for vehicle NVH refinement: Application on EV / HEV vehicles

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Subchapter 3A

2.1 Transfer Path Analysis for vehicle NVH refinement: Application on EV / HEV vehicles

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Abstract. In the current framework, hybrid and electric vehicles have the advantage of polluting the environment less than other conventional technology-based vehicles, nevertheless one issue of concern has been the noise impact that they generate. The main objective of this proposal is to improve the acoustic perception of the driver in the cabin. This is the reason why Transfer Path Analysis and Operational Path Analysis methods were performed in an electric vehicle. A comparison between both methods taking into account the structure-borne paths from the motor and the suspension points to the cabin has been performed. One of the aims of this study is to check the versatility of these methods, commonly used in petrol engine cars. This assessment includes the analysis of the noise contribution of each path depending on the frequency and vehicle speed range and moreover, the path noise impact for harmonic excitation due to the physical components of the electric vehicle.

2.1.a Introduction

NVH issues in vehicle design have been receiving increasing interest from the manufacturers during last decades. An interesting testimony of this fast development can be found in [1] where the author describes how a small NVH department composed by about 12 engineers with limited testing facilities was increased in only 5 years up to a team about 90 persons working in a new center with more than 15 additional facilities.

One of the reasons for this success can be founded in the market and in the necessity of providing quiet and refined NVH vehicles to satisfy customer demands. The development of more advanced, powerful and cheaper technologies for NVH measurements jointly with the necessity of reducing the time expend in trial and error test has been a fruitful context where several NVH analyses techniques were developed.

One of these techniques was the so called Transfer Path Analysis (TPA) which main aim was the identification of the individual contributions via different transmissions paths from “sources” as powertrain or wheels to the “receivers” which will be the passengers. This identification provides the basis for further troubleshooting and product improvement from the point of view of NVH. Moreover in spite of automotive industry has been the field where TPA has proven their utility, its use has been extended to other sectors as printers, refrigerators, ships, wind turbines, helicopters and aircrafts [2].

In this chapter the main questions concerning TPA will be described and the current available approaches to carry out this kind of analysis in practice will be presented. Afterwards, the main concerns regarding their application in EV / HEV will be reviewed. Then a case study corresponding to the experimental test carried out at Loughborough University on the EV provide by PSA group will be discussed. Finally some preliminary conclusions from the results obtained will be presented.

2.1.b Techniques for TPA

There are several terms to describe the same kind of technique. Source Path Contribution (SPC) [3-4] or Noise Path Analysis (NPA), are also common acronyms used to designate the same technique. The former techniques used for this kind of analysis were developed in the late 70's and it was initially based on analysis of coherence between signals coming from several channels [5]. But it was during 80's that the so called classic TPA was described in [6].

According to the classical approach the system can be divided in an active part and a passive part (or a source part and a receiver part) which are connected together by some kind of interface [7]. The objective is to be able to discern what contribution is due to each transmission path considering both structural and acoustic paths and how excitation is transmitted towards the reception point.

Classical TPA approach considers that the response ($R(\omega)$) at a certain location (and at a certain frequency) can be obtained as an addition of contributions due to different paths, structural of acoustical according to the expression, Eq. 1-A.

$$R(\omega) = \sum_{i=1}^n FRF_i(\omega)F_i(\omega) + \sum_{j=1}^r NTF_j(\omega)Q_j(\omega) \quad \text{Eq. 1-A}$$

Where FRF_i denotes the Frequency response function between the receiver and an input force at location i (F_i) while NTF_j represents the Noise Transfer Function between the receiver and the acoustic load (Q_j).

Thus if operating excitations (F_i and Q_j) and transfer functions (FRF and NTF) are known it is possible to determine what is the contribution of each path as well as the frequency range where this contribution is more important. If there is not missing paths in the analysis, the synthesized response and the measured one should be nearly the same and therefore this provides a simple mean for validating the analysis.

Nevertheless there are some important problems involved in the application of classical TPA. Two steps are necessary to build the expression (Eq. 1-A): i) measuring excitations in operational conditions and ii) determining transfer functions.

Measuring operational excitations

Measuring operational excitations is a complex task in real systems as it is the case of automobiles. Due to space restrictions and practical issues usually it is not possible to measure operational excitations directly by a simple transducer like a load cell. In fact that is one of the main practical problems concerning real TPA applications. To circumvent these difficulties two main techniques have been proposed: the mount stiffness method and the so called inverse force identification method.

The mount stiffness method

The mount stiffness method is useful when active and passive parts are connected by simple flexible joints. In these cases, operational forces are obtained from the movement (usually acceleration) of the input and output sides of flexible mounts, applying Eq. 2-A, under the assumption that its dynamic stiffness $K_i(\omega)$ it is known.

$$F_i(\omega) = K_i(\omega) \frac{a_{ai}(\omega) - a_{pi}(\omega)}{-\omega^2} \quad \text{Eq. 2-A}$$

Determination of dynamic stiffness of flexible mounts requires specific test procedures as their behaviour is strongly non linear by nature and it is necessary to consider different load levels reproducing the real operating conditions. Moreover dynamic stiffness involving cross coupling effects should be defined in modern set ups making much more difficult their test and assessment. As a consequence dynamic stiffness method usually does not solve the problem and instead it includes more uncertainty in the TPA analysis.

Inverse force identification

In order to avoid the direct measurement of forces, theoretically it is possible to obtain it from the measured response from a certain number of locations in the structure.

$$H_{jk}(\omega) = \frac{a_{qk}(\omega)}{F_k(\omega)} \quad \text{Eq. 3-A}$$

When the number of measured response locations is equal to the number of unknown forces, the resulting frequency response matrix is square and it is possible to obtain the corresponding force by its inversion according to

$$\{F(\omega)\} = [H(\omega)]^{-1} \{a_q\} \quad \text{Eq. 4-A}$$

Nevertheless, while it is possible to calculate a unique response from known excitations, the inverse problem is not straightforward as the same response could be produced with a different forcing history. Moreover, practical application of this theoretical concept does not provide acceptable results likely due to the ill-condition of the frequency response matrix at frequencies near the resonances [8]. As a consequence significant errors are committed when matrix is inverted and therefore the results become not useful. To avoid this problem several numerical techniques have been proposed involving a redundant number of measurement points (some authors [9] claims the overdetermination factor should be at least 2). Thus, if response is measured in q locations while loads are applied in p sites, the relationship between responses and excitations will be given by

$$\begin{matrix} \{a_q\} \\ (q \times 1) \end{matrix} = \begin{matrix} [H(\omega)]^* \\ (q \times p) \end{matrix} \begin{matrix} \{F(\omega)\} \\ (p \times 1) \end{matrix} \quad \text{Eq. 5-A}$$

Arriving to a formulation where a rectangular matrix (H(w)) should be inverted. Pseudoinverse and SVD are techniques which have been applied by researchers to accomplish this task [10-11]. Thus, the force can be obtained from the accelerations by

$$\{F(\omega)\} = [H(\omega)]^+ \{a_q\} \quad \text{Eq. 6-A}$$

Where H(w)+ represents the pseudoinverse. The application of this procedure can be compromised by different kind of errors as inaccuracies and noise in the experimental measurements as well as insufficient, inadequate or inaccurate sensor data.

Determining transfer functions

On the other side, determining transfer functions usually requires dismantling the active part assuming that the behaviour of the passive part remains the same. This task is time consuming and the last assumption is not always acceptable. Several excitation techniques

can be used in this task (shakers, loudspeakers or impact hammers) and commonly the reciprocity principle is applied due to space restrictions in the connection points between active and passive parts. It can be noted that if the load identification method uses the stiffness or matrix approach, the FRF can be performed at the same stage, measuring the acoustic pressure as well as the accelerations.

Once the transfer functions have been determined, the response at the receiver might be predicted by Eq.1-A. Then, it is possible to identify the contribution of each path to the response in the so-called partial path contribution (PPC) plot. This plot provides by rows the contribution of every single path as a function of frequency (or engine RPM). Care should be taken on the consideration of phase for each path and frequency as the interaction between paths could result in a lower value of the response.

In order to validate the procedure the predicted response is compared with the record actually obtained in the location of interest. Differences between recorded and predicted might be attributed to lack of paths considered in the analysis, inaccurate acquisition procedures and numerical problems involved in the approach used for force identification.

2.1.c Techniques for OPA

In spite of classical TPA is powerful and can be applied in a wide range of problems, some practical issues constrain its usefulness. The main drawback of this kind of technique is related with the amount of time necessary to complete the analysis. Mounting and dismounting tasks associated with the calculation of FRF as well as the inherent limitations on the measurement of forces have led researchers to the development of new techniques entirely based upon *in-situ* measurements. The main concern of these techniques is avoiding the identification of loads by measuring instead the response (commonly accelerations) at certain locations distributed near the loads and (or) along the transmission paths. In essence, this approach can be derived from the combination of equations 1 and 6 that focusing in structural paths and removing noise-related terms, for the sake of simplicity, becomes

$$R(\omega) = \sum_{i=1}^n FRF_i(\omega) [H(\omega)]^+ \{a_q\} \quad \text{Eq. 7-A}$$

As the main advantage of this kind of techniques is linked with the fact that only measurements of machinery in operation are required for characterization of system transmission paths, this group of techniques are commonly designed as Operational Path Analysis (OPA). Thus, in the case of multiple receivers $R_i(\omega)$, Eq. 7-A can be expressed in matrix form as

$$\{R(\omega)\} = [T(\omega)] \{a_q\} \quad \text{Eq. 8-A}$$

Where matrix $T(\omega)$ represents system transmissibilities between the reference points located in the structure and the target receivers. Despite the fact this method has shown some limitations due to the requirement of considering all the paths and using low-coherence signals, companies are deeply interested in improving them because of the possibility of obtaining TPA-like results quickly.

In the following some possible approaches for OPA are briefly described,

Direct application of transmissibility concepts

Three main stages can be considered for the application of the most direct approach of OPA [12], i) Operational measurements, ii) Transmissibility function and iii) Results and assessment. Operational measurements are taken in the same way as was described for the TPA. With the acceleration and the acoustic pressure measurements, the transmissibility functions are calculated using Eq. 9-A.

$$R(\omega) = \sum_{i=1}^n T_i(\omega) \cdot a_i(\omega) \quad \text{Eq. 9-A}$$

Finally, the obtained results are assessed, determining which transfer path is the most critical one for a specific frequency.

Indirect application of transmissibility concepts

This method is a variation of the general OPA method previously explained and can be divided into three main stages, i) Operational measurements, ii) Frequency response function calculation and iii) Results and assessment. The main difference between these methods is that while in the previous one the transmissibility functions are calculated, in this method, the FRFs are calculated instead. This method is more theoretical than practical, because the applicability of these concepts in a vehicle has still to be proven. The fundamentals of this method are based on the application of the transmissibility concepts. According to [13], the key point to obtain the FRF without disassembling the system (in TPA this is a requirement) is to apply a known force in the subsystems of interest. In Fig. 1-A, the simplified model of the engine and body can be appreciated as an example of application. If the FRFs of the subsystems of interest need to be determined, the F1, F2 and F3 forces have to be known. This is a difficult task, because the devices used to introduce the excitation forces (hammers or shakers) on each subsystem are not designed to co-exist with the forces generated by the engine.

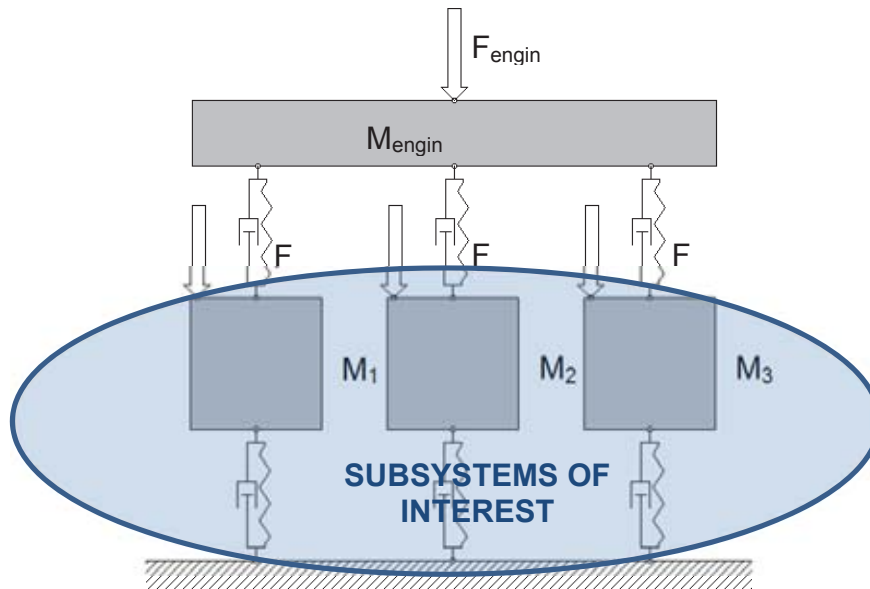


Fig. 1-A. Lumped model of a vehicle

OPA with exogenous inputs (OPAX)

This method is another variation of the general OPA method explained previously and can be divided in five main stages [14], i) Operational measurements, ii) Frequency response function iii) Parametric load model iv) Calculation of operational loads and v) Results and assessment.

The first stage is the same as in the OPA method obtaining the accelerations and acoustic pressure in operational conditions at the points of interest. Then (second stage), once the motor is removed, FRF procedure takes place in the same way as in the second stage of the TPA procedure.

In third stage, using the accelerations and forces obtained in the FRF stage, an exogenous force parametric model can be determined. To achieve this, the accelerations and forces are introduced in the chosen formulation (Eq. 10), calculating the parameters or constants of the equation.

$$F_i(\omega) = f(\text{parameters}, a_{ai}(\omega), a_{bi}(\omega)) \quad \text{Eq. 10-A}$$

Once the equation is solved and calibrated, the accelerations in operational conditions are introduced in the formula, allowing the simulated forces of the engine at the interest points to be found.

Lastly, the simulated acoustic pressure is determined by means of the transfer function (second stage) and the operational forces calculated in the previous stage. Assessing whether the model is valid by the comparison between the calculated acoustic pressure and the measured one (first stage).

2.1.d Comparison between methods / approaches

In order to determine which method is the most suitable for this case study, the strengths and weaknesses of each method have been assessed:

TPA

STRENGTHS: The most important aspect of this approach is that the results are usually close to the real scenario. Moreover the validity of the results is easy to determine because a comparison between the real acoustic pressure value and the simulated one can be made. Another interesting aspect is that the cross-coupling effects, introduced when the engine is mounted, are always considered and an error is avoided.

WEAKNESSES: One of the main problems with this method is the execution time because a lot of tests hammer have to be used in order to determine every transfer functions. Also, in this case study, this method presents the further problem that the engine must be removed in order to perform most of the tests.

OPA (Direct)

STRENGTHS: This method has two main advantages. The first is that it is not necessary to calculate the loads that the engine is introducing in the system because the transmissibility functions do not require it. Following from the first advantage, a large numbers of tests are not required making this method the quickest and at the same time the easiest.

WEAKNESSES: Although this method is the easiest and quickest, it might produce more problems because of the difficulties that are presented in its validation. The first weakness is the estimation of the transmissibility matrix because the inverse acceleration matrix is needed. To solve this problem an estimator is usually used (H1 estimator) and the input forces are limited to low-coherence. Unlike the TPA method, cross-coupling affects the transmissibility function hugely, for this reason low cross-coupling is required to obtain reliable results. Also this method is sensitive and all the active paths must be considered

in order to obtain valid results. This is because the contribution of the missing path is added to the rest paths.

OPA (Indirect)

The strengths and weaknesses of this method are similar to the previous one but there is a main difference. To perform this method, the forces of the isolated subsystems of interest must be known.

OPAX

STRENGTHS: Measurements are simpler compared to the traditional matrix inversion TPA method because mount stiffness data is not required. On one hand the estimated parametric model of forces used allows the determination of additional information such as dynamic stiffness and forces in other operational conditions by introducing other accelerations. On the other hand, it is more accurate than dynamic stiffness method way of load calculation.

WEAKNESSES: One of the main problems with this method is the execution time, although it spends less time than the classical TPA, it is more time consuming than general OPA. Also, in this case study, this method presents another problem which is that the engine must be dismounted in order to perform most of the tests.

Conclusions

With regard to all the information gathered from the literature review, a ranking of the different methods studied has been done. In this ranking, different general aspects have been taken into account and shown in the Table 1-A.

Table 1-A. Ranking among passive transfer path methods

GENERAL ASPECTS	TPA	OPA	OPAX
SPEED	3	1	2
ACCURACY	2	3	1
LESS COMPLEXITY	2	1	3
EASY TO VALIDATE	1	3	1

2.1.e Particular issues of application TPA in HEV / EV:

Electric vehicles (EV) are typically quieter than comparable Internal Combustion Engine (ICE) vehicles. However, they do produce a significant amount of higher frequency tonal noise than traditional ICE vehicles. For example, whining tonal noise from electric motors and generators is typically in the frequency range 400-2000 Hz [15]. Once elements of the EV powertrain are included then the whining tonal noise can be produced up to 3000 Hz [16-17] note that the engine orders excited by an electric drive are higher than for those of an ICE engine, thus, leading to higher frequency interior noise in an EV.

Since the noise from an electric powertrains is no longer restricted to the low frequency region of a traditional ICE, it can be expected that the transfer paths from the powertrain to the vehicle interior now include a relatively stronger contribution from the airborne paths rather than being dominated by the structure borne paths as in a traditional ICE powertrain architecture [18].

Florentin et al. [19] report a comparison of the noise from a small conventionally powered vehicle and a similar sized electric vehicle. As expected, the EV is quieter than the ICE vehicle at low and medium speeds. However, at high speeds the high frequency ringing from the EV can exceed the ICE noise in certain frequency bands. Also, at high speeds the tonal EV noise was perceived as more annoying even in the frequency bands where the measured sound was less than for the ICE vehicle.

A further problem is that without the masking noise of the ICE the sound from the auxiliary components (pumps, inverters etc.) of an electric vehicle are noticeable even at low speeds [9].

Hybrid electric vehicles also present novel challenges. For example, the transition between the electric and the combustion engines stages of operation of a hybrid EV can produce unfamiliar sounds for the driver and passengers [20].

2.1.f Case of Study

TPA techniques have been applied in a vehicle demonstrator in order to validate its usefulness in EV and provide output for the other WG involved in the project. The vehicle under test is based in the Mitsubishi i-MiEV car which variants are also sold in Europe by PSA Peugeot Citroën (PSA) as the Peugeot iOn and Citroën C-Zero. It is a five-door hatchback electric car, driven by a single permanent magnet synchronous motor in the rear axle powered by a 16 kWh lithium-ion battery pack. The vehicle uses a single-speed reduction gear transmission driving the rear wheels and the motor and batteries are water cooled by a conventional radiator with an electrical fan disposed in the front of the car.

Objectives

The main objective of the work carried out on the car under test is demonstrating capability of experimental available methods and techniques to be applied in a real Electric Vehicle development context (interior noise).

In this context and taking into account the means available for these task two particular objectives were defined as the following:

1. Identify noise sources: Develop and apply techniques to separated annoying tonal or multi-tonal sources (electric motor, control system, gear ratio, differential) from broad-band noise (road noise and aerodynamic noise).
2. Apply TPA techniques in order to define and assess the structure-borne paths for identified sources in targeted operative conditions on a roller bench.

Test description

In order to implement both the TPA and OPA methods, some test were performed. In the case of study, two kinds of test were executed, tests in operating conditions on a roller bench and impact tests on a bay lift.

Measurement points location

The first stage to carry out these tests is to locate the transducer depending on the requirements of the measurements. In this case, since the noise perceived by the driver in the cabin wants to be assessed, a microphone in the driver's seat was located. The speed of the wheel shaft was necessary for the analysis stage, therefore a laser tachometer were installed in the rear wheel shaft. Furthermore, it was envisaged that the motor and the road (through the suspensions) generated most of the noise in the cabin. This is the reason why, 8 accelerometers were strategically located in the motor mounts (6 accelerometers) and suspensions (2 accelerometers) in the directions presented in Fig 2-A.

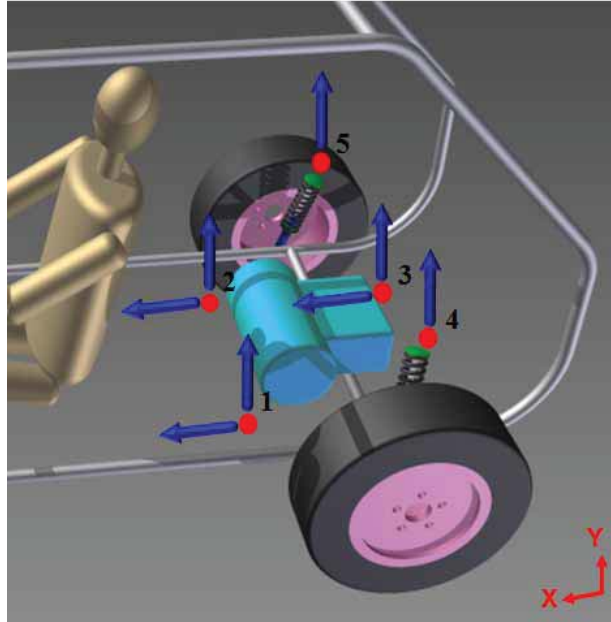


Fig. 2-A. Measurement point locations

In order to clarify the nomenclature and the location of each accelerometer, the Table 2-A is presented, where the information of each accelerometer is shown.

Table 2-A. Location of the accelerometers installed and their directions on the vehicle.

LOCATION	POINT	DIRECTION	ASSOCIATED PATH
Engine mount 1	1	Vertical direction(Y)	1 st
Engine mount 2	2	Vertical direction(Y)	2 nd
Engine mount 3	3	Vertical direction(Y)	3 rd
Suspension 1	4	Vertical direction(Y)	4 th
Suspension 2	5	Vertical direction(Y)	5 th
Engine mount 1	1	Axial direction (X)	6 th
Engine mount 2	2	Axial direction (X)	7 th
Engine mount 3	3	Axial direction (X)	8 th

Test 1: Roller bench with motor switched off

This test was performed with the electric motor switched off on the roller bench. This means that the roller bench provided the energy to the car. Three different specifications were used in this kind of test which are shown in Table 3-A.

Table 3-A. Test specifications.

N° accelerometers	Minimum speed (km/h)	Maximum speed (km/h)	Acceleration (km/h/s)	Noise outside (dB)
3	5	60	3	45
5	5	60	3	45
8	5	100	5	45

It can be noticed that all of them are not in stationary condition. In fact, in order to carry out an OPA, which analyses a wide range of speed, it is necessary to measure vibrations and noise while the vehicle is doing a run-up. This kind of test was performed mainly because some issues with the data acquisition system occurred during the roller bench test 2 which is going to be presented.

Test 2: Roller bench propelled by the EV

In this test, the car motor provided the energy to the system. The specifications of the test performed are gathered in Table 4-A.

Table 4-A. Test specifications.

N° accelerometers	Minimum speed (km/h)	Maximum speed (km/h)	Acceleration (km/h/s)	Noise outside (dB)
8	5	100	5	45

It can be noticed that all of them are not in stationary condition. In fact, in order to carry out an OPA, which analyses a wide range of speed, it is necessary to measure vibrations and noise while the vehicle is doing a run-up. This kind of test was performed mainly because some issues with the data acquisition system occurred during the roller bench test 2 which is going to be presented.



Fig. 3-A. Vehicle under test mounted on the chassis dynamometer (roller bench)

This kind of test was utilised to identify the noise sources and to calculate the transfer paths in the OPA and TPA methods.

Test 3: Impact test

This test was performed with the car on a bay lift, therefore in static conditions. The performance of this test was exciting the different points of interest with a test hammer as the same time as the data acquisition system is collecting all the required data. In this case, the force introduced by the hammer was measured by the force transducer as well as the accelerometers and the microphone were measuring in the considered points.

Data acquisition

As commented, to collect all the data measured by the transducer, an acquisition system was needed. In this case, the acquisition equipment was: Brüel & Kjaer, accelerometers type 4517-002 DeltaTron and Brüel & Kjaer microphone Type 4942-A-021 were used with a Brüel & Kjaer Type 2693-0s4 signal conditioner. All this equipment was checked by laboratory calibrators.

All signals were sampled and acquired by National Instruments acquisition card cDAQ-9178 + 2x National Instruments 9215 drive by a home-made virtual instrument developed in LabVIEW code (see Fig. 4-A).

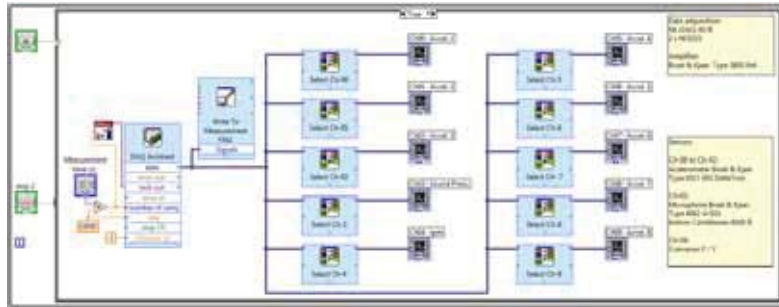


Fig. 4-A. Application to record data in LabVIEW software

Additional tools were developed in MATLAB environment in order to process the recorded signals. A specific script was made in order to plot the different acquired signals on a graph, in time domain; in this way it was possible to focus the attention only on the region of interest (the run-up). After that, a Short Time Fourier Transform was applied to each signal in order to accomplish a time-frequency analysis. This technique enables to find the natural frequencies of the system and, at the same time, determine the most excited orders. The script was created with the possibility to modify the width of the time's window and the overlap in each analysis.

Identification of noise sources

In order to complete this objective the following guidelines were defined:

- Identify source signals from a colour-map or similar plot by simply extract the relevant order from the experimental data in the frequency domain.
- Pay special attention to tonal and multi-tonal sounds coming from gears and electric devices which are subjectively the most annoying noise in EV.

This task was carried out using the colour-maps obtained from the run-up records of Test 1 (electric drive switch off). In Fig. 5-A, the results corresponding to the measurement layout with 5 accelerometers are shown. Acoustic pressure recorded inside the vehicle -at the driver location- is presented at the bottom on the right.

According to previous results and the dynamic behaviour of motor-gearbox system, three main natural frequencies are expected, one in the region between 200 and 250Hz, one between 600 and 650Hz and one more between 700 and 800Hz. These frequency regions can be recognized in acoustic pressure measurements and more clearly, as expected, in the acceleration records (see Fig. 5-A).

A more detailed view for the low frequency region is presented in 6-A. In the upper side, the attention is focused on the orders between the 8 and the 48. As expected, the orders

10.71 and 25 are clearly recognizable because related to the gearbox, while the orders 8 and 48 are not excited because related to the poles of the engine that was switched off.

Attention should be paid to orders 0.16, 0.49 relative to the first order which is the spin velocity of the engine shaft. Order 0.16, that appears to be the most excited, is related to the spin velocity of the tires and the order 0.49 is his third harmonic. With this analysis of the results coming from the measured data it was possible to consider them validated and ready for the second part of the post-processing, made by another group's researcher.

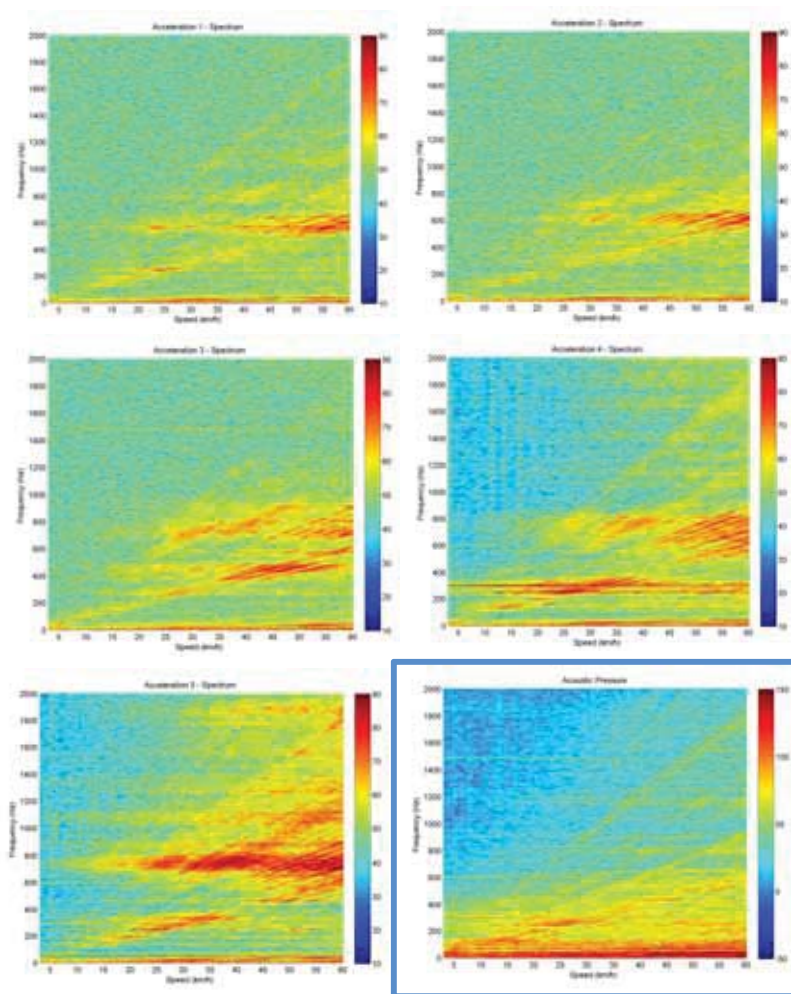


Fig. 5-A. Run-up results in Test 1 (5 accelerometer). Acoustic pressure (lower-Right)

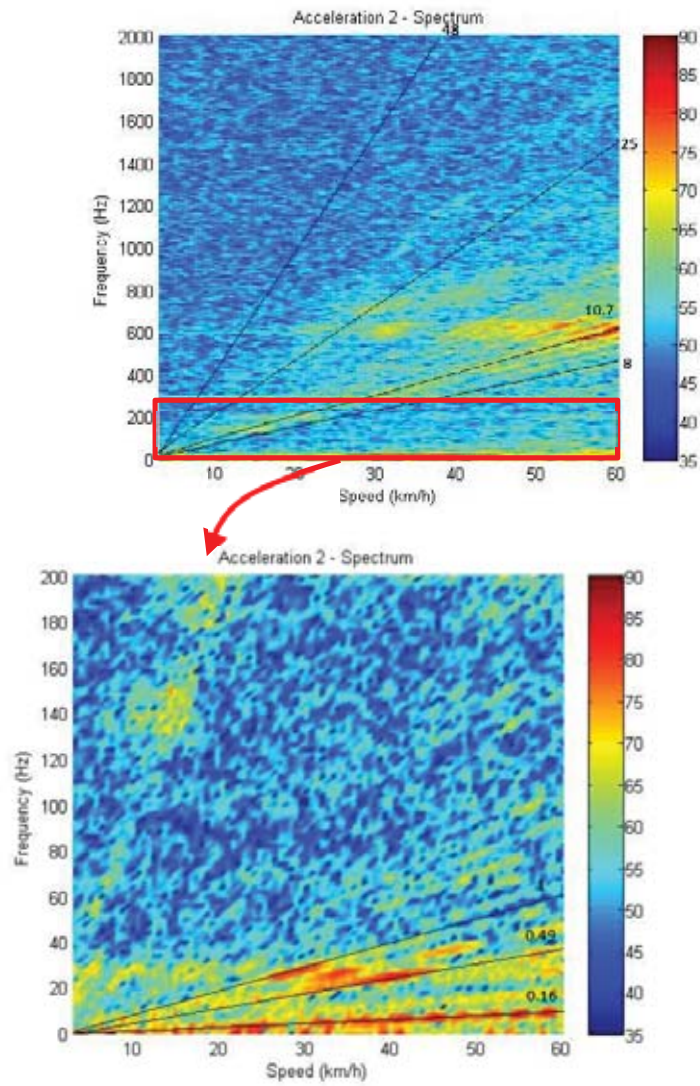


Fig. 6-A. Detail in the low frequency region (engine mounting 2 accelerometer)

OPA results

In this section, the procedure to obtain the OPA results is to be explained firstly, assessing the results lastly. To perform the OPA method, two operational tests are necessary. With the first one, the transmissibility functions are calculated and with the second one the validity of them is checked. The procedure to calculate the transmissibility functions requires the accelerations and acoustic pressure gathered in the acquisition data stage and transform them to the frequency domain. Once this transformation is done, the transmissibility functions are calculated by a mathematical method based on H_1 estimator. Once they are calculated, the acoustic pressure of the another operational test is calculated by the new accelerations and the transmissibility functions. Finally, this computed acoustic pressure is compared with that measured in the second test, checking the validity of the procedure.

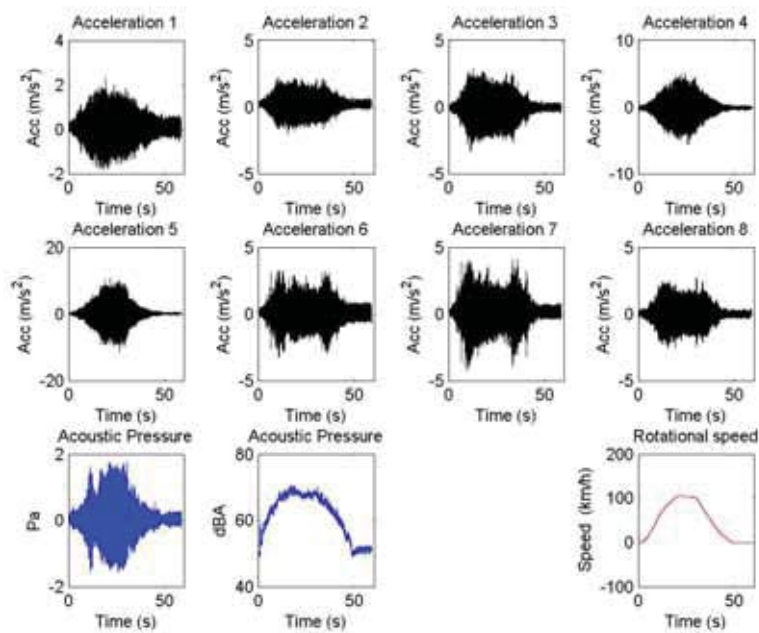
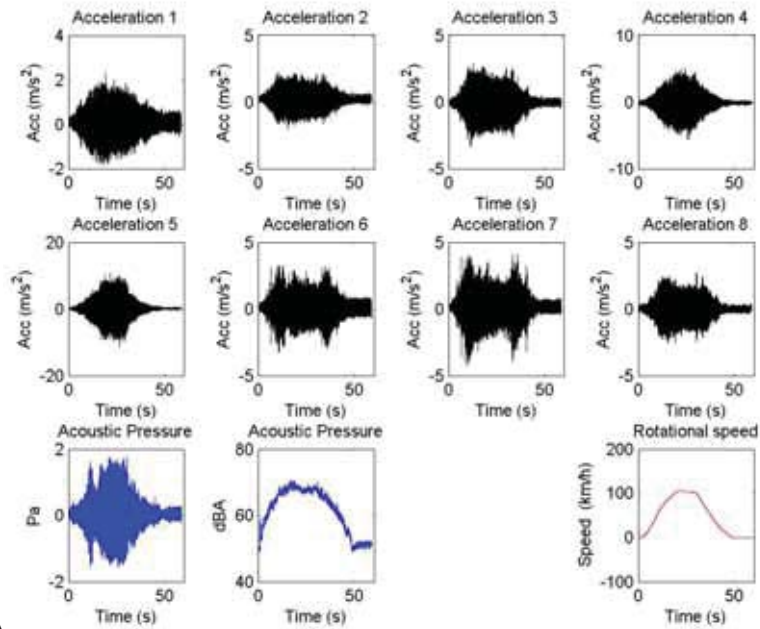


Fig. 7-A. Time-dominie accelerations (6 on the motor mounts and 2 on the suspensions), acoustic pressure inside (in Pascal and dBA) and wheel rotational speed



In Fig. 7-A

Fig. , it can be shown the typical set of results obtained from the measurements in Matlab environment. To calculate the OPA, the first step is to obtain each transmissibility functions by means of the H1 estimator procedure. This procedure requires the auto-spectrum and the cross-spectrum of the accelerations and acoustic pressure measured to calculate these transmissibility functions. In Fig. 8-A, the transmissibility functions in the frequency domain can be seen.

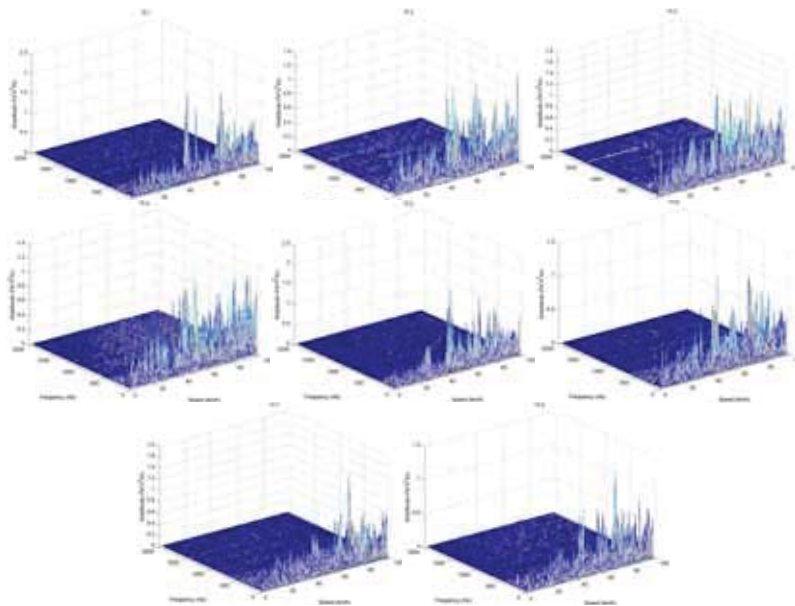


Fig.8-A. Transmissibility functions

In order to check that the mathematical procedure was precise, the calculation of the acoustic pressure and the real measurement has been compared and shown in Fig. 9-A in the frequency domain.

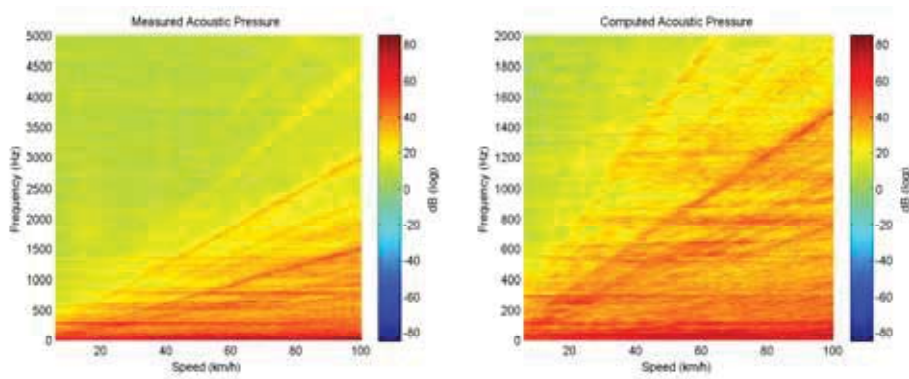


Fig.9-A. Measured and computed acoustic pressure

The next stage is to assess which path affects more inside the pilot's cabin. In Fig 10-A, the contribution of each path is presented depending on the speed and frequency case of study.

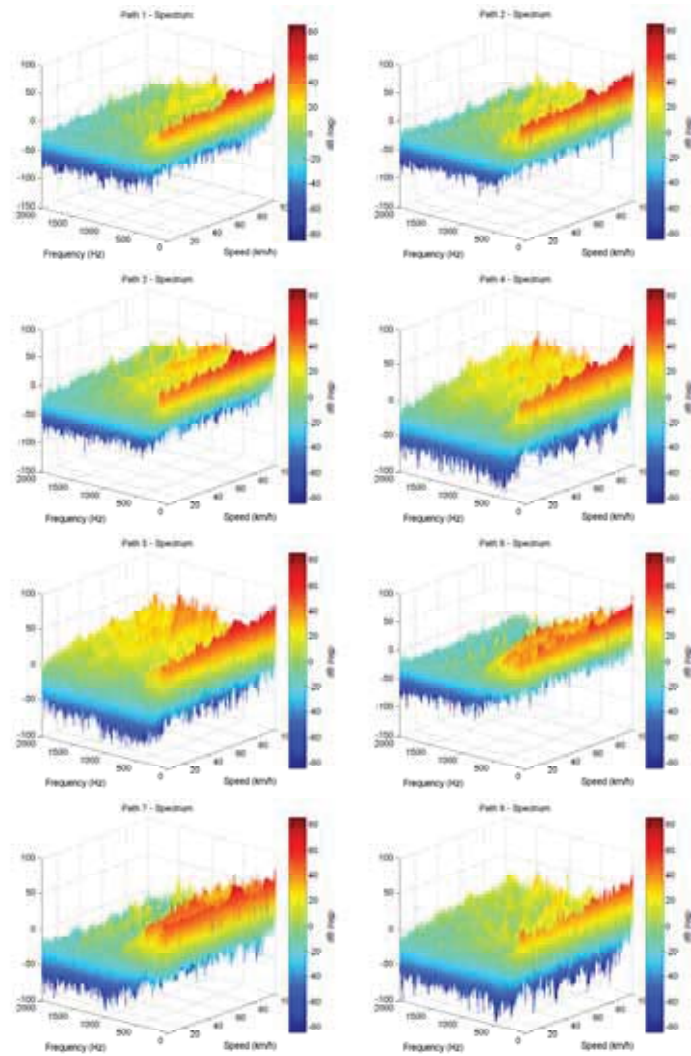


Fig.10-A. Acoustic pressure of each transfer path in the frequency spectrum

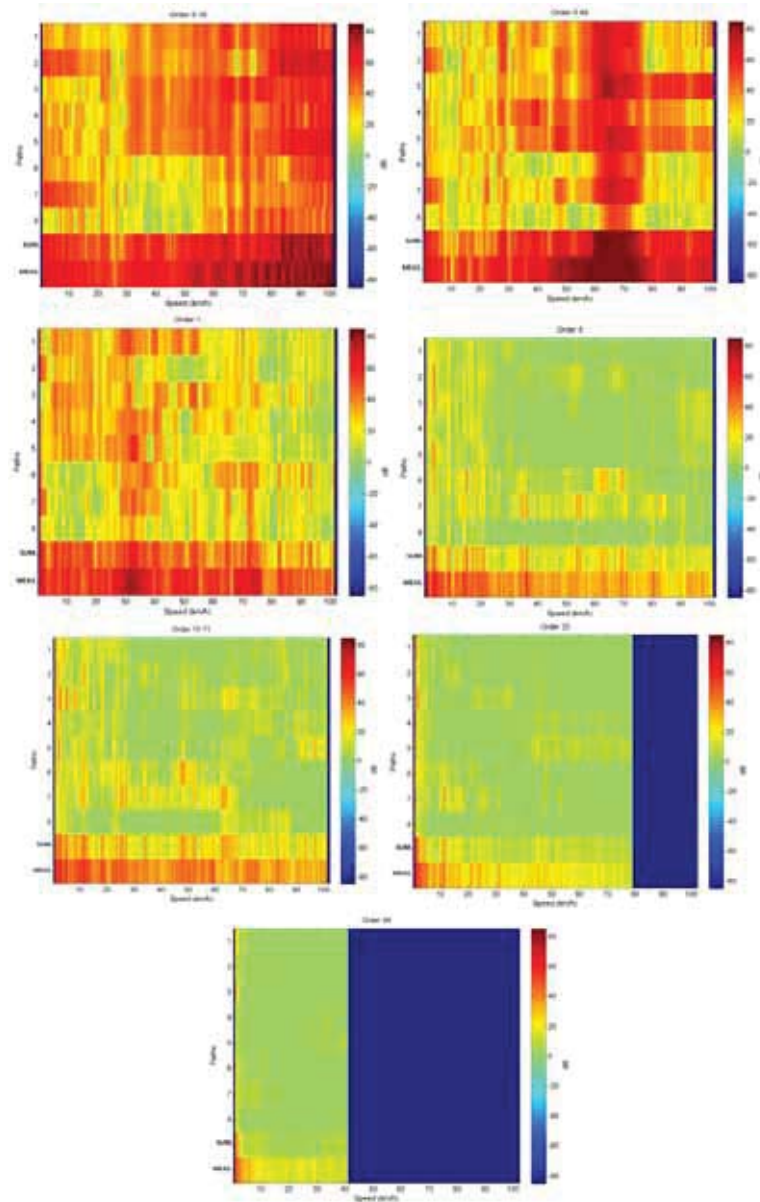


Fig.11-A. Path analysis for the orders of interest

The last stage was to extract the amplitude values of the different orders of interest. Specifically, the orders case of study has been: 0.16 related to the wheel, 0.49 related to the third harmonic of the wheel, 1 due to the rotation of the motor, 8 due to the poles of the motor, 10.71 and 25 due to the gear transmission and 48 due to the coils of the stator

TPA results

In this section, the TPA results are to be shown and assessed. The methodology consists in the post-processing and assessment of two different tests, impact and operational tests as is detailed next. The impact tests were performed with the electrical vehicle on a bay lift to calculate the frequency response functions (FRF) between accelerations and forces first and between forces and acoustic pressure next. Once the paths have been characterize, the operational measurements are used to validate this functions and if new paths need to be considered by the calculation of the acoustic pressure and comparison with the measured one. The acoustic pressure is calculated using the accelerations obtained in operational test and multiplying it by the acceleration-to-force FRF (the equivalent forces are calculated) and then multiplying the results by the force-to-acoustic pressure FRF.

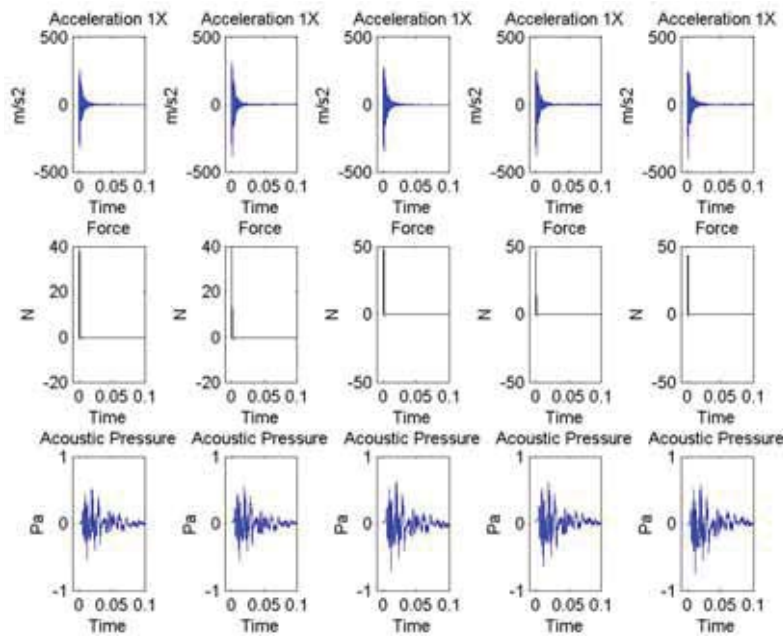


Fig.12-A. Impact test example

From the several impact tests in all the points of interest, five of these impacts have been gathered randomly. As an example, the five impacts in the point 6, the respective accelerations in the same location and the recorded acoustic pressure in the cabin are showed in Fig. 12-A.

With all the acceleration and force data transformed to the frequency domain and applying the inverse matrix method for each frequency, the acceleration-to-force matrix was calculated (Fig. 13-A). Furthermore, with a similar procedure than the previous one, the FRF matrix between forces and acoustic pressures has been obtained (shown in Fig. 14-A).

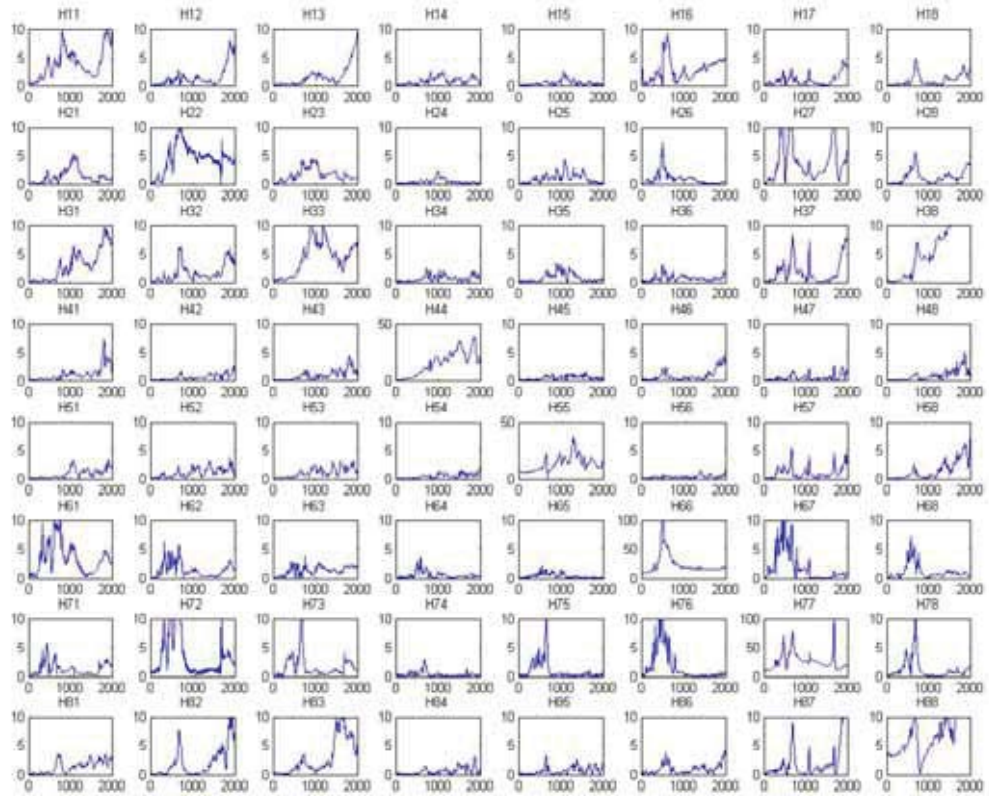


Fig.13-A. Acceleration-to-force frequency response functions

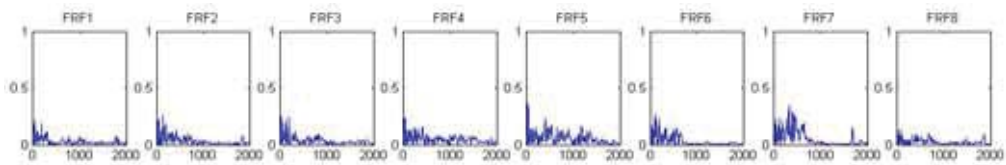


Fig.14-A. Force-to-acoustic frequency response functions

The main aim of the impact test is to calculate these two matrixes which describe the system behaviour. The next step is to gather accelerations and acoustic pressure data of the operational test. With the accelerations in frequency domain and the acceleration-to-force matrix, the equivalent operational forces can be calculated (Fig. 15-A).

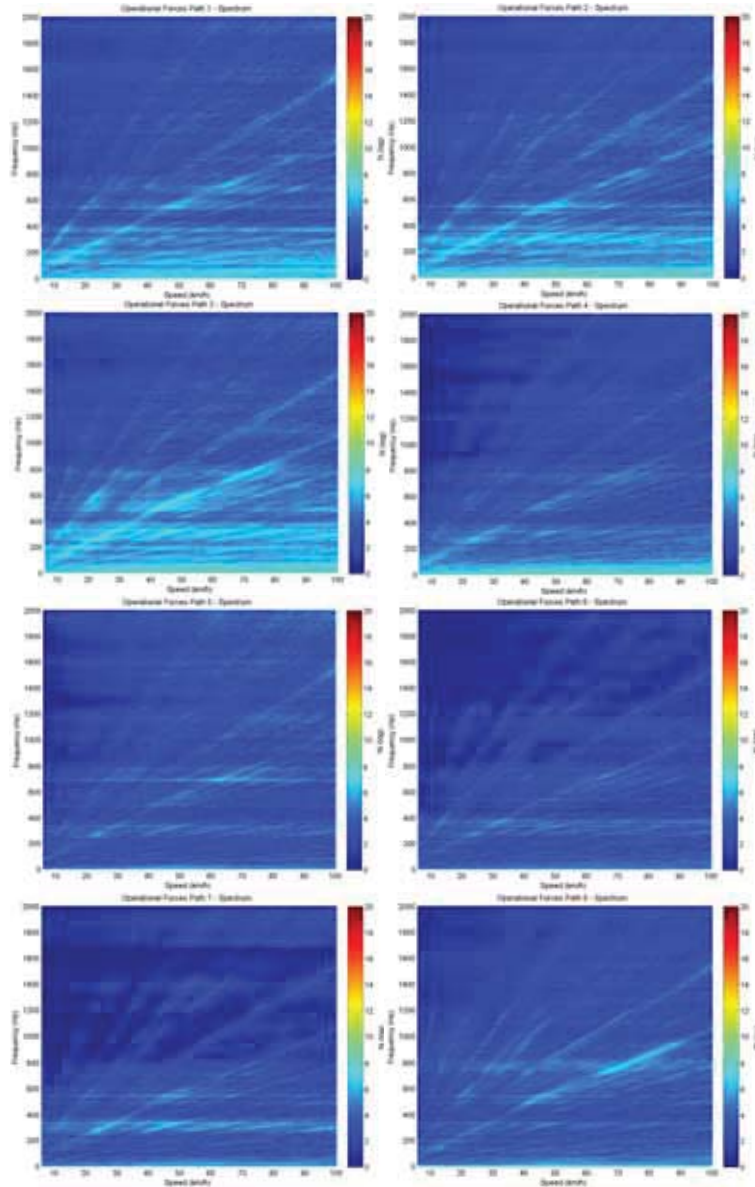


Fig. 15-A. Operational forces

Then, with the equivalent operational forces in each path and the FRF, the equivalent noise that is passing through each path can be calculated. These noise contributions are one of the set of results presented later. Adding all the individual noise contributions, the computed acoustic pressure in the cabin is obtained. To assess the

validity of this calculation, a comparison between with the measured one has been done and showed in Fig. 16-A.

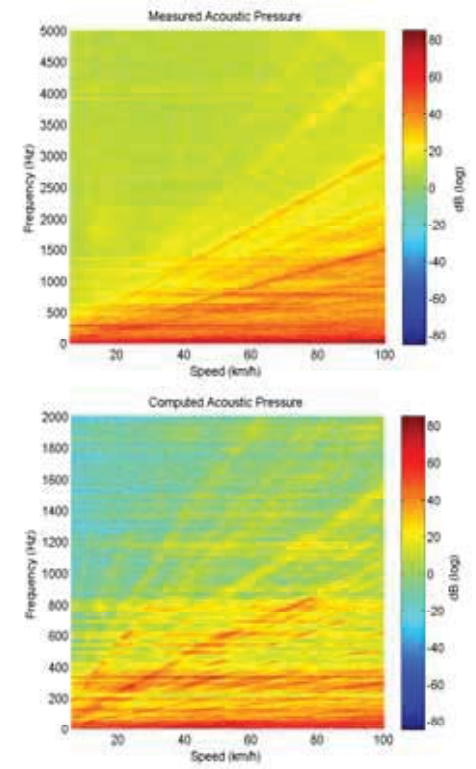


Fig. 16-A. Measured and computed acoustic pressure

The main conclusion that can be extracted from this fact is that the sources which are producing the high frequency noise are not taken into account. Thus, to obtain more reliable results, an increment in the number of paths has to be done.

Once the correlation between the measured and the computed acoustic pressure, two different set of results are presented. The first set of results is the equivalent acoustic pressure of each path in different speed and frequency conditions, see Fig. 17-A. Moreover, the second set of results which present the acoustic pressure for each order of interest depending on the vehicle speed, see Fig. 18-A.

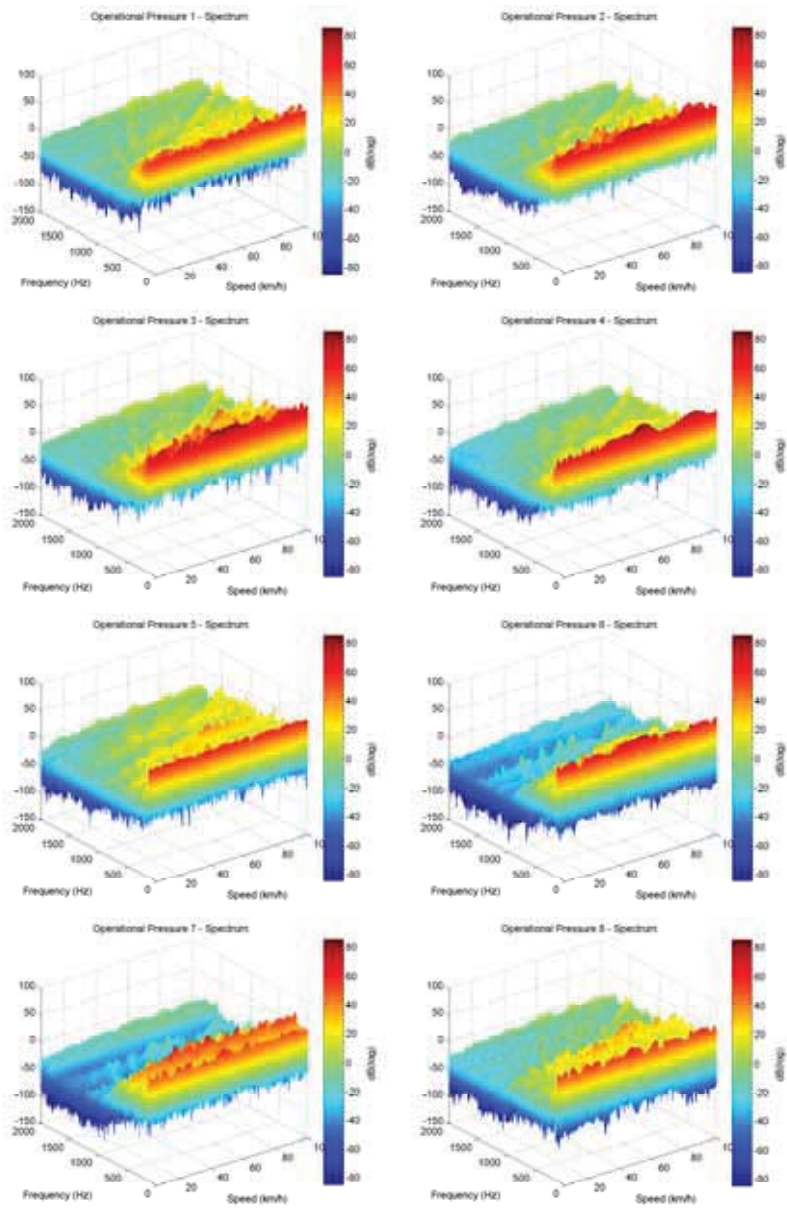


Fig. 17-A. Acoustic pressure of each transfer path in the frequency spectrum

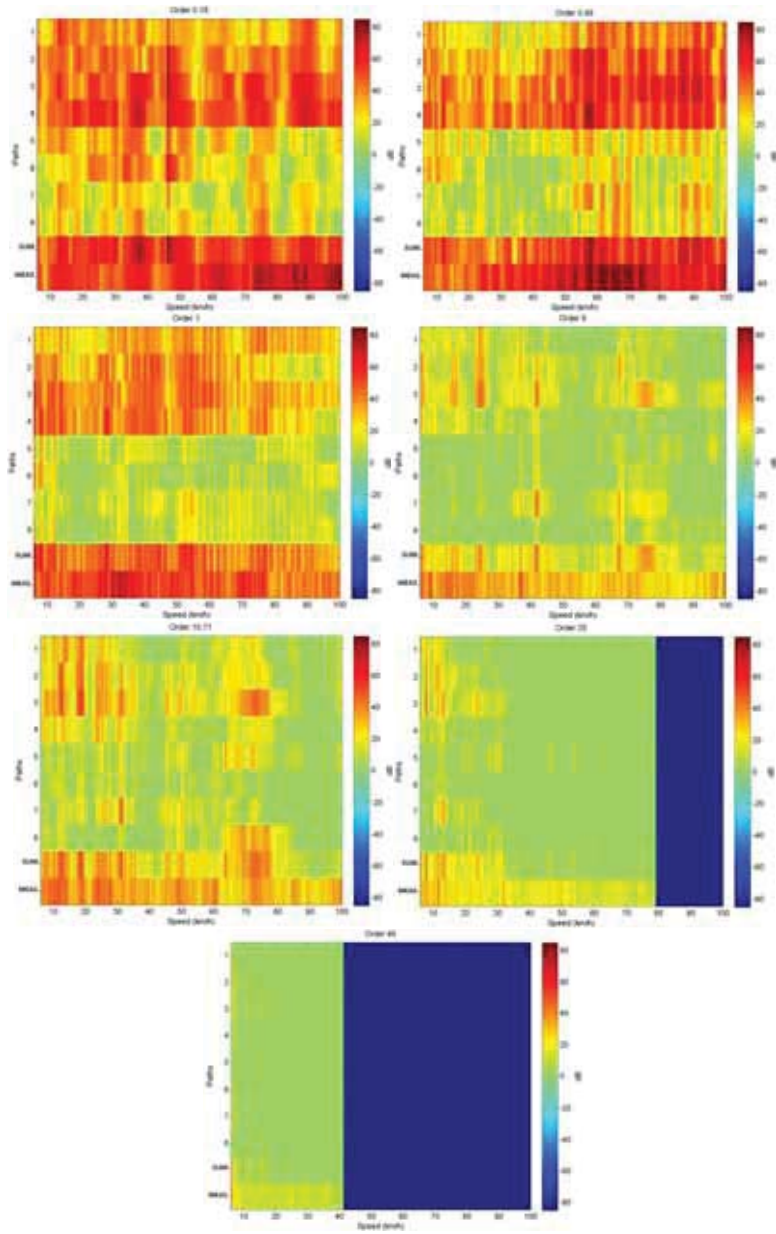


Fig. 18-A. Path analysis for the orders of interest

Comparison between methods

In this subsection, a comparison between the both methods explained is to be made. Firstly the main differences regarding the methodology and data needed to perform it are to be presented and lastly, the main differences among the results calculated by each method.

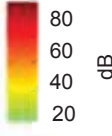
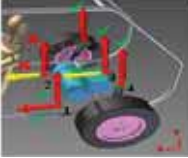
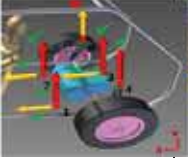
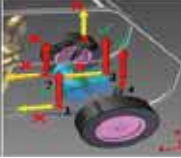
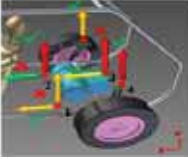
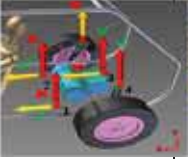
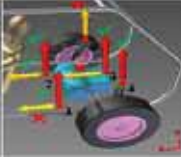
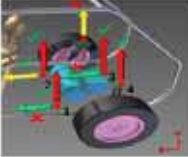
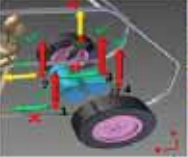
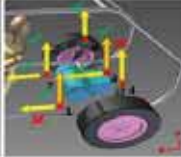
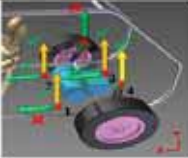
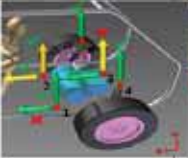
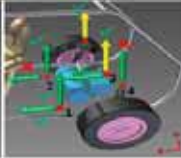
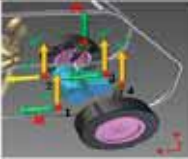
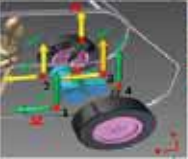
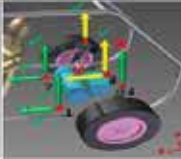

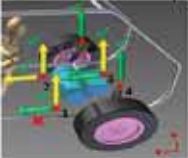
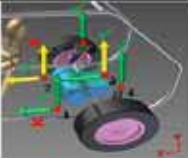

With regard to the methodology, several dissimilarities can be found. Whilst in the OPA only operational tests are necessary to execute the method, in the TPA both operational tests and static tests are compelled (in the case of study impact tests, although other kinds of test can be performed). With respect to this difference, as the transmissibility functions are obtained by the measured acoustic pressure, other operational tests are needed to check the validity of these functions in the OPA method. Nonetheless, in the TPA, as the system properties (FRFs) are calculated by static tests, only one operational test is required. Moreover, whilst the OPA mathematical method to obtain the transmissibility functions uses the measured acoustic pressure in operational conditions, the TPA does not use it to calculate properties of the system. In the TPA case, if all the paths are not considered, there will be a deviation between the measured and the computed acoustic pressure whilst in OPA method, an error in the transmissibility functions is introduced. The former error is noticeable y quantifiable, nevertheless, the latter is not easy to find and check. By comparing 9-A and Fig. 16-A, it can be seen the OPA acoustic pressure agrees better with the measured one than the TPA one. Nevertheless, this does not mean that OPA is better than TPA method.

From the assessment of the first set of results of both methods, a comparison between TPA and OPA acoustic pressure impact of each path has been made. In general terms, it can be appreciated that the spectrum shape and behaviour is the same in both methods, although the amplitude in the TPA results is higher than in the OPA ones

Regarding the second set of results, a comparison between TPA (right hand side) and OPA (left hand side) can be made by

Table 5-A. Similarities between both methods can be found.

Table 5-A. Comparison between methods

Noise Contributions  dB 80 60 40 20	LOW SPEED (< 30 km/h)	MEDIUM SPEED (30 – 80 km/h)	HIGH SPEED (> 80 km/h)
ORDER 0.16			
ORDER 0.49			
ORDER 1			
ORDER 8			
ORDER 10.71			
ORDER 25			X
ORDER 48			X

Although there are some dissimilarities quantitatively speaking, a good agreement can be deduced from

Table 5-A.

2.1.g Conclusions

The methodology can be applied to EV and HEV as demonstrated. One of the aims of this study is to check the versatility of these methods, commonly used in petrol engine cars, in electrical vehicles, which has been successfully fulfilled.

Technical goals.

Two different sets of results have been presented; the noise contribution spectrum of each path and the path noise impact for some harmonics due to physical components of the electrical vehicle.

One of the conclusions extracted from this study is that TPA results are more easy to realise when an error occurs than OPA ones. The error in the TPA is observed in the computed acoustic pressure which can be compared with the measured one having and instantaneous feedback. Nevertheless, the error of not considering paths in the OPA is computed to the transmissibility functions, and then transmitted to the computed acoustic pressure. Thus, although the last appearance of the acoustic pressure is really good, this does not mean that all the paths are considered.

The OPA results give a good approximation in a short period of time whilst to obtain the TPA results a longer and more arduous procedure has to be performed.

Another conclusion is inferred from the study, although the more number of accelerometers, the more accurate results, a good agreement with the real situation has been obtained with eight accelerometers.

2.1.h References

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