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How should acoustics adapt to meet future demands?

A novel path contribution analysis method for test-based NVH troubleshooting

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ABSTRACT

Fifteen years of NVH applications make Transfer Path Analysis (TPA) appear a commodity tool. Despite the fact that TPA is used in a large variety of applications today, the method requires an expert in both the NVH problem and all the related signal and system analysis constraints. TPA is proven to be reliable, but the main bottleneck remains the huge measurement time to build the full data model.

For this reason, industry is constantly seeking for simpler and faster methods. One such method is Operational Path Analysis (OPA), which was introduced about two years ago. OPA is a fully-operational method, requiring only operational measurements of the path references (body-side mount accelerations, pressures close to vibrating surfaces, nozzles and apertures, etc.) and target response(s). The OPA method is indeed very time-efficient, but suffers from several limitations leading to false path contributions and wrong engineering decisions. Its major limitations are: (i) cross-coupling effects between path references which may lead to faulty interpretations, (ii) potential errors due to missing paths and (iii) numerical ill-conditioning problems related to the estimation of transmissibilities from operational data. So, despite the fact that OPA is a very time-efficient approach, its benefit is limited in most application cases.

This paper introduces a novel path contribution method which combines the advantages of classical TPA and OPA. The method is based on simplifications that allow balancing path accuracy and speed of execution. The principles of the method are first outlined. Then, the method is compared with the existing TPA and OPA methods using an automotive example.

INTRODUCTION

Transfer Path Analysis (TPA) is an experimental technique for identifying the vibro-acoustic transfer paths in a system, from the active system component(s), generating the structural and acoustic loads, through the physical connections and along airborne pathways, to the target location(s) at the passive system component(s) responding to these loads. The acoustic and vibration responses at the target location(s) (e.g. interior noise, seat vibration, steering wheel vibration) are expressed as a sum of path contributions, each associated with an individual path and load. For example, for a target response $y_k(\omega)$ at point k, this is formulated in equation (1), with $y_{ik}(\omega)$ the path contribution of path i, ω the frequency and n the number of paths:

$$y_k(\omega) = \sum_{i=1}^{n} y_{ik}(\omega) \tag{1}$$

The oldest approach to this problem was to use coherence analysis to assess the inter-relations between the various contributions, with all problems related to separating partially correlated sources [1]. In the late 80's, an alternative formulation making use of a source-system-receiver model was developed, expressing each of the partial response contributions as the result of an individual structural or acoustic load acting at a localized interface, and a system response to this interface load [2,3]. This effectively corresponds to cutting the global system at the interface into an active part generating the interface load and a passive part reacting to this load. For structural loads, this cut typically corresponds to the physical connection points (e.g. mounts, subsystem connections). For acoustic loads from vibrating surfaces or pulsations from nozzles or apertures, a discretization by omni-directional volume acceleration point sources is typically applied [4,5].

This systems approach allows making explicit each of the partial contributions as the result of a load acting at each contribution location and a Frequency Response Function (FRF) between the load location and the considered target response. This can be expressed as follows:

$$y_k(\omega) = \sum_{i=1}^{n} FRF_{ik}(\omega) * F_i(\omega) + \sum_{j=1}^{p} FRF_{jk}(\omega) * Q_j(\omega) \quad (2)$$

with $F_i(\omega)$ (i = 1,…, n) the structural loads or forces, $Q_j(\omega)$ (j = 1,…, p) the acoustic loads, typically volume accelerations, and $FRF_{ik}(\omega)$ and $FRF_{jk}(\omega)$ the system response functions from the input loads to the target. Concise visualizations of the transfer path contribution results allow to assess critical paths and frequency regions and the separation into loads and FRF's is the key to identify dominant causes and propose solutions (e.g. act on specific load inputs, act on mount stiffness, act on specific system transfer).

The test procedure to build a conventional TPA model typically requires two basic steps: (i) identification of the operational loads during in-operation tests (e.g. run-up, run-down) on the road or on a chassis dyno; and (ii) estimation of the FRF's from excitation tests (e.g. hammer tests, shaker test). The procedure is similar for both structural and acoustical loading cases, but the practical implementation is governed by the nature of the signals and loads.

The measurement of the FRF's between input loads and target response(s) is probably the easiest to control well. The identification of the operational loads is the main accuracy factor. For the structural excitation case, there currently exist three ways to identify the forces [2,6-10].

The first approach is to measure the forces directly by using dedicated measuring devices such as load cells. But such direct measurement is up to now not possible in the majority of cases as the load cells require space and well-defined support surfaces, which often makes application impractical or even impossible without distorting the natural mounting situation.

The second approach is the mount stiffness method which can be used when the active and passive system components are connected through flexible mounts. This approach combines the differential operational responses across the mounts and the mount stiffness characteristics to estimate the transmitted forces. For a mount i, this can be expressed mathematically as follows:

$$F_i(\omega) = K_i(\omega) * \frac{(a_{ai}(\omega) - a_{pi}(\omega))}{-\omega^2} \quad (3)$$

with $F_i(\omega)$ the mount force, $Ki(\omega)$ the mount stiffness profile and $a_{ai}(\omega)$ and $a_{pi}(\omega)$ the active and passive side mount accelerations. The mount stiffness method is a fast method, but its drawback is that accurate mount stiffness data is seldom available and furthermore depends on the load conditions and excitation amplitudes.

The third approach is the inverse force identification method which identifies the operational loads $F_i(\omega)$ (i = 1,...,n) from closeby acceleration indicator responses $a_j(\omega)$ (j = 1,…,v) at the passive system side, by multiplying these with the pseudo-inverse of the measured force-acceleration FRF matrix between all force inputs and indicator responses. Mathematically, this is as follows:

$$\begin{bmatrix} F_1(\omega) \\ F_2(\omega) \\ \vdots \\ F_n(\omega) \end{bmatrix} = \begin{bmatrix} FRF_{11}(\omega) & FRF_{21}(\omega) & \cdots & FRF_{n1}(\omega) \\ FRF_{12}(\omega) & FRF_{22}(\omega) & \cdots & FRF_{n2}(\omega) \\ \vdots & \vdots & \vdots & \vdots \\ FRF_{1v}(\omega) & FRF_{2v}(\omega) & \cdots & FRF_{nv}(\omega) \end{bmatrix}^{-1} \cdot \begin{bmatrix} a_1(\omega) \\ a_2(\omega) \\ \vdots \\ a_v(\omega) \end{bmatrix} \quad (4)$$

The matrix inversion and force identification are done frequency by frequency. The number of indicator responses (v) must significantly exceed the number of forces (n), with a factor 2 as a rule of thumb, to minimize ill-conditioning problems when calculating the pseudo-inverse. The main drawback of this is the need to perform a large number of FRF measurements to build the full matrix. The latter costs a lot of time and is a main bottleneck for industry.

Today, the main driver for innovations in TPA is the industry's demand for simpler and faster methods. Existing techniques like inverse load identification are very time-consuming. Several attempts have been made to speed up the TPA process. One example is the recently developed Operational Path Analysis (OPA) approach [11,12]. This approach attracts quite some attention as it requires only operational data measured at the path references (e.g. passive-side mount accelerations, pressures closeby vibrating surfaces, nozzles and apertures) and target point(s). No FRF's need to be measured. Essentially, it is a transmissibility method as known from structural dynamics, characterizing the co-existence relationship between the target response(s) and path references.

The OPA method is indeed very time-efficient, but has several limitations [13-16]. One of the main limitations is the cross-coupling between the path references. Because of the system's modal behavior, a single force in one of the mounts causes vibrations at all path references. A high reference level does not imply that a force or acoustic load is entering the system. Co-existence of signals does not imply causality. Due to the apparenty simple presentation of the results, this cross-coupling effect hence may lead to a false interpretation of significant paths and wrong engineering decisions. Next to this, the method suffers from ill-conditioning problems related to estimating transmissibilities from operational data. These problems lead to unreliable transmissibility estimates in many situations. Finally, there are potential errors due to missing paths in the analysis. It has been shown through simulations that the contributions of missing paths are distributed over the other ones, introducing errors which are hard to recognize. Due to the backward-forward use of the same data, a good synthesis of summed contributions is not representative for completeness and quality of the results.

This paper introduces a novel TPA approach, which combines the efficiency of the operational path method and the effectiveness of the existing conventional TPA methods. The principles of the new TPA method are first outlined, after which an automotive example is presented, comparing the new method with the existing mount stiffness and inverse force identification techniques.

NOVEL TPA METHOD USING PARAMETRIC LOAD MODELS

The novel TPA approach differentiates from the existing ones in the identification of the operational loads. Key is the use of parametric models characterizing the operational forces and acoustic loads as a function of measured path inputs such as mount accelerations and acoustic pressures. The parametric load models are estimated from (i) in-situ measured operational path inputs and target response signal(s) and from (ii) transfer path FRF's using mathematical techniques, like for example a Least Squares (LS) estimation approach. Extra acceleration and/or pressure indicators can be included in the set of equations to obtain more robust parameter estimations, but this is optional. The big advantage here lies in the fact that the source, no longer needs to be removed from the model to do some additional time-comsuming FRF measurements used for the operational force estimation with the inverse force identification method in classical TPA.

A schematic representation of the different variables to be measured and identified is given in figure 1. The figure presents a system with an active part generating forces $F_i(\omega)$ ($i = 1,\dots, n$) and acoustic loads $Q_j(\omega)$ ($j = 1,\dots, p$) and with a passive part reacting to these loads. A typical example of such a system is a vehicle body on which a powertrain is mounted. In this example, the powertrain forms the active part of the system, while the vehicle body with passenger compartment is the passive component.

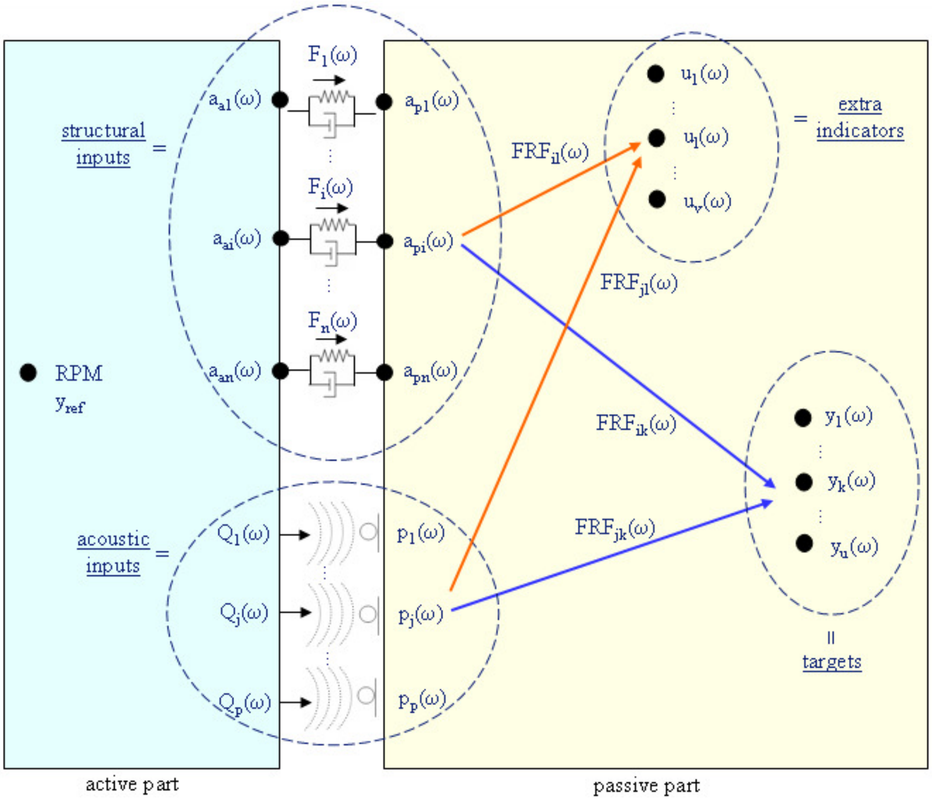


Figure 1: Schematic representation of a system with an active component generating forces and acoustic loads and a passive component responding to these loads.

In order to identify the noise transfer paths from the active system part, through the physical connection elements and along airborne pathways, to the target point(s) of interest, the following variables are typically collected:

- Operational responses during run-up and/or run-down:
  - Target(s): pressures and/or accelerations $y_k(\omega)$ ($k = 1,\dots, u$)
  - Extra indicators if desired: accelerations and/or pressures $u_l(\omega)$ ($l = 1,\dots, v$)
- Operational path inputs during run-up and/or run-down:
  - Structural path inputs: active and/or passive-side accelerations $a_{ai}(\omega)$ and $a_{pi}(\omega)$ ($i = 1,\dots, n$)
  - Acoustic path inputs: pressures $p_j(\omega)$ ($j = 1,\dots, p$) near vibrating surfaces, nozzles, etc.
- Tacho signal (pulse train, RPM) or phase reference during operation
- FRF's from excitation tests:
  - FRF's from load inputs to target(s): $FRF_{ik}(\omega)$, $FRF_{jk}(\omega)$
  - FRF's from load inputs to extra indicators if used: $FRF_{il}(\omega)$, $FRF_{jl}(\omega)$

The TPA method comprises 5 major steps. A flow diagram of the method is shown in figure 2.

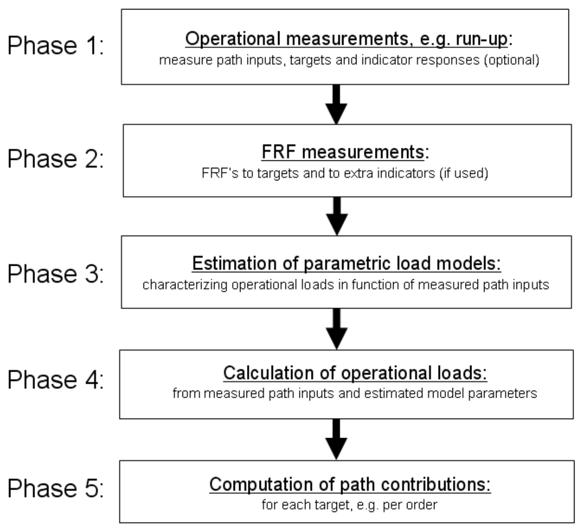


Figure 2: Flow diagram of the novel TPA approach, comprising 5 major steps.

- Phase 1: operational measurements

First, operational measurements are performed. These can be a single run-up or run-down or several of these measurements at different conditions (e.g. various throttles, gears, etc.). It all depends for which condition(s) the TPA model must be developed. During the operational measurements, all mount acceleration and pressure inputs and all responses at the target point(s) and extra indicators are measured synchroneously. Order envelopes (amplitude and phase in function of rotational speed) are then tracked for all measured input and response channels. Strictly, only the orders of interest must be processed. However, the more orders are used for identifying the parametric load models, the more robust the model parameters can be estimated and the more accurate the path contributions can be derived.

- Phase 2: FRF measurements

In a second phase, FRF's are measured between the input loads and target response(s). The FRF's can be measured in a direct or reciprocal way. The use of reciprocal measurements (exciting at the target location(s), measuring the response at the interfaces) has two advantages: (i) only one excitation is needed per target point while the direct approach requires one excitation per input load; (ii) the limited space at the path inputs can lead to direction errors in the direct FRF measurements of up to 10 dB. In case additional indicators are used, the FRF's from the inputs to the indicators must also be measured. It is to be noticed that the sequence of phases 1 and 2 may be changed.

- Phase 3: estimation of parametric load models

Phase 3 is the key step of the method. Parametric load models are estimated, characterizing the operational forces and acoustic loads as a function of the acceleration and pressure path inputs:

$$F_i(\omega) = f(parameters, a_{ai}(\omega), a_{pi}(\omega))$$
$$Q_j(\omega) = g(parameters, p_j(\omega)) \qquad (5)$$

By substituting these parametric load models in the classical TPA formulation (2), equation (6) is obtained. A similar equation can be written for all the additional indicator points.

$$y_k(\omega) = \sum_{i=1}^{n} FRF_{ik}(\omega) * F_i(parameters, a_{ai}(\omega), a_{pi}(\omega))$$
$$+ \sum_{j=1}^{p} FRF_{jk}(\omega) * Q_j(parameters, p_j(\omega)) \qquad (6)$$

The parametric models may be any suitable model describing the loads. A priori known relations among parameters (e.g. mount stiffnesses in x- and y-direction are known to be similar, etc.) may be taken into account to reduce the number of parameters to be estimated and obtain a better conditioning.

For a given number of operational path inputs, response data and FRF's, measured in phases 1 and 2, the above equation gives rise to a linear system of equations that can be solved for the model parameters using conventional mathematical techniques, like for example a Least Squares (LS) estimation approach.

It is clear that the more input information is used, i.e. the more orders, targets and indicator responses, the more accurate the model parameter estimations can be. Furthermore, the use of a balancing factor to scale the order components and the structural and acoustic terms helps improving the parameter estimations.

The estimated model parameters may allow determining additional interesting information regarding the system. For example, the ability of estimating the mount stiffness characteristics from TPA measurement data is an interesting additional feature of the method.

- Phase 4: identification of operational loads

In phase 4, the operational input forces (i = 1,…, n) and volume accelerations (j = 1,…, p) are determined by substituting the obtained model parameter values in equation (5). The loads are typically calculated per order.

- Phase 5: computation of path contributions

Finally, once the operational loads are identified, the path contributions can be calculated for each target point k, by multiplying the loads with the corresponding FRF, as expressed in equation (2). Visualizations of the path contribution results then allow to (i) assess critical paths, orders and frequency regions and (ii) propose modifications of, for example, mount stiffness characteristics, transfer path FRF's, etc.

AUTOMOTIVE EXAMPLE

A TPA analysis was carried out on a 6-cylinder car to assess the novel TPA method and compare it with the conventional mount stiffness and inverse force identification methods. The main focus of the analysis was on the structural noise transfer from the powertrain through the 5 mount connections to the acoustic target at the driver's ear. Airborne contributions from engine, intake, exhaust, etc. were not analyzed in this study. The following data were measured:

- Operational data during engine run-up from 1200 to 6000 RPM
    - Responses: i) 1 pressure target at the driver's seat and ii) 13 extra acceleration indicators at the passive system side nearby the mount connections
    - Path inputs: i) 15 active side accelerations and ii) 15 passive side accelerations (5 mounts in x-, y- and z-directions)
    - Tacho pulse signal
- FRF's from excitation tests:
    - FRF's from 15 load inputs to target: 15 in total
    - FRF's from 15 load inputs to 15 passive side mounting loactions and to 13 extra acceleration indicators: 15 times 28 = 420 in total

Orders (amplitude and phase as a function of RPM) were tracked for all measured input and response channels. Mount stiffness data (in x-, y- and z-direction) were available for all mounts. A simple and a more complex mount model were used for the novel TPA method.

A TPA analysis was done for order 3. This order causes a booming noise in the passenger compartment at 4850 RPM. The following four methods were used:

- Existing mount stiffness method:
    - Using mount stiffness data to identify forces
- Existing inverse force identification method:
    - Frequency per frequency force identification

    - Requiring FRF's from 15 input loads to 15 passive side mount accelerations and 13 additional acceleration indicators (420 FRF's in total, overdetermination with factor 2 as a rule of thumb) to identify forces without ill-conditioning problems

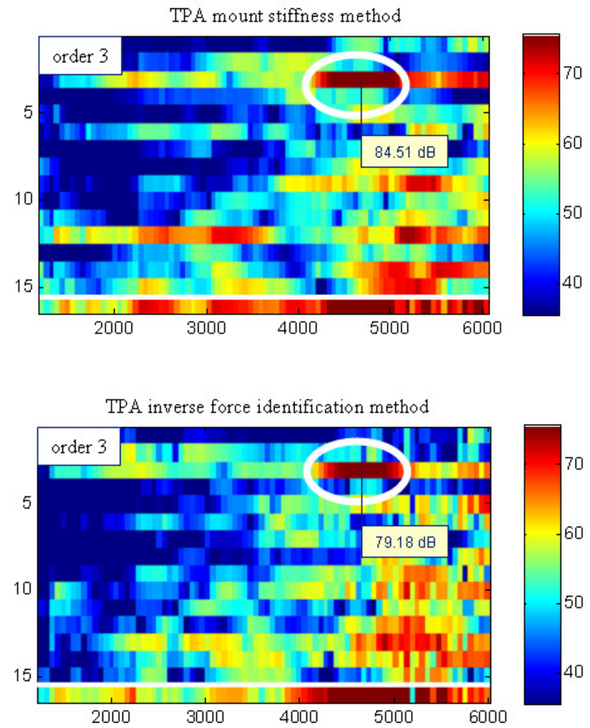
- Novel TPA method using a simple mount model
    - Using FRF's from 15 input loads to 3 acceleration indicators (45 FRF's in total) to identify forces without ill-conditioning problems

- Novel TPA method using a more complex mount model:
    - Assuming constant complex mount stiffness over small frequency bands of 30 Hz
    - Using FRF's from 15 input loads to 13 acceleration indicators (195 FRF's in total) to identify forces without ill-conditioning problems

The path contribution results for order 3 are presented in figure 3. All four TPA methods were very well capable to spot the critical path (mount 1 in z-direction) and frequency region (4850 RPM).

One can also see that the mount stiffness method seems to overestimate the contribution of the critical path at 4850 RPM (+/- 5 dB above the other methods). The identified critical spot is even 3 dB higher than the measured order 3 at the target which is not realistic. This overestimation may be due to inaccurate mount stiffness data.

The advantage of the simple and complex mount stiffness estimator methods is that they are able to correctly spot the critical path without the need to (i) have the mount stiffness data (are seldom available and not always accurate) and (ii) measure the full FRF matrix (huge FRF measurement efforts to build a matrix with overdetermination of factor 2). Only a limited number of FRF's need to be measured as illustrated in this example. In many cases (e.g. multiple orders in the data, full RPM range, etc.), only the FRF's to the target(s) are yet sufficient. This makes the method very fast.



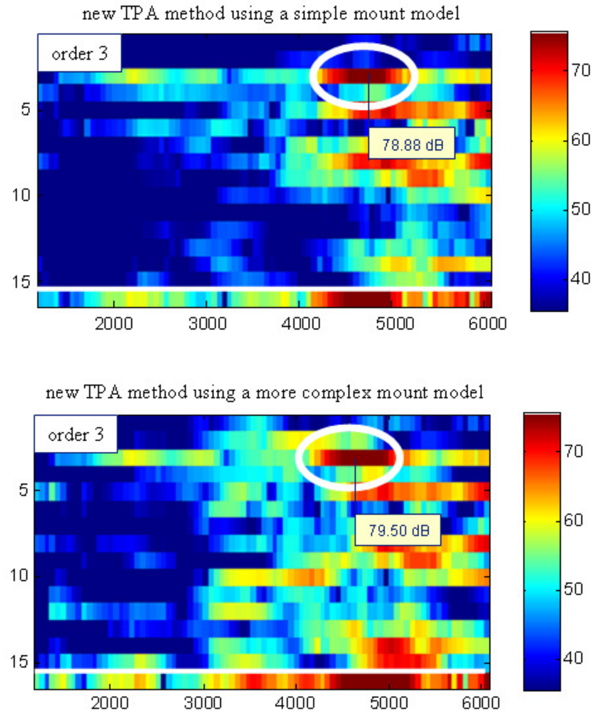


Figure 3: Order 3 path contribution results of the (i) mount stiffness method, (ii) inverse force identification method, (iii) novel TPA method using a simple mount model and iv) novel TPA method using a more complex mount model.

CONCLUSIONS

A new TPA method was developed, combining the speed of the operational path method (OPA) and the effectiveness of the conventional TPA methods. Key is the use of parametric load models characterizing the operational forces and acoustic loads in function of measured path inputs such as mount accelerations and pressures. The parametric load models are estimated from (i) in-situ measured operational path inputs and target response signal(s) and from (ii) transfer path FRF's using mathematical techniques. Extra acceleration and/or pressure indicators can be included in the set of equations to obtain more robust parameter estimations. The proposed TPA method has several advantages:

- The method is fast and accurate.
- It allows balancing between speed of execution and path accuarcy. The more extra indicators used, the more robust the estimations and the better the path accuracy, but the higher the FRF measurement efforts and time.
- The measurement efforts are small in comparison to the existing inverse load identification technique. Next to the operational measurements of path inputs and target(s), the new method requires in many cases only one reciprocal FRF measurement per target point. Adding extra indicators for improving robustness requires additional FRF measurements, but this is still a small effort compared to the huge measurement efforts to build the full FRF matrix for inverse load identification.
- The method does not require mount stiffness data. Such data is seldom available and not always accurate.
- The estimation of the parametric load models is numerically stable. Ill-conditioning problems like those in Operational Path Analysis (OPA) hardly occur.
- The estimated model parameters may allow determining extra interesting information of the system. For example, the ability of estimating the mount stiffness characteristics from TPA measurement data is an interesting additional feature of the method.

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