Numerical Transfer Path Analysis for NVH System-Simulation Models

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Abstract For the assessment of the Noise, Vibration and Harshness (NVH) behavior of complex systems in early phases of the development process, validated modelling methods are available that allow the prediction of structure-borne and airborne noise at system level. However, due to large model sizes, the identification of weak spots in the vibroacoustic transfer behavior and the derivation of optimization measures are highly complex, time-consuming, and mostly not practical. In the field of experimental NVH analysis, transfer path analysis (TPA) has been established as target-oriented methods for identifying vibroacoustic anomalies at system level. In this contribution, TPA methods from the field of measurement technology are selected and applied to numerical NVH system models, aiming for high accuracy and low computational effort in post-processing. The applications of the methods are shown and discussed using an elastic multi-body simulation model of a tractor drivetrain with a transient run-up of the vehicle speed as an example. The classical direct-force and component-based blocked-force TPA methods selected and adapted for this study allow for efficient calculation of the sound contributions of numerical models. At the same time, they overcome typical challenges of experimental TPA, such as exact force determination or consideration of rotational degrees of freedom. In addition, the comparison of the two methods shows that the path contributions are in general different for classical and component-based TPA and only under specific conditions the same. Both numerical TPA methods allow for the identification of weak spots in the NVH behavior in an efficient and target-oriented way.

1. Introduction
Manufacturers and suppliers are increasingly aiming to shorten development times and reduce development costs in order to remain competitive in the market. Long development times and high costs are mainly caused by prototypes, measurements and product changes in late phases of the product development process (PDP). To reduce the number of prototypes and measurements to a minimum and to avoid changes in prototyping phase, methods of virtual product development are currently used in many industrial sectors.

In addition to the central goal of function fulfillment, comfort features such as the Noise, Vibration and Harshness (NVH) behavior are increasingly becoming the focus of attention for customers and consumers and are therefore an important requirement and development goal. The NVH behavior describes all audible and perceptible vibrations on machines and vehicles. According to the machine acoustic transfer chain, sound arriving at a receiver point, e.g. the driver’s ear, is characterized by excitation, transmission and radiation properties. Sound is transmitted along airborne or structure-borne sound paths, the so-called transfer paths. The propagation of structure-borne sound depends on the
structural dynamics of the individual subsystems as well as on their couplings and thus on the system topology. Due to the high dependencies and interactions, the prediction of the NVH behavior requires investigations at system level. In the field of virtual product development, numerous validated modelling methods are available today which provide reliable noise predictions for entire systems up to the medium and higher frequency range (<5 kHz), e.g. [1–4]. However, the interpretation of simulation results and the identification of root causes of acoustic anomalies at system level can be highly time consuming and challenging. The large number of components and couplings involved in a vibration event and the huge amounts of data often make an interpretation and target-oriented analysis impossible or only possible with great effort.

Conventional methods of evaluating NVH results from numerical models can be divided into punctual and distributed evaluation methods. Punctual evaluation methods consider single sensor positions analogous to experimental measurements. For analysis, the signals are typically transformed into the frequency domain and visualized in the form of spectrograms. The selected sensor points often show a high sensitivity with regard to the exact location for evaluation. In addition, individual sensor points usually provide only an insight into a specific section of the system behavior. Therefore, with evaluation of individual points the derivation of optimization measures influencing the global system behavior is difficult. Punctual evaluation methods are, nonetheless, suitable for the verification of requirements at specified interfaces.

Distributed evaluation methods typically consider the entire structure involved in the vibration event. For evaluation, quantities such as deformations, vibration velocities or accelerations are typically used, for example for analyzing mode shapes or operational deflection shapes. Sometimes also quantities such as energy distribution or structure intensity are used for analysis of structure-borne sound. The selected quantities are evaluated either for individual timesteps or individual frequencies. Typically, a large number of evaluation points are considered at the same time. The analysis is hardly manageable with complex three-dimensional structures or many interacting components. Due to the high complexity, the identification of weak spots is very costly and partly not manageable.

In the field of measurement technology, however, target-oriented evaluation methods of analyzing the NVH behavior of entire systems are known: the methods of transfer path analysis (TPA). The aim of these methods is to identify individual sound contributions to the total sound at defined receiver points. The receiver points are typically at positions in the system for which specific sound requirements are defined, such as maximum acceleration levels at specified interfaces. The sound contributions can be allocated to individual sound sources or individual transfer paths. With the knowledge of dominant sound sources or paths, optimization measures can be determined and derived in a target-oriented way. However, the experimental methods are applicable only in later phases of the PDP when hardware prototypes are available. Changes implemented in later phases of the PDP as well as the large number of prototype variants required generate high costs.

In this publication, methods known from experimental TPA are applied for the target-oriented evaluation of NVH simulation models. For this purpose, TPA methods are selected which deliver sound contributions with high accuracy and minimum calculation effort in post-processing. Section 2 shows the principles of TPA and gives an overview of the available TPA methods. In section 3, two suitable methods are selected and the process for determining of sound contributions with the use of numerical simulation models is presented. Section 4 shows the application and results of the methods for an exemplary system model of a tractor drivetrain. The results are discussed in section 5 and a comparison of the sound contributions of the classical and component-based methods is given.

2. Principles of Transfer Path Analysis
The basic idea of TPA is the decomposition of the total sound at an evaluation point into individual sound contributions, also known as paths. Complex vibration phenomena are reduced to a limited number of paths. The TPA considers all properties that are relevant for the NVH behavior. This includes the interaction between excitation and structural dynamics or transfer paths as well as interference
effects caused by phase differences, that result in increasing or decreasing sound amplitudes. The sound contributions help to determine dominant sources or paths in a structured and target-oriented way. The sound contributions are calculated by multiplying forces or sound quantities from operational measurements with measured transmission properties. According to the paradigm of dynamic substructuring [5], the considered system AB is therefore divided, i.e. substructured, into the subsystems A and B, Figure 1.

![System AB](image)

**Figure 1.** General procedure for TPA: substructuring of the System AB.

The interfaces and thus the paths are determined by the choice of the cutting plane. Subsystem A is usually the active side which contains the excitation sources with the excitation forces $f_1$. Subsystem B is the passive side which is excited by system A via several interfaces and which contains the receiver points. The analyzed sound quantity $u$ can be displacements, velocities or accelerations. The complex-valued summation of the sound contributions $u_{3\text{path }i}$ of all $n$ paths results in the total sound $u_3$ at the receiver point:

$$u_3 = \sum_{i=1}^{n} u_{3\text{path }i}$$

The transfer characteristics of the structure are described in the form of the frequency response functions (FRFs) $Y_{21}^A$, $Y_{22}^A$, $Y_{22}^B$ and $Y_{32}^B$. The superscript indices indicate the subsystem and the subscript indices indicate the points between the FRFs are obtained.

Over the last decades various TPA methods have been developed. In [6], an overview and a classification of the methods is given. The methods are divided into three TPA families:

- Classical TPA
- Component-based TPA
- Transmissibility-based TPA

The classical and the component-based TPA are physical methods, whereas the transmissibility-based TPA is based on correlations and statistics. The transmissibility-based TPA offers the advantage of a reduced measurement effort. For application to numerical models, the measurement effort is irrelevant. In this publication, only the two physical methods will be discussed.

The calculation of classical and component-based TPA can be derived using the general approach for substructuring. On the conditions of displacement compatibility $u_2^A = u_2^B$ and force equilibrium $g_2^A + g_2^B = 0$ in the interfaces, the total sound $u_3$ at the receiver caused by the source force $f_1$ can be determined as follows:

$$u_3 = Y_{32}^B (Y_{22}^A + Y_{22}^B)^{-1} Y_{21}^A f_1$$

For both, classical and component-based method, the sound contributions are determined from time-dependent forces and frequency-dependent transmission properties in the form of FRFs. The methods...
differ in how the quantities of equation (2) are combined. The selected combination results in different subdivisions of the overall system for the measurements.

In classical TPA, the forces are determined in the assembled system AB. These forces correspond to cutting forces respectively interface forces. The FRFs are determined in the isolated passive structure B. The product of interface forces $g_2^B$ and FRFs $Y_{32}^B$ provides the path contributions.

$$ u_3 = Y_{32}^B (Y_{22}^A + Y_{22}^B)^{-1} \cdot Y_{21}^A \cdot f_1 = Y_{32}^B g_2^B $$ \hspace{1cm} (3)

with

$$ f_2^B = (Y_{22}^A + Y_{22}^B)^{-1} \cdot Y_{21}^A \cdot f_1 $$ \hspace{1cm} (4)

For the component-based TPA, the equivalent forces $f_{eq}^A$ are determined which depend only on the source structure A. Together with the transmission properties $Y_{32}^{AB}$, determined in the assembled system AB, the path contributions are calculated. The derivation is shown in (5), where (2) is extended by $Y_{22}^A (Y_{22}^A)^{-1}$.

$$ u_3 = Y_{32}^B (Y_{22}^A + Y_{22}^B)^{-1} \cdot Y_{22}^A (Y_{22}^A)^{-1} \cdot Y_{21}^A \cdot f_1 = Y_{32}^{AB} f_{eq}^A $$ \hspace{1cm} (5)

with

$$ Y_{32}^{AB} = Y_{32}^B (Y_{22}^A + Y_{22}^B)^{-1} \cdot Y_{22}^A $$ \hspace{1cm} (6)

and

$$ f_{eq}^A = (Y_{22}^A)^{-1} \cdot Y_{21}^A \cdot f_1 $$ \hspace{1cm} (7)

As shown in equation (7), the equivalent forces are dependent only on the active system A.

In experimental TPA, the determination of forces is challenging. Since the integration of force sensors changes the structural dynamics of the system, methods for indirect force determination were developed. For this purpose, acceleration or displacement quantities and transmission properties are measured, from which the forces are subsequently determined, for example by matrix inversion. An overview of the methods is given in Table 1, that is an excerpt of [6].

**Table 1.** Overview and classification of TPA methods according to [6].

| Family         | Method         | Operational measurement | Source characterization | Apply to          |
|----------------|----------------|-------------------------|-------------------------|-------------------|
|                |                | Quantity              | On system              | Quantity         | Using                  | FRFs                   |
| Classical      | Direct-force   | $g_2^B$               | AB                     | Interface force  | $g_2^B$               | $Y_{32}^B$             |
|                | Mount-stiffness| $u_A^B, u_2^B$         | AB                     | Interface force  | $g_2^B$               | $Z_{mt}$               | $Y_{32}^B$             |
|                | Matrix-inverse | $u_4$                 | AB                     | Interface force  | $g_2^B$               | $Y_{42}^A$             | $Y_{32}^B$             |
| Component-based| Blocked-force  | $g_2^B$               | A (blocked)            | Equiv. force     | $f_{eq}^A$             | $Y_{32}^{AB}$          |
|                | Free-velocity  | $u_4^\text{free}$     | A (free)               | Equiv. force     | $f_{eq}^A$             | $Y_{22}^A$             | $Y_{32}^{AB}$          |
|                | Hybrid-interface| $g_2^B, u_2$          | AR                     | Equiv. force     | $f_{eq}^A$             | $Y_{22}^A$             | $Y_{32}^{AB}$          |
|                | In-situ        | $u_4(u_A)$            | AR/AB                  | Equiv. force     | $f_{eq}^A$             | $Y_{AR/AB}^{22(42)}$   | $Y_{32}^{AB}$          |
|                | Pseudo-forces  | $u_4$                 | AR/AB                  | Pseudo force     | $f_{ps}$               | $Y_{AR/AB}^{4ps}$      | $Y_{32}^{4ps}$         |
Only for the classical direct-force and the component-based blocked-force method, the forces are determined directly without further calculations. For all other methods, the forces are determined indirectly e.g. with mathematically expensive operations like matrix inversion. The blocked-force method uses the fact that the equivalent forces are equal to forces that fixate the active system rigidly. These forces are called blocked forces $b_{2}^{bl}$ and can be derived with the boundary condition $u_A = 0$.

\[ u_A = 0 = Y_{21}^A f_1 - Y_{22}^A b_{2}^{bl} \]  

By rearranging equation (8) the following is obtained:

\[ b_{2}^{bl} = (Y_{22}^A)^{-1} Y_{21}^A f_1 \]  

This corresponds exactly to the equivalent forces $f_{eq}^A$ shown in equation (7) that shows that both forces are equal.

The use of TPA for the evaluation of numerical models is known in the literature as numerical TPA (NTPA). In previous publications the application of two TPA methods to numerical models has been shown, although in each case only under consideration of a single operation point: [7] and [8] show the applications of the classical direct-force method. In [9], the application of the component-based in-situ method for a linearized system in frequency domain is presented. Due to the indirect force determination of the in-situ method, extensive post-processing is necessary. The advantage of numerical models of direct availability of forces without further complex calculations is not used.

In contrast, this publication extends the NTPA methods by the component-based blocked-force method to achieve high accuracy and low computational effort. The parallel use of the classical direct-force method allows for a direct comparison of the sound contributions between the classical and component-based TPA families.

3. Solution approach

In early phases of the PDP, elastic multi-body simulation (EMBS) is widely used for analyzing the NVH behavior of virtual prototypes. The EMBS enables efficient calculation of the structural dynamics of large, flexible structures considering non-linear transmission elements and transient excitations under variable operating conditions in frequency as well as in time domain. Coupling elements such as bushings or roller bearings are typically modeled using concentrated force elements. Variables like acceleration or forces can be determined at any desired location in the system. Typical weaknesses of experimental TPA, such as exact force determination, consideration of rotational degrees of freedom, installation space limitations and measurement noise, do not exist for numerical simulations.

For selecting of suitable TPA methods for the application to numerical models, two requirements in particular are considered: high accuracy and low computational effort especially in post-processing. The two requirements are met by selecting physical TPA methods with direct-force determination. As described above, the concentrated force elements of the EMBS allow for direct force signal output at interfaces without further calculations. The amount of data exchanged is also kept low by the direct-force determination, which enables efficient calculation in the post-processor. Therefore, the classical TPA with direct-force and the component-based TPA with blocked-force determination are selected.

For both methods the procedure and the calculation chain were created. Each method requires two separate simulations which can be derived based on an initial model covering the overall system AB. The methods differ mainly in the definition of the system boundaries of the simulation models.

Figure 2 shows the method proposed for determining path contributions according to the classical TPA method with direct-force determination for numerical models.
Figure 2. Proposed method for performing classical NTPA with direct-force determination.

As shown in the previous section, in classical TPA the interface forces are obtained in the assembled system AB. Using time integration, the time-dependent interface forces are calculated in time domain for example for a run-up of a drivetrain. Both, translational and rotational forces are directly available as output variables. Using discrete Fourier transform (DFT), the time signals are transferred into frequency domain. The FRFs for the classical NTPA are determined in frequency domain on the passive structure using linear system analysis (LSA) calculation. For this purpose, the active subsystem is deactivated in the simulation model. The FRFs are determined successively between all interface degrees of freedom (DOFs) and the receiver DOFs. The path contributions are obtained by multiplying the complex-valued forces and FRFs.

The proposed method for determining path contributions using the component-based TPA method with blocked-force determination for numerical models is presented Figure 3.

Figure 3. Proposed method for performing component-based NTPA with blocked-force determination.

The equivalent forces required in the component-based TPA equals to the so-called blocked forces. The blocked forces are determined by fixing the source structure to an infinitely stiff environment. In the model this is achieved by deactivating the passive structure and connecting the interfaces to the ground. The LSA is performed on the assembled system. The operations for post-processing are the same as in the classical method.

The number of paths results from the number of interface DOFs and receiver DOFs. In general, all force-transmitting interface DOFs should be considered. Receiver DOFs can be freely selected by the user according to his interests. Translational as well as rotational DOFs can be considered. For clarity, individual paths can be summarized in the visualization tool. For example, the interface DOFs of individual interfaces can be summarized and the receiver DOFs can be combined to one receiver signal.

4. Application of NTPA to a tractor drivetrain
The NTPA methods presented above are applied to a demonstrator EMBS model for illustration. A validated model of a tractor drivetrain with a hydrostatic-mechanical power split transmission is used, which is described in detail in [3]. The model used in this study is shown in Figure 4.
Figure 4. EMBS model of a tractor drivetrain as demonstrator for NTPA.

The main excitation source of the system are the hydrostatic units of the drivetrain. The hydrostatic units are mounted in a separate inner housing and connected to the drivetrain housing elastically via four elastomer bushings. In the NTPA, the paths from the bushing suspensions to the cabin suspension are considered. Therefore, the cutting plane is placed along the bushings. In the following study, the cabin suspension front right, marked with ‘R’ in Figure 4, is selected as the receiver point. The bushings transmit both translational and rotational forces along all DOFs. This results in six possible interface DOFs per bushing, in total 24 interface DOFs. In this example the total translational acceleration at the receiver is considered.

Using the methods shown in section 3, the sound contributions are calculated for the tractor drivetrain for a transient run-up of the vehicle speed. For verification of the calculation, the acceleration level at the receiver point measured in the reference model is compared with the synthesized total acceleration level generated by the NTPA using equation (1). The results are shown in Figure 5.

Figure 5. Comparison of the simulated and synthesized acceleration levels at the receiver.

The acceleration signals show a high degree of conformity. Small deviations of maximum 1-2 dB are visible. These deviations can be caused by effects of linearization or numerical inaccuracies. The deviations are relatively small and do not influence the qualitative results of the NTPA. Overall, the diagram shows a high accuracy of the calculation for both methods.

In the following the results of the NTPA are presented in the form of the sound contributions. As mentioned above, the sound contributions can be evaluated on different levels for visualization purposes. Figure 6 shows the sound contributions of the four bushings to the total sound for the classical NTPA. For comparison, the acceleration level of the reference model is presented in the first row.
The color bars indicate the acceleration level of the individual paths. This visualization allows for a transparent identification and ranking of dominant paths. For example, it becomes clear that the peak at the time 3.5 s is mainly caused by bushings 1 and 2.

For a more in-depth investigation, the contributions of the bushings are divided into the six interface DOFs, Figure 7.

It is visible that especially the paths in x- and y-direction of bushing 1 and y- and z-direction of bushing 2 have the highest contribution to the total sound.

The result of the component-based NTPA is shown in Figure 8.
Figure 8. Component-based NTPA: Sound contributions of the bushings divided into six interface DOFs.

The results are similar to the results of classical NTPA, although not exactly the same. For example, the path of bushing 2 in z-direction is less dominant in component-based NTPA in comparison with classical NTPA with a difference of about 8 dB at the time 3.5 s. The reason for the differences in the sound contributions of the two methods will be discussed in the following section.

5. Discussion
The results presented in section 4 show that TPA can be performed on numerical models with the proposed calculation chains. The comparison between the simulated and synthesized acceleration levels at the receiver demonstrates the high accuracy of the calculation, as presented in Figure 5. Weak spots in the complex transfer behavior become clearly detectable by the visualization of the sound contributions in the color bar diagrams, Figure 6-8. These results can be used directly to derive optimization measures, such as a stiffening of the passive-side connection of the bushing.

The calculation time of the post-processing including the visualization takes only few minutes. Therefore, the routines developed in this study are well applicable in the PDP. Among them, the component-based NTPA with blocked-force determination offers another essential advantage: significant reductions of the calculation times of the time integration simulation. The time integration for this method was performed only on subsystem A and not on the overall system due to the substructuring strategy presented. Depending on the system, this can lead to significant reductions in calculation time. In the case of the tractor model, the calculation times for time integration were reduced by about 50%. Another advantage of the component-based method is that the blocked forces depend only on system A. In the case that changes or optimization measures are applied to system B, the blocked forces remain unchanged. For this reason, only an LSA calculation with its short calculation times (few seconds or minutes) must carried out to evaluate optimization measures. Thus, the sound contributions for a modified system can be generated within a few minutes.

As mentioned in the previous section, the sound contributions of the classical and component-based methods are similar but not equal for the system under study. While the total sound at the receiver $u_{3}$ is
mathematically the same for both methods, the path contributions of the two methods are in general not equal. This is caused by the different way of substructuring and thus the different calculation order of equation (2). The difference in the mathematical calculation is derived in the following for one frequency grid point. For clarity, equation (2) is rewritten as

\[ u_3 = A \cdot B \cdot C \]  

(10)

with

\[ A = Y_{32}^B \in \mathbb{C}^{1 \times n} \]  

(11)

\[ B = (Y_{22}^A + Y_{22}^B)^{-1} \cdot Y_{22}^A \in \mathbb{C}^{n \times n} \]  

(12)

\[ C = (Y_{22}^A)^{-1} \cdot Y_{22}^A \cdot 1 \in \mathbb{C}^{n \times 1} \]  

(13)

The dimension \( n \) corresponds to the number of interface DOFs. In the case of classical TPA, entry \( i \) of \( A \) is multiplied by entry \( i \) of the matrix product of \( B \) and \( C \) for calculation of the sound contribution of path \( i \):

\[ u_{3,\text{path } i}^{\text{classic}} = A_i \cdot (B \cdot C)_i \]  

(14)

For calculation of the sound contribution of path \( i \) in component-based TPA, entry \( i \) of the matrix product of \( A \) and \( B \) is multiplied by entry \( i \) of \( C \):

\[ u_{3,\text{path } i}^{\text{comp.}} = (A \cdot B)_i \cdot C_i \]  

(15)

In general, equations (14) and (15) are not equal, and thus the path contributions of classical TPA and component TPA are not the same.

In the special case that \( B \) is diagonal, equations (14) and (15) and thus the path contributions of both methods are the same. \( B \) is diagonal when no crosstalk between the interface DOFs exists, because crosstalk is indicated by the side diagonal entries of \( B \). Crosstalk means that a force at one interface DOF causes a response, e.g. movement or force, at another interface DOF. The level of crosstalk depends on the system and on the definition of the cutting plane. The special case of a system without crosstalk can also be illustrated as follows: an equivalent force acting at one interface DOF in the system \( AB \) causes only an interface force at the same interface DOF, whereas the interface forces at all other DOFs are zero.

However, real systems typically exhibit a certain amount of crosstalk. In the case of the tractor drivetrain presented in the previous section, this amount is relatively small, resulting in slightly different sound contributions for the two methods.

In systems with a high level of crosstalk, the results of the two methods can vary significantly, leading to different interpretations and optimization measures for the systems in question. Hence, the choice of method should be guided by the specific changes and optimization measures to be evaluated. For a fully filled matrix \( B \), in classical TPA a sound contribution \( u_{3,\text{path } i}^{\text{classic}} \) contains exactly one transfer path between an interface DOF and the receiver, and \( n \) forces acting within the interface DOFs. In contrast, for component-based TPA with a fully filled matrix \( B \), a sound contribution \( u_{3,\text{path } i}^{\text{comp.}} \) results from \( n \) transfer paths between the interface DOFs and the receiver, with exactly one equivalent force acting at one interface DOF. Thus, classical TPA can provide more transparent results for evaluating single transfer path changes in the passive system. The results of the component-based TPA, on the other hand, can be more transparent for evaluating optimizations that aim to change a force dependent on the active system.
6. Summary and Outlook
This contribution shows the application of the classical and component-based TPA methods to numerical elastic multi-body system-simulation models and their comparison. The classical direct-force and component-based blocked-force TPA methods were chosen for achieving high accuracy with low computational effort even for the analysis of wide and transient operation ranges. Both methods are challenging for use in experimental TPA because of the direct way of force determination, but particularly suitable for the application to numerical models. The calculation chains developed for both methods were applied to a demonstrator model of a tractor drivetrain for investigating the structure-borne noise for a transient run-up of the vehicle speed. The NTPA allows for in-depth insights into the system behavior with low computational effort. The efficient and target-oriented analysis tool is suitable for being used in early phases of the product development process for identifying optimization potential in a transparent and structured way.

It was shown that the component-based blocked-force method additionally offers a high potential for the reduction of simulation time. Since the classical and the component-based NTPA were carried out in parallel, a direct comparison of the sound contributions was possible, whereby slight differences were identified for the demonstrator model. Further investigations showed that these differences are caused by the different calculation order of the path contributions for the two methods. Both methods provide equal path contributions only for systems without crosstalk, which are barely found in reality. For systems with high crosstalk, on the other hand, a theoretical discussion is given about which method allows evaluating optimization changes with higher transparency.

In this contribution, the NTPA was presented on structure-borne sound level. In further works, the calculation chain of classical NTPA presented in this study has been extended in order to perform path evaluations on airborne sound level for numerical system-simulation models [10].

Furthermore, the NTPA methods are to be integrated into automated optimization processes. The component-based NTPA method is particularly suitable for this purpose due to its high potential for reducing simulation times. Especially for optimizations on the passive system, only FRF calculations are required per iteration, so that each iteration can be performed within few minutes.

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