# Study of Engine NVH in a passenger car interior through in-situ TPA method

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## Abstract

In-situ TPA method, which belongs to the classification of component based TPA methods, allows the characterization of a source in the assembled state. In this paper the successful implementation of experimental in-situ TPA to synthesize structure borne vehicle interior noise considering engine as a major source in a hatchback passenger car is described. Considering engine mount locations as transfer paths, the required Noise Transfer Functions (NTFs) between the paths and targets and the local structural Frequency Response Functions (FRFs) between paths and indicators, are determined by conducting laboratory tests and successively the operational data at all points are measured at various pedal pressure conditions. Lastly, path contribution analysis is carried out from the TPA results to identify the critical paths for the critical frequencies. Siemens Simcenter testing solution is used to carryout TPA and LMS Scadas mobile data acquisition system is used to collect the data required for the same.

## 1 Introduction

In the automotive industry, NVH (Noise, Vibration, and Harshness) technology is becoming more important at all times in order to meet the acoustic performance requirements of the customers. To meet these requirements, methods for predicting and optimizing NVH problems are developed. Transfer path analysis (TPA) is a wide range diagnostic tool for the assessment of noise and vibration. In 1950 – 1960, many publications document on isolation of ship engines to minimize the transmission through the interfaces by means of decoupling mechanisms and absorbers[1] were studied. Transfer path analysis is a vastly used state of the art method in diagnosing the vibration problems. TPA studies the system of actively vibrating component (source) and the transmission of these vibrations towards the passive component (receiver). Several other such methods are being developed to predict the vibration levels at specific sites in the coupled systems. Of all other evolving methods, TPA has brought in a paradigm shift and is being largely commercialized to cater the various industrial needs in solving vibration transmission problems. One of the advantages of TPA is that it can be used even when it is difficult to measure the vibrations on the passive side. This is done either by source characterizations or using transmissibility-based concepts. Classical methods like direct force measurements, matrix inverse, mount stiffness methods and operational TPA require measurements to be performed when there is a change in the passive side. So, these methods do not define the source independent of the receiver. This challenge of characterizing the source in TPA has led to development of several other methods and of which the most popular is the component based TPA.

# 2 Literature

## 2.1 History

Over the decades many methods were developed that has led to a quick expansion of TPA concepts and applications. The initial technique, now-a-days referred to as classical TPA, is actually the productive work of Verheij., developed in 1980. He looked at vibration transmission via resilient mounts in ship machinery which led to further exploration of classical TPA [2]. Though there are many theories, Verheij., was the first to succeed in determining interface forces and moments through experimentation. This method was attractive in academic point of view but for the real time engineering, a less complex method for estimating the forces is desired.

A general method called GTDT - abbreviated as Global Transfer Direct Transfer was proposed by Magrans., in 1981 for measuring the transmissibility between the network terminals[3]. This was further developed by Guasch., and later termed as Advanced TPA. Many researchers have contributed their work in determining the properties of transmissibility matrices, which help in developing the transmissibility-based method, also termed as Operational TPA. This method was presented first by Noumura., in 2006 [4].

To depict the transfer of vibration, researchers Mondot and Petersson., in 1987 put forward a method using characteristic power of the source & coupling function involving the receiving structure dynamics[5]. This led to exploring the idea of source characterization by blocked forces or free velocities [6], [7], [8] that was proposed by Elliott (2008) [9] and Moorhouse (2009) [9]. An alternate method to these is a pseudo force method developed by Janssens and Verheij., [10], [11].

To assemble the dynamics of sub-structure, Dynamics sub-structuring technique is partially useful. For this mostly the admittance of either the source or receiver or assembled structure is required. The first dynamic sub-structuring was exercised in 1960, to evaluate the natural vibration modes and frequencies of aircraft structures. With the development of Frequency based Sub-structuring (FBS) [4], it was made possible to assemble multiple substructure FRFs either by numerical or experimental values. This makes the dynamic sub-structuring theory very usable for TPA analysis.

Applications of TPA are being widely adopted in automotive, aerospace, marine, building acoustics, bridges, musical instruments, and other mechanical machineries used in industries. The developed TPA methods are now able to provide solutions for secrecy, safety and comfort.

### 2.1.1 General Framework of TPA

Generally, TPA involves the following steps:

- Operational measurements: The purpose is to measure accelerations when the source is in operation
- Passive measurement: The determination of coupled system (source and receive) characteristics (FRF’s) when the source is turned off.
- Estimation of interface loads between the source and the receiver when the source is in operation
- Perform detailed analysis on the potential paths contributing for vibration transmission.

This is illustrated in the following Figure 1 for classical, component based and transmissibility based TPA procedures and the first two steps of the operational measurements and passive measurements need to be evaluated before the steps 3 and 4. It can be observed that the main difference between the classical TPA and the other families is that the source needs to be isolated to determine FRFs.

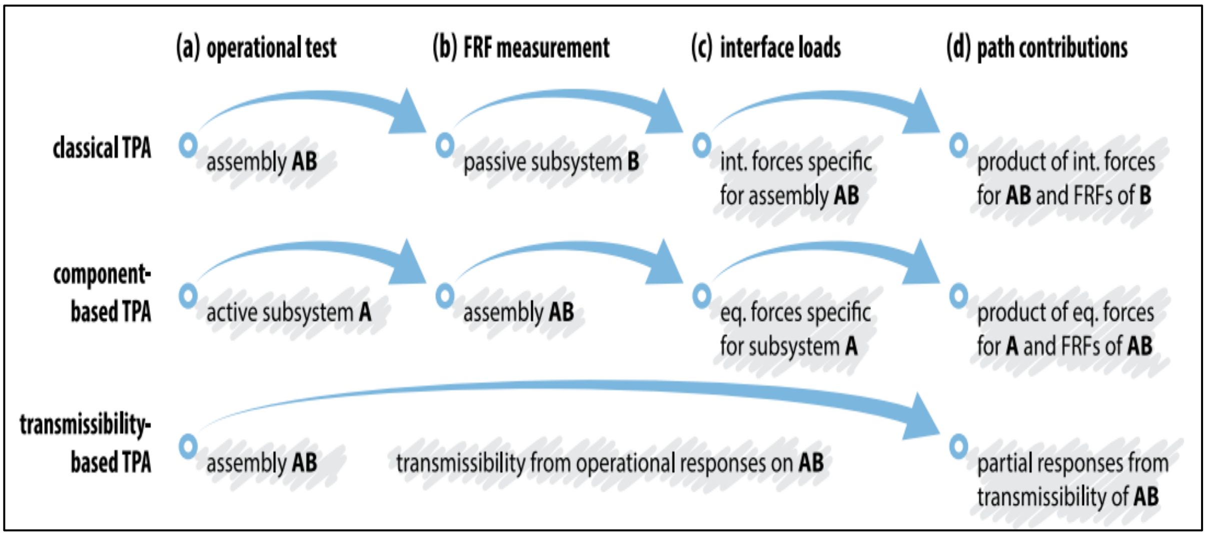


Figure 1: Illustration of various steps in TPA and its classification into three families. [4]

### 2.1.2 Classification of TPA

Based on the methods of evaluation, TPA is classified into 3 families – Classical TPA, Component based TPA and Transmissibility based TPA. The flowchart Figure 2 gives an overview of different methods in the families and the type of forces used for evaluation.

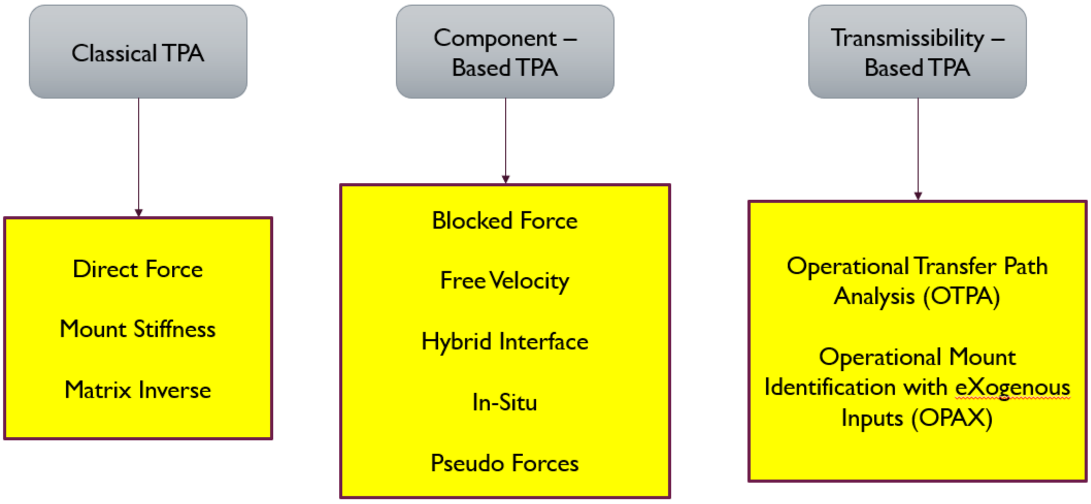


Figure 2: Different methods followed in TPA families.

The classical TPA family estimates the interface force by different methods and these forces can be used to represent the source in the corresponding assembly. In the Component based TPA, equivalent forces and pseudo forces are estimated. Equivalent forces are a special case of pseudo forces and much of the methods are based on the equivalent forces' evaluation. The Transmissibility based TPA is used for path contribution analysis based on the partial responses of the assembly and hence mostly skips the force evaluation.

### 2.1.3 Component based TPA

When a previously designed source is being used in a new receiving structure, then the vibrational behaviour on a receiving subsystem can be predicted without performing the new operational tests[12]. This is possible by defining the forces at the source-receiver interface independent of the receiving structure and thereby making it possible to plug in any new receiving structure. So, defining the source by transferable interface forces is termed as source characterization. The component based TPA characterizes the source by defining the source excitations in terms of equivalent forces or velocities. The responses on the receiving structure are evaluated by applying the forces with the corresponding FRFs of the new receiver thereby eliminating the need for operational measurement. This is where the real advantage of component based TPA can be noticed, the FRFs of the new receiver can be the predicted either from finite element method (FEM) or from a real time measurement. This approach helps the product development team to predict the vibration level of their source on a new receiver even if it is not physically available. Thus, lowering down the development time and as well as the design iterations [12]. The history or the roots of the component based TPA theory is diverse as different researchers in their respective domain have developed the methods with different approach. Hence these methods are slightly independent.

### 2.1.4 In–situ TPA

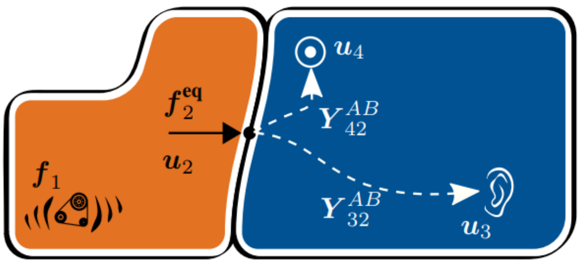


Figure 3: Equivalent force evaluation in in-situ method. [13]

For achieving $\boldsymbol{f_2^{eq}}$, methods such as blocked force method (which assumes that the receiver structure is a fully rigid with infinite stiffness), free velocity method (which considers that the active part is freely suspended) have evolved. As both these methods pose their limitations in practice, one often prefers to conduct operational tests on an appropriate support structure. A hybrid interface method, which forms the sum of contributions of both the blocked force and the free velocity experiment, (also called as Robin boundary condition in a mathematical sense) was published by Klerk et.al., [14]. Although generally applicable in theory, it should be mentioned that the method can be rather costly and time-consuming in practice, as one needs to explicitly measure collocated forces and motion in all directions for every interface node [15]. This method derives the equation, where $B$ is the receiving structure, $A$ is the active sub-system, $u_2$ is the operational responses, $f_2^{eq}$ is the equivalent forces and $Y_{22}{}^{B}$, $Y_{22}{}^{A}$ are the admittance FRF matrices.

$$f_2^{eq} = [Y_{22}^{B}]^{-1}u_2 + [Y_{22}^{A}]^{-1}u_2 \tag{1}$$

According to the coupled system mobility[12], equation (1) can be written as,

$$f_2^{eq} = [Y_{22}^{AB}]^{-1}u_2 \tag{2}$$

To attain over determination, few other points on the receiving structure are chosen as indicator sites. The indicator responses in the coupled system are used for the evaluation of the equivalent forces (illustrated in Figure 3). Though obtained from different derivations, this approach was first proposed by Moorhouse [16] and Elliot., [9].

$$f_2^{eq} = [Y_{22}^{AR}]^{+1}u_4 \tag{3}$$

In the evaluation of the interface forces, the most complicated part is the computation of the inverse FRF matrix. The '+' is the Moor-Penrose pseudo inverse, which uses singular value decomposition technique to evaluate the inverse of any order (nxm, n,m Є N+) for more on the inverse refer to Thite et.al, [17] and Choi et.al [18].

Condition number defines how sensitive is the compound to the inverse measurement errors, generally a lower condition number is preferred. To have a check on the number of variables independently defined, rank of the matrix is used. A rank greater than or equal to the number of unknown forces is favoured. A singular value decomposition (SVD) is used to decompose the complex matrix into simpler matrices with matrix diagonalization [19].

The number of independent responses on the passive side should be sufficient to define the total number of degrees of freedom at the interface. The response of the sensors on the passive side is used for the evaluation of the interface forces, these sensors are termed as indicator sensors. The number of indicator sensors should be equal to or more than the degrees of freedom of interface forces to be estimated. Though there is no strict constraint on the number of indicator sensors, the thumb rule of at least two additional indicators is followed to effectively use the available sensors to achieve better results. The indicator sensors are placed in the proximity of the interface sites to capture more vibration information. Commonly tri-axial accelerometer sensors are used.

Important conditions to note before applying the in-situ method:

- Operational excitations are within the domain of the active component. Any excitations from the passive side will disturb the determination of the equivalent forces.
- The responses used in matrix inversion are located randomly along the transfer paths on the receiving side. However, they are bound to give good results when located close to the interface domain.
- While choosing the sensor positions it is good to note that the vibrations at the active side are caused by interface forces and direct source excitations.

The in-situ, as the name suggests direct measurements are made at the sites without decoupling the systems. This saves a lot of time in practice, thus attracting the interest of many automotive and structural dynamics engineers. Hence, several numerical and experimental studies are made to characterize the source by in-situ method by Verheij [2], Elliot [9], Van der Seijs [15], Thite [17], Choi [18]. More detailed information about applying the in-situ method is suggested in the ISO 20270:2019 norm [20].

### 2.1.5 Case studies on in-situ TPA

Sakthivel. P [22] carried out research on the interior vehicle noise due to tire/ road interaction. Structure borne (<500Hz) interior vehicle noise was synthesized using TPA methods such as conventional and component-based methods (free velocity and in-situ methods). Multi-reference TPA using principal component analysis was done considering interface at spindle as well as tire contact patch. Matrix inversion method was used for estimating the excitation of road input. In comparison of different methods, except at low frequencies, relatively less deviation has been observed between virtual sound pressure levels and the measured sound for different speeds during road tests. Ahmed El. M et.al., worked on the source characterization through in-situ for engine-transmission unit NVH analysis. For two separate configurations of the rubber engine mounts of the vehicle, soft and stiff, equivalent forces are compared and used in prediction of net vibration responses of the vehicle. Cross validation is done by predicting the response in with other configuration (stiff structure) by taking the identified blocked forces of a different configuration (soft rubber mounts), because the internal excitation shall remain unchanged. For 1000 rpm, in steps of 1000 rpm, to 4000 rpm blocked forces were determined. The non-linear behavior of the soft mount and the offset among the blocked forces that were identified are the probable reasons for slight deviations in the comparative plots of the blocked forces at various conditions [13].

# 3 Application of In-situ TPA

The Figure 4 goes into much detail of the workflow of the in-situ method employed for the engine NVH of the Vehicle. As stated, to realize the advantage of In-situ method, in the laboratory, indicator accelerations and the target sound pressures (for local structural FRFs & NTFs) are measured exciting at the engine mount locations (three mounts with 3 translational DOFs each) without separating engine from the vehicle with engine power off condition. Then with different engine operating conditions the same measured at the same locations. Operational loads are estimated through Principal Component Analysis (PCA) where in the calculated multi reference crosspowers are used to determine virtual referenced spectra. Finally, the structure borne vehicle interior noise due to engine vibration is synthesized from estimated operational loads and NTFs at the targets and is validated with the measured data obtained during operational measurements.

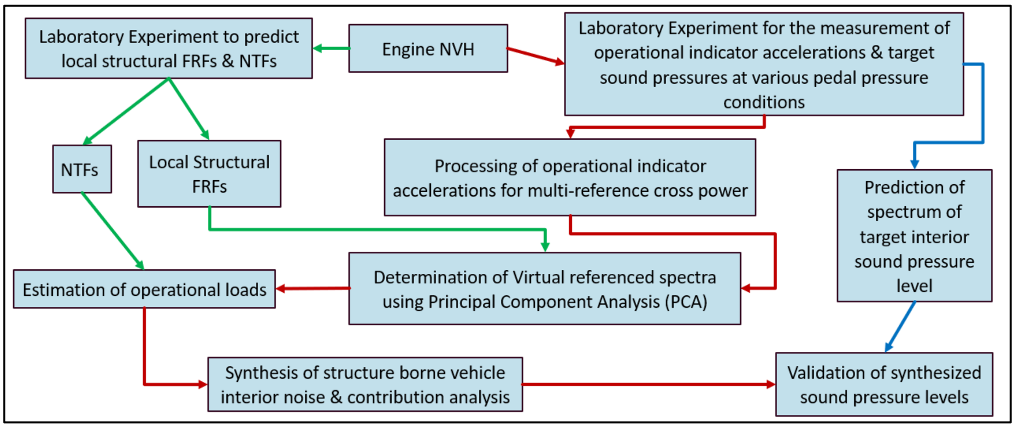


Figure 4: Methodological flowchart of In-situ TPA method employed for engine NVH.

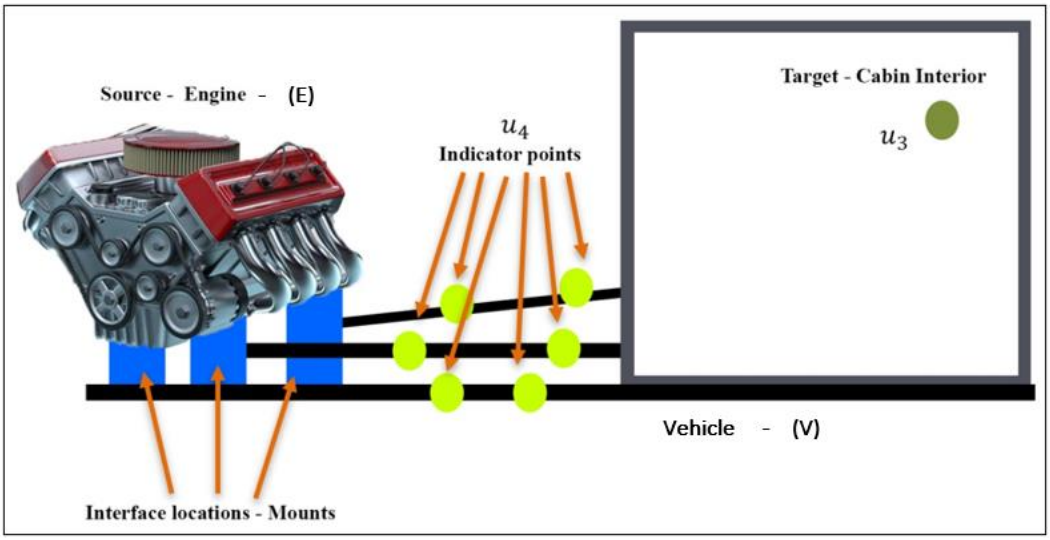


Figure 5: Schematic representation of In-situ TPA for engine noise.

$$f_2^{eq} = [Y_{42}^{EV}]^{+1} u_4 \tag{4}$$

Using the equation 4, the equivalent forces at the interface location are calculated. Where, $E$ is Engine, $V$ is vehicle, the $Y_{42}^{EV}$ is the indicator admittance matrix (also called as indicator FRFs corresponding to the active source excitations) and $u_4$ is the operational acceleration measurements measured at the indicator locations through means of tri-axial accelerometer sensors. Target admittance matrix (NTFs) are also calculated $Y_{32}^{EV}$. Target sensor responses $u_3$ are also measured during operational condition through microphones for validation.

$$u_3^{In-situ\,TPA} = Y_{32}^{EV} f_2^{eq} \approx u_3^{measured} \tag{5}$$

The indicator points are chosen accordingly as per the Figure 5 such that the indicator points must be located along the transfer path. There are three mounts on which the engine is being supported are taken as interface locations. The indicator locations considered for this work along with paths (Engine mounts) are on cradle member-in bonnet (front left (1), front right (4)), above Suspension-in bonnet (front left (2), front right (5)), on rail-in cabin (front left (3), front right (6)), on firewall-in bonnet (left (7), middle (9), right (8)) and in total 9 locations and the same can be seen in Figure 6 along with the target location microphones.

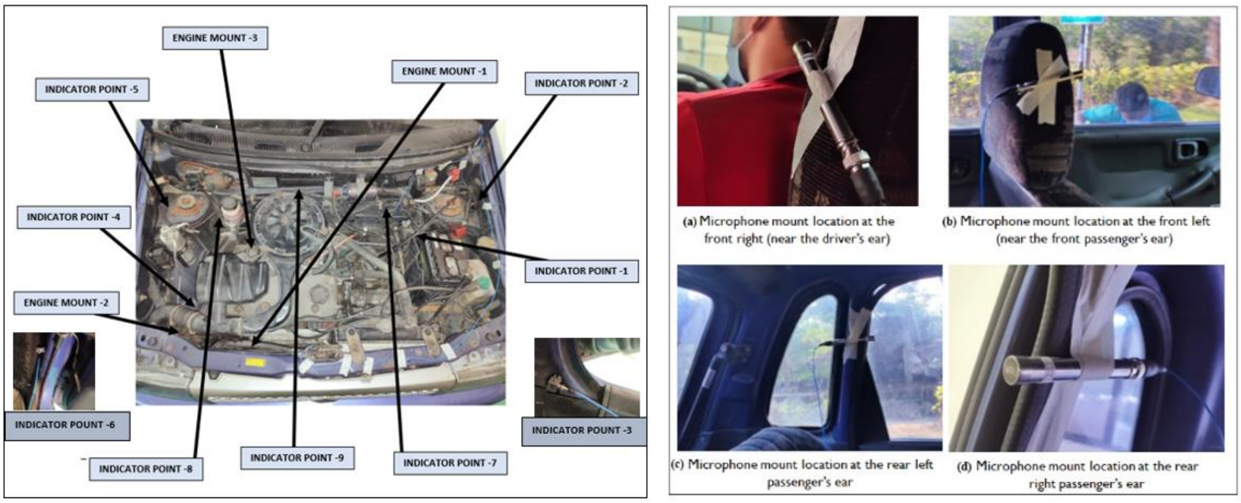


Figure 6: Indicators and target microphones locations for In-situ method on Maruti Zen

The 16-channel LMS Scadas data acquisition system (DAQ) is used to acquire the data from the sensors to the Siemens Simcenter software for processing. The laptop is connected to the DAQ system with a help of a LAN cable. 3-Tri-axial accelerometers, 1-impact hammer, 4-microphones were simultaneously channelled during the experiments. The numbers of measurements were optimized intelligently before the start of the experiments. The tool 'Signature Acquisition Testing – Advanced' is used for operational measurement. The tool 'Spectral Testing' is used for impact hammer excitation. The vehicle was lifted as shown in Figure 7 with the help of lift post to reach out all three mount locations easily for the spectral test.

The operational measurements are done in the laboratory at various pedal pressure conditions such as idling, 25% pedal pressure, 50% pedal pressure and a run up condition. Simcenter Testlab. software of version 18 was used for the acquisition of the data and post processing. The multi-reference matrices are generated for principal component analysis. The TPA model tool is defined. These operational data are combined with local structural FRFs of the vehicle to estimate the operational loads at the transfer paths using matrix inversion method. Then, the NTFs are multiplied with operational loads to synthesis the contribution of individual paths to the total structure borne vehicle interior noise caused due to engine vibration. Lastly, path and vector contribution analysis are carried out from the TPA results to identify the critical paths for the critical frequencies.


Figure 7: Experimental set up for spectral test.

## 4 Results and discussion

To ensure the quality measurements, the drive point coherence always witnessed during spectral test to realize the responses measured are due to the given input excitations. Figure 8 shows the drive point coherences at the interface locations in X direction for mount 1, X direction, in Z direction for mount 2 and in Y direction for mount 3 for the excitation in the respective direction for the respective mount. Each mount has shown a better drive point coherence in any/either its longitudinal or vertical or lateral directions.

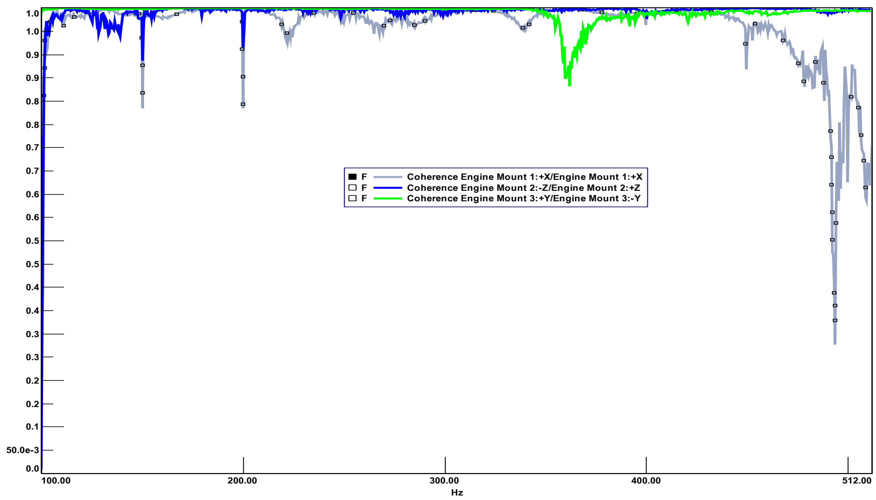


Figure 8: Sample of drive point coherences.

Figure 9 shows the frequency response functions due to the impact excitation at mount - 1 location in all three directions. Such data is collected for all indicator points. These are the elements of $Y_{42}^{EV}$ matrix. It is a matrix of order 36x9. Interesting observation is that excitation in longitudinal directions (X direction) do contribute to lateral (Y direction) and vertical (Z directions) directions not only at a particular mount, but also among the mounts, so that it necessitates formulating multi-reference problem in this work though in general engine TPA is considered as single reference TPA to investigate critical orders.

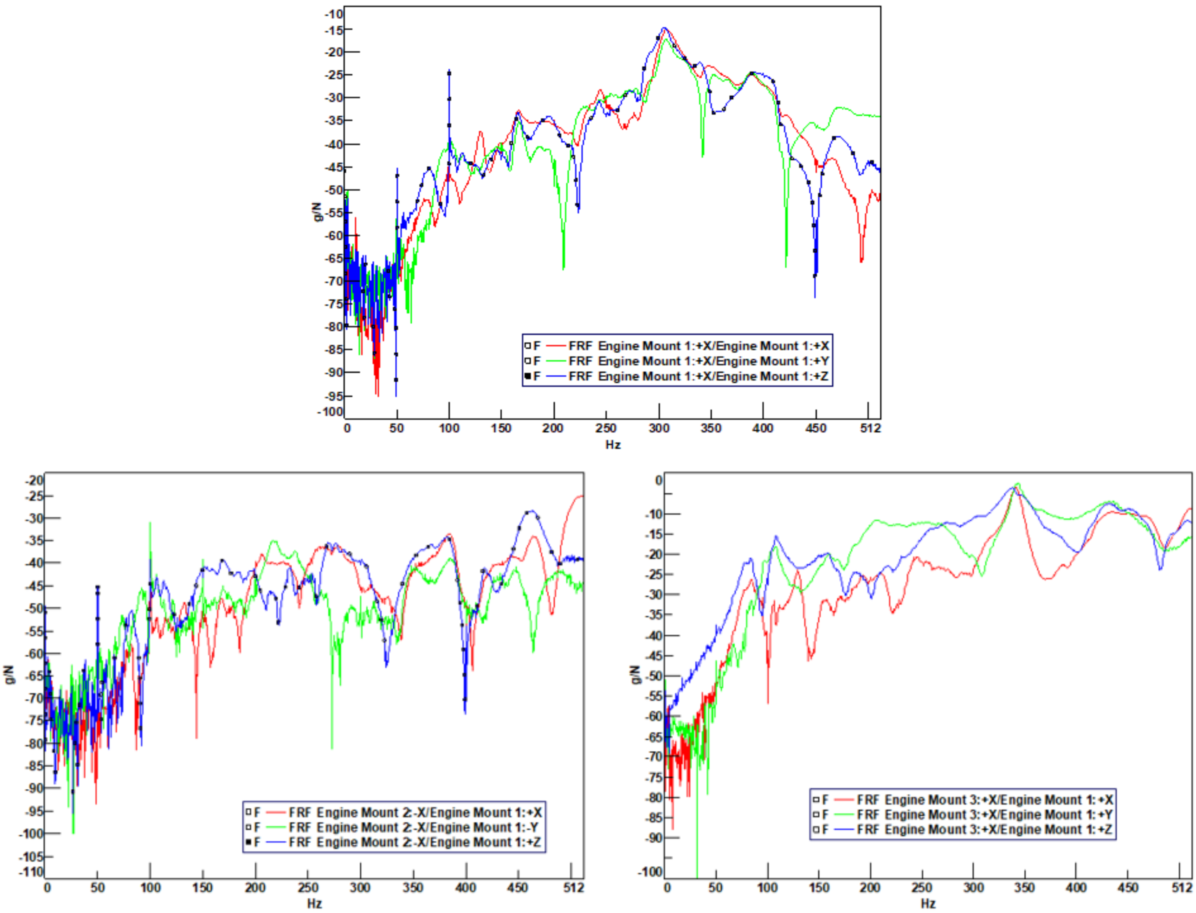


Figure 9: Local structural FRFs in X direction at engine mount 1, 2 and 3 due to the excitation at engine mount 1 at interface locations in all three directions.

Figure 10 shows the variation of noise transfer functions along with the respective coherence functions given as function of frequency due to the impact excitation at mount - 1 location from X, Y and Z directions. Such data is collected for other mount locations excitation as well. It can be observed that the coherence for the excitation in X and Z directions is better than the one obtained from Y-direction. But there is a partial coherence for excitation at mount - 1 in Y direction in the frequency range 350 Hz to 500 Hz. All the NTFs are stacked together to constitute the elements of $Y_{32}^{EV}$ matrix. It is a matrix of order 4x9.

Sample of estimated operational loads (3 out of 9) at engine mount 1 in all three directions due to various operational conditions are plotted in the Figure 11. The maximum interface loads are observed to be appearing in the idling condition at low frequency (around 45 Hz). These operational loads essentially quantify the source strength. At 50 % pedal operating state the estimated operational loads are negligible and the contribution for synthesized sound mainly structural force transfer characteristics that would amplify. Hence the path vector contribution analysis is carried out for the same and explained subsequently.

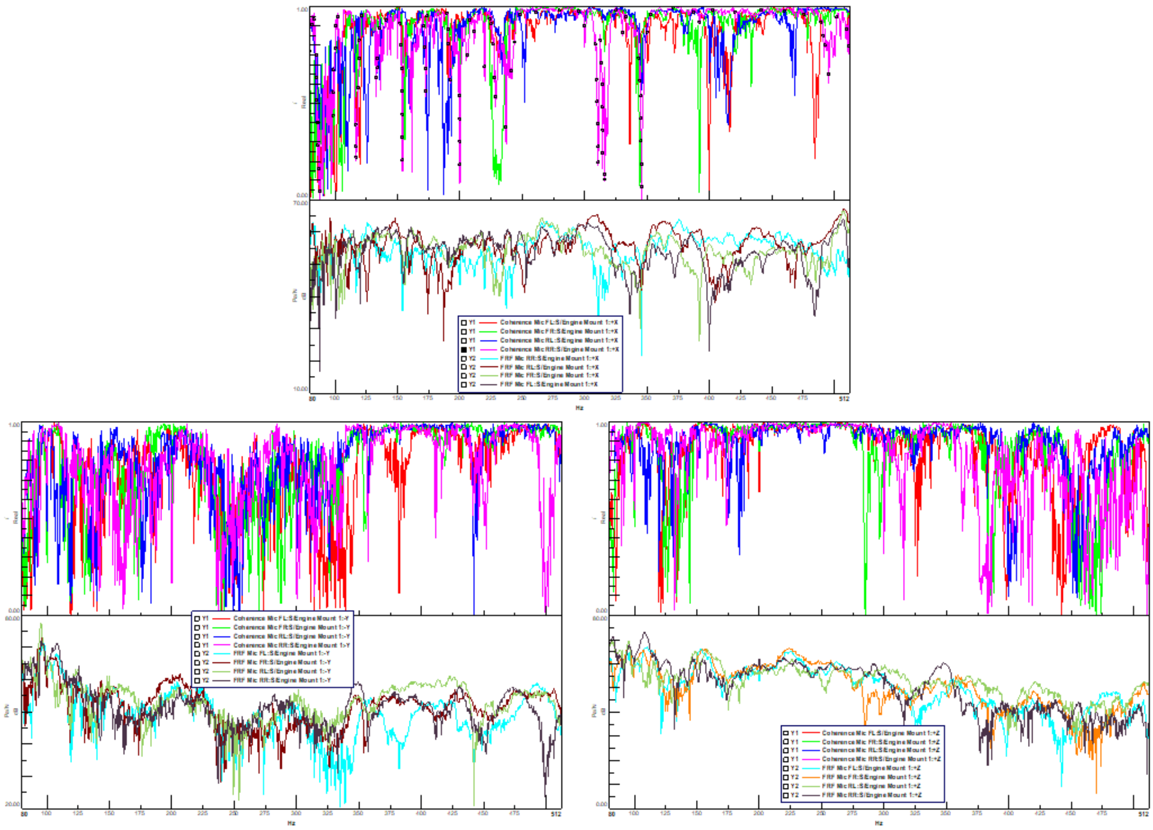

Figure 10: Coherence and the respective NTFs for excitation at mount 1 in X, Y & Z directions.

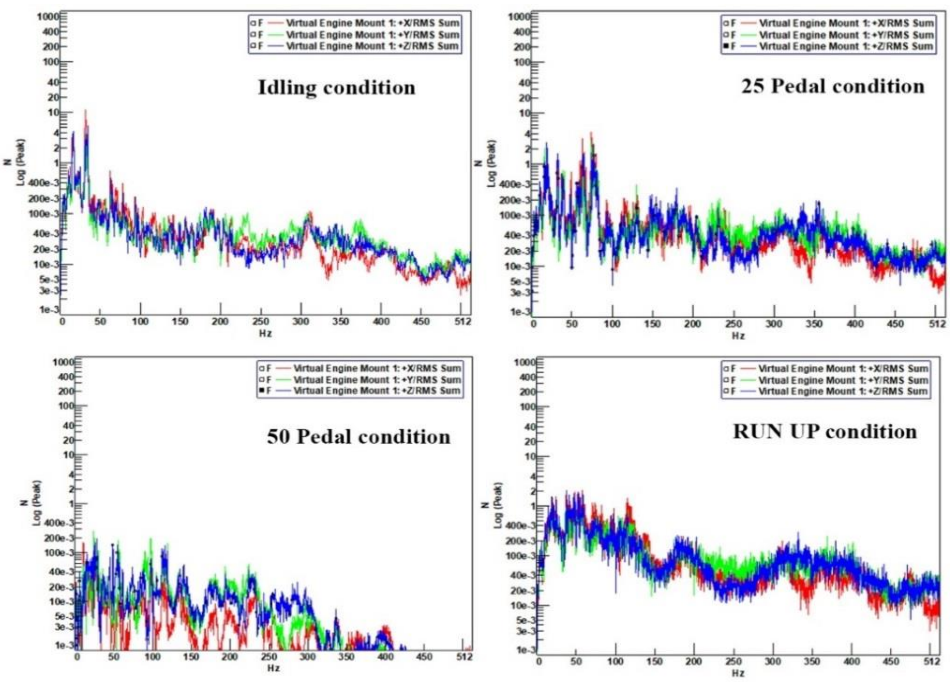


Figure 11: Estimated operational loads at mount 1 for different conditions.

A peak value of 80.51 dB in the interior cabin for idling condition at a frequency of 34.5 Hz is contributed from Z direction of engine mount 3 and can be seen from path contribution plot shown in Figure 12. This contribution plot gives the ranking of individual path contribution towards the total sound pressure level (SPL) in idling condition. It is readily observed, that the contribution from Mounts in X, Y and Z directions are more at lower frequencies (below structure borne noise range) in the Figure 12. The individual path contributions of the mounts are also analysed for the other operating conditions. Also, the path contribution is used to identify major contribution at different frequencies of these paths. But the limitation of this plot is that it does not show the phase relationship between the noise contributing paths and the total interior noise explicitly [5]. Hence, another plot called vector contribution plot can be drown for the given frequency, which would give the phase relationship of transfer paths with total interior noise.

The Figure 13 shows the results of vector contribution analysis at a frequency of 279.88 Hz, for which a sound pressure level amplitude is observed a value of 64.77 dB for 50% pedal pressure condition; it is also inferred from the observation of operational loads from Figure 11 due to minimal loads relative to the other operating condition, the contribution is primarily due to path characteristics. Hence, this frequency (279.88 Hz) was chosen because the sound pressure level at this frequency is higher in the 100 Hz-512 Hz range. From this plot the tail of the vectors placed with origin of the four quadrants helps to distinguish the contributing vectors and non-contributing vectors readily. For example, the plot chosen at 279.88 Hz for 50% pedal pressure condition, 3 out of 9 were non-contributing to the total and in fact, interestingly on one hand, increase in energy transfer along these paths at the chosen frequency would reduce the total sound pressure level and on the other hand, 6 out of 9 were contributing paths the energy transfer needs to be reduced to reduce total sound pressure. It is to be noted that for a chosen frequency this plot would specific and will vary as we look for other critical frequency. Hence, a critical analysis and observation to be made on the estimated loads, quantified transfer paths to decide the critical frequency that may result in contribution of the load as well as the path for total.

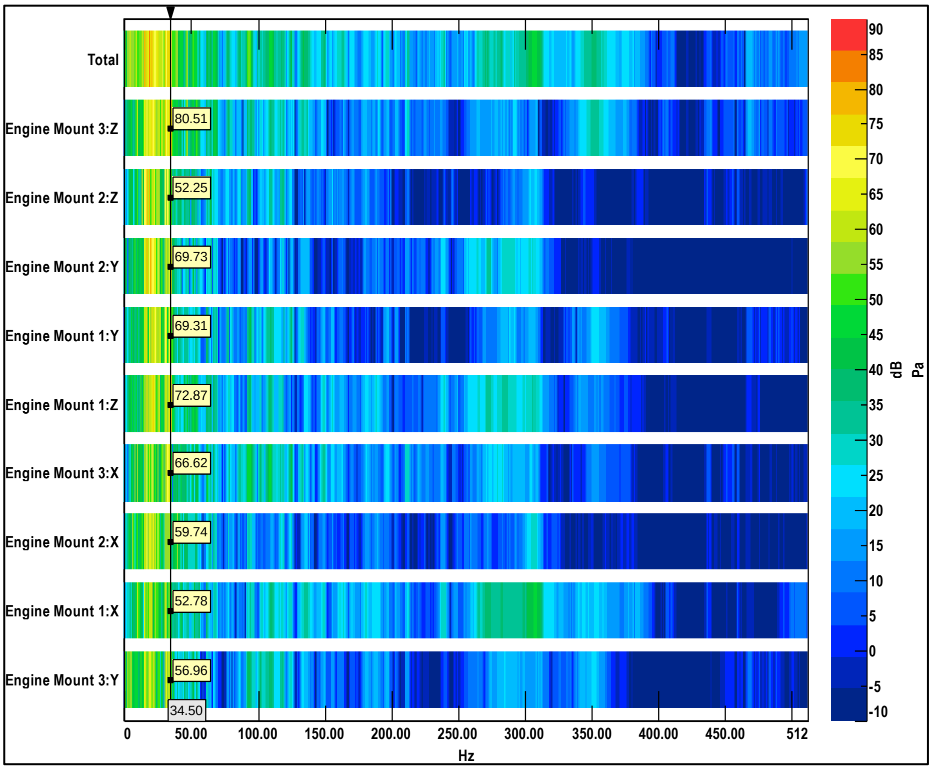


Figure 12: Individual path contribution results for idling condition for PRRM:1.

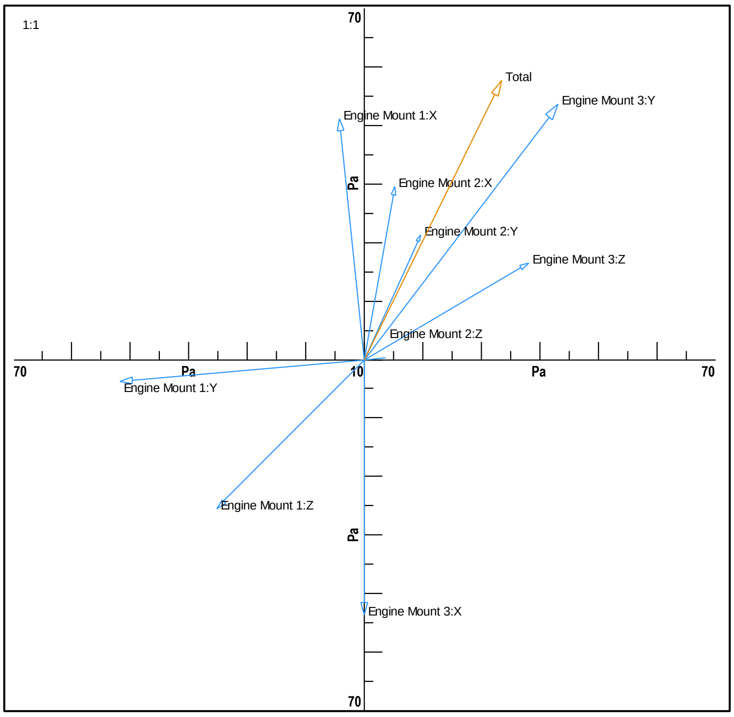


Figure 13: Vector contribution analysis at frequency 279.88 Hz for PRRM:1 for 50% pedal pressure condition.

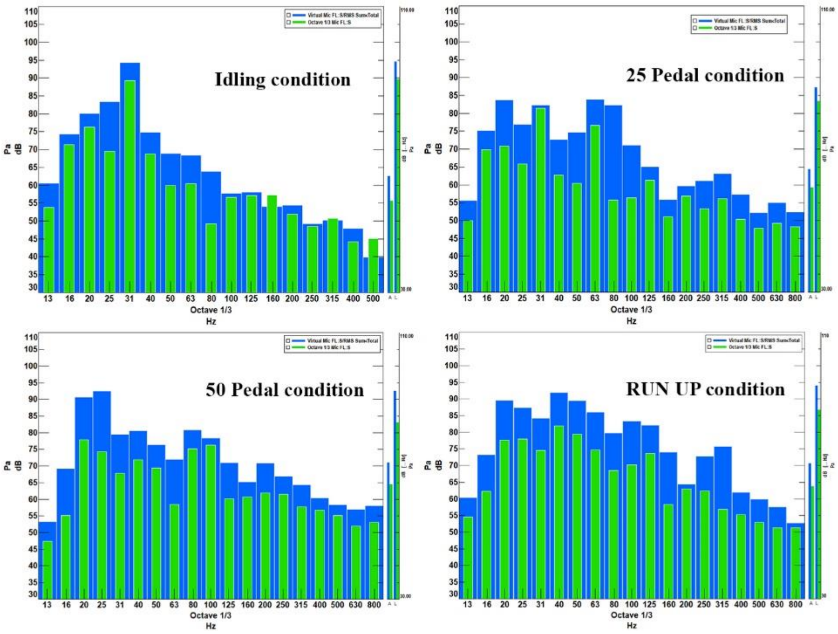


Figure 14: Comparison of estimated sound vs measured sound for Front Left microphone for different operating conditions.

Figures 14 shows the comparison of the results of synthesized sound pressure level with measured sound pressure levels for front left microphone location for all the considered operational conditions. Similar results were compared for other target locations as well but not reported in this paper. The blue coloured bars are the estimated sound pressure levels through TPA whereas the green coloured bars represent the measured sound pressure levels through the microphone during operation. Between 100 Hz – 500 Hz actual frequency of interest for structure borne interior noise nicely captured through In-situ TPA procedure, in the idling and 50% pedal pressure conditions with minimal difference. Whereas at lower frequencies (<100) the sound pressure levels have difference of 5-10 dB levels. The main reason for this is human error during measurements. The measurements were not simultaneous and total of 4 sets of measurements carried out to get all indicator responses during different operating conditions; the error would have caused during 25 % and run up conditions as it was manually provided throttle.

# 5 Conclusion

Maruti Zen, a hatchback class passenger, car's structure borne interior noise is synthesized at four interior positions successfully, by formulating multi-reference in-situ engine transfer path analysis problem. The critical paths are identified through path and vector contribution analysis. Variation in sound pressure level of 5 to 10 dB is observed between the synthesized and the actual measurement for frequency of interest 100 Hz to 500 Hz is attributed due to manual error in holding the pedal position for the operating states 25 % pedal position and run up condition respectively whereas the idling and 50% pedal position conditions were not involved error during repeated measurements. Coherence analysis helps us to understand the quality of the measurement and as a whole correctness of experiments conducted. The variation in sound pressure level between synthesized and measured data is observed, the trend in variation in both the cases is same and interesting to note.

This study was limited to carrying out laboratory tests alone to account only the engine source. However other important sources such as tire/road interaction, transmission noise, wind noise etc. also do contribute to the cabin. In real environments, riding speed is typically related to traffic conditions, road profiles, imposed speed limits, driver behaviour. Thus, increasing the measurement duration and running tests in real time will give more accurate results to assess human perception of noise inside the cabin.

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# Appendix

## A Nomenclature

| | |
|---|---|
| DAQ | Data Acquisition System |
| dB | Decibels |
| E | Engine |
| $f_2^{eq}$ | Equivalent forces |
| + | Moor-Penrose pseudo inverse |
| FBS | Frequency based Sub-structuring |
| FEM | Fine Element Methods |
| FRF | Frequency Response Function |
| g | Acceleration responses ($m/s^2$) |
| GTDT | Global Transfer Direct Transfer |
| Hz | Hertz |
| LMS | Leuven Measurement Systems International |
| $N^+$ | Positive Natural numbers |
| NTF | Noise Transfer Function |
| NVH | Noise, Vibration and Harshness |
| OPAX | Operational Mount Identification with eXogenous Inputs |
| OTPA | Operational Transfer Path Analysis |
| rpm | Revolutions per minute |
| s | Second |
| SPL | Sound Pressure Level |
| SVD | Singular Value Decomposition |
| TPA | Transfer Path Analysis |
| $u_2$ | Operational responses at the interface locations |
| $u_3$ | Operational responses at the target locations |
| $u_3^{measured}$ | Measured operational responses at the target locations |
| $u_3^{In\text{-}situ\ TPA}$ | Synthesized operational responses at the target locations |
| $u_4$ | Operational responses at the indicator locations |
| V | Vehicle |
| $Y_{22}^A$ | Admittance FRF matrix at the interface locations for the active structure |
| $Y_{22}^B$ | Admittance FRF matrix at the interface locations for the receiver structure |
| $Y_{32}^{EV}$ | Admittance FRF matrix at the target locations for the assembled sub system |
| $Y_{42}^{EV}$ | Admittance FRF matrix at the indicator locations for the assembled sub system |
| $\in$ | Belongs to |