Finding and Fixing Vehicle NVH Problems with Transfer Path Analysis

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This article discusses the use of experimental transfer path analysis (TPA) to find optimized solutions to NVH problems remaining late in vehicle development stages. After a short review of established TPA methods, four practical case histories are discussed to illustrate how TPA, FE models and practical experiments can supplement each other efficiently for finding optimum and attribute-balanced solutions to complex NVH issues late in the development process.

Experimental transfer path analysis (TPA) is a fairly well established technique,[1,2] for estimating and ranking individual low-frequency noise or vibration contributions via the different structural transmission paths from point-coupled powertrain or wheel suspensions to the vehicle body. TPA is also used to analyze the transmission paths into vibration-isolated truck or tractor cabs etc.

TPA can also be used at higher frequencies (above 150-200 Hz) in road vehicles, although it may be reasonable to introduce a somewhat different formulation based on the response statistics of multimodal vibro-acoustic systems with strong modal overlap.[3]

When NVH problems still remain close to start of production (SOP), experimental TPA is often a favored technique to investigate further possibilities to fine-tune the rubber components of the engine or wheel suspension with respect to NVH. The aim is to further improve NVH with minimal negative impact on other vehicle attributes, such as ride comfort, handling, drivability, durability, etc.

The only design parameters that can directly be changed in a "what if?" study based purely on experimental TPA, are the dynamic properties of rubber elements connecting the source and the receiving structure. Also, any reduction of transfer path contributions to noise or vibration in that case will be a result of reducing some of the dynamic stiffness' for the connecting elements. To take any other design changes into account, additional measurements are normally necessary.

Theoretical Background

Each degree of freedom (DOF) acting at interface points between a vibration source system and a receiving, passive vibro-acoustic system is a transfer path in TPA. TPA can also be performed analytically, using FE models or specialized system analysis software.[4]

The experimental TPA method involves:

1) An indirect measurement procedure for estimating operating force components acting at the coupled DOFs.

2) The direct or reciprocal measurement of all transfer frequency response functions (FRFs) between response in points of interest (e.g. at the drivers ear) and points where these forces act. The FRFs are measured with the receiving subsystem disconnected at all the DOFs included in the TPA analysis, while all other connections remain in place.

Assume that the structure is linear and time invariant and that the operating forces and transfer functions have been determined. The partial contributions and total response (e.g. the sound pressure or vibration at a specific point), are obtained as illustrated in Figure 1.

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Classic (Traditional) TPA. Traditional TPA is based on the superposition principle that is valid for linear, time-invariant systems. The individual path contribution to the sound pressure (or vibration) in point m from a force acting in point n in direction k is given by:

$$p_{mnk}(\omega) = H_{mnk} \cdot F_{nk}(\omega) \tag{1}$$

where:

p_{mnk} = (complex) sound pressure spectrum
H_{mnk} = (complex) frequency response function (NTF) of the receiving system when decoupled at interfacing DOFs
F_{nk} = complex force spectrum

The total response (e.g. sound pressure) is then obtained as:

$$p_m(\omega) = \sum_{n=1,k=1}^{N,3} p_{mnk}(\omega) = \sum_{n=1,k=1}^{N,3} H_{mnk}(\omega) \cdot F_{nk}(\omega) \tag{2}$$

if only the translational DOFs are included in the measurements. Note that the TPA for vibration response becomes analogous by using the appropriate vibration-to-force (VTF) frequency response functions.

The determination of operating forces for each DOF is done indirectly and can be performed in three different ways. The first and most common method is to use resilient connecting elements as "force transducers," provided that the complex dynamic transfer stiffness is known for the different DOFs.[8]

The forces are then obtained as:

$$F_{nk}(\omega) = k_{nk}(\omega)(x_{nk1}(\omega) - x_{nk2}(\omega)) \tag{3}$$

where:

k_{nk} = complex *transfer* stiffness for mount n in direction k
x_{nk1} = operating displacement at the source side
x_{nk2} = operating displacement at the receiver side

The second method is to use inversion of the measured FRF matrix between structural responses and exciting forces acting at all interfacing DOFs on the receiver side. This inverted matrix is then multiplied with the vector of operational measurements of vibration on the receiver side.

The matrix inversion method has to be used when the transfer paths include rigid connections or the mounts are very stiff compared to the receiving structure, since the relative displacement across the mount becomes too small to use Equation (4). The forces are obtained as:

$$\begin{Bmatrix} F_1 \\ \vdots \\ F_N \end{Bmatrix} = \begin{bmatrix} H_{11} & H_{12} & \cdots & H_{1N} \\ \vdots & \ddots & & \vdots \\ H_{M1} & \cdots & \cdots & H_{MN} \end{bmatrix}^{-1} \begin{Bmatrix} \ddot{x}_1 \\ \vdots \\ \ddot{x}_M \end{Bmatrix} \tag{4}$$

where the number of responses M can and should be larger than the number of force DOFs N. An over-determination factor of about 2 is often used.

The third, very simplified method is to estimate the force for each DOF by multiplying the measured point FRFs with the measured operating responses on the receiver side. This method neglects any transfer FRFs between the different DOFs, and the force estimated for each DOF is obtained as:

$$F_i = H_{ii}\ddot{x}_i \tag{5}$$

This seems simple and attractive, but the estimation error usually becomes very large, especially at lower frequencies where the response contribution due to forces acting at all other DOFs is quite large.[9]

Mid- and High-Frequency Range TPA. The contribution

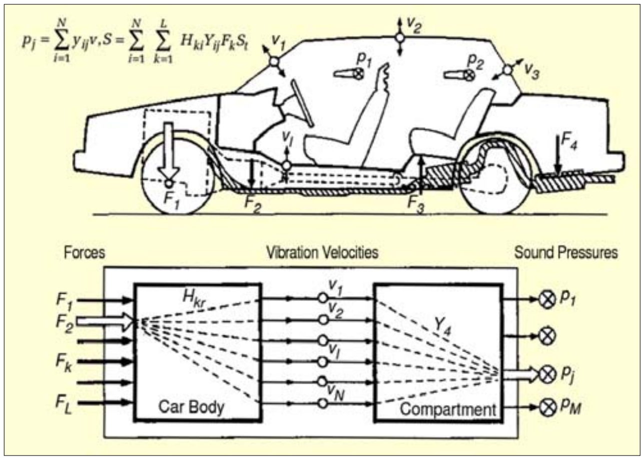

Figure 1. The principle of transfer path analysis.

analysis can be expanded into the mid- and high-frequency range[3] (where the modal overlap factor > 1) by considering FRFs in sufficiently wide frequency bands or using energy related quantities at the receiver locations. Both or either one of these modifications to the ordinary TPA can be applied to obtain more stable and relevant path contributions in the mid- and high-frequency ranges.

We redefine the contribution to the sound pressure (or vibration response) at a receiver point m from a force acting in point n and in direction k as:

$$\tilde{p}^2_{mnk}(\omega) = \left\langle H^2_{mnk}(\Delta\omega)\right\rangle \cdot \tilde{F}^2_{nk}(\omega) \qquad (6)$$

where $H(\Delta\omega)$ is the notation for an averaged frequency response function (typically from 5% bandwidth up to 1/3rd octave) around the frequency considered. The force and the response are still defined at discrete frequencies. Depending on the strategy used for transfer path ranking, one may use the maximum FRF values in these frequency bands (worst-case scenario) instead.

The total mean-squared sound pressure at point m is obtained by summation of the *uncorrelated* path contributions as:

$$\tilde{p}^2_m(\omega) = \sum_{n=1,k=1}^{N,3} \tilde{p}^2_{mnk}(\omega) \qquad (7)$$

The spatially averaged squared response (proportional to the receiving system energy) is a much more stable quantity in the frequency region with modal overlap than the point-to-point frequency response function.[7] See the example in Figure 2.

To obtain even more stable path contributions one should also determine a *space-averaged* frequency response function for the receiving location. The forces F_{nk} are still acting at the specific point interface locations, while the response $<p_{mnk}>$ is measured and averaged for a number of points in a domain around the receiving point m.

For a homogenous receiving system (like an acoustic space or an unstiffened plate), the energy-related, space-averaged frequency response function is calculated for each force component as:

$$\left\langle H^2_{mnk}(\Delta\omega)\right\rangle = \sum_{l=1}^{L} H^2_{mnklk}(\Delta\omega) \qquad (8)$$

and Equation (6) becomes:

$$\tilde{p}^2_{mnk}(\omega) = \left\langle H^2_{mnk}(\Delta\omega)\right\rangle \cdot \tilde{F}^2_{nk}(\omega) \qquad (6')$$

Equation (7) can then be used to calculate the path contributions and the total mean square response. The analysis may be carried out for each frequency point or for an arbitrary frequency band.

Case Histories

Case 1 – Idle Boom and Vibrations. The idle boom and vibration situation varied during development of a front-wheel-drive car with a resiliently mounted subframe. It was still un-

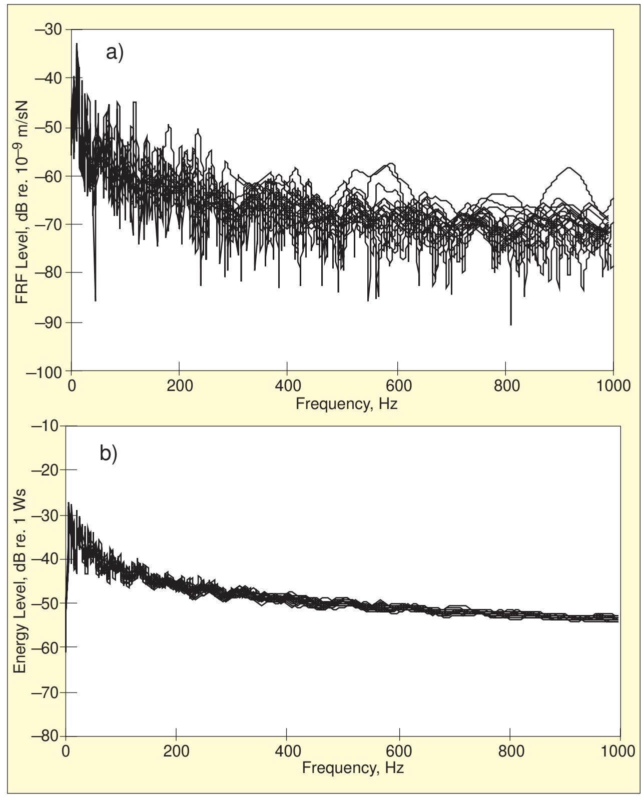

Figure 2. a) Variation of FRF levels between points on two subsystems, calculated for an L-plate ensemble. b) Variation of spatial average velocity level in the receiving plate of the L-plate ensemble. Combined local parameter variation standard deviations: 2% for eigen frequencies and 20% for the logarithm of modal damping.[7]

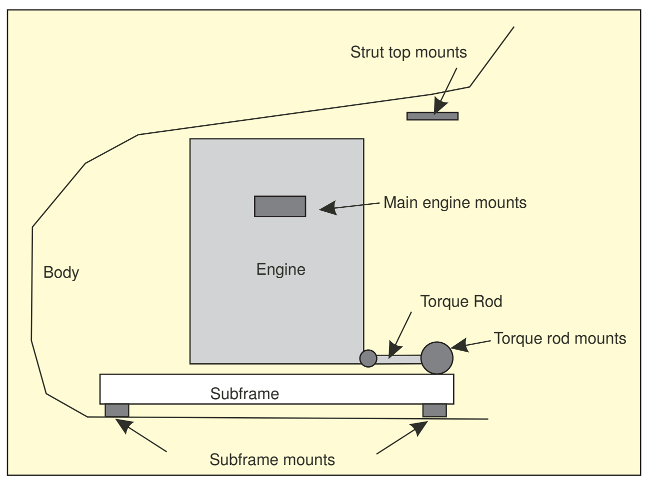

Figure 3. Sketch of the engine installation for the idle noise and vibration case. The wheel suspension has McPherson struts with lower wishbone link arms that are attached to the subframe.

acceptable during the final attribute development and tuning process. The principle of the engine and front-wheel suspension is sketched in Figure 3. The following discussion of this case is generic and not specific to any particular car.

This engine installation concept is quite common today, using a pendulum suspension with the engine hanging on the left and right engine mounts that are attached to the main front longitudinal body beams. Torque rods are introduced to handle most of the static wind-up of the power unit when loaded. The dynamic rotation of the power unit due to torque pulsations will result in forces transmitted through the main engine mounts and torque rods. In addition, torque pulsations are transmitted through the drive shafts to the wheels and the wheel suspension. At idle, dynamic forces transmitted through strut top mounts to the body may often dominate the idle noise and vibration levels in the interior.

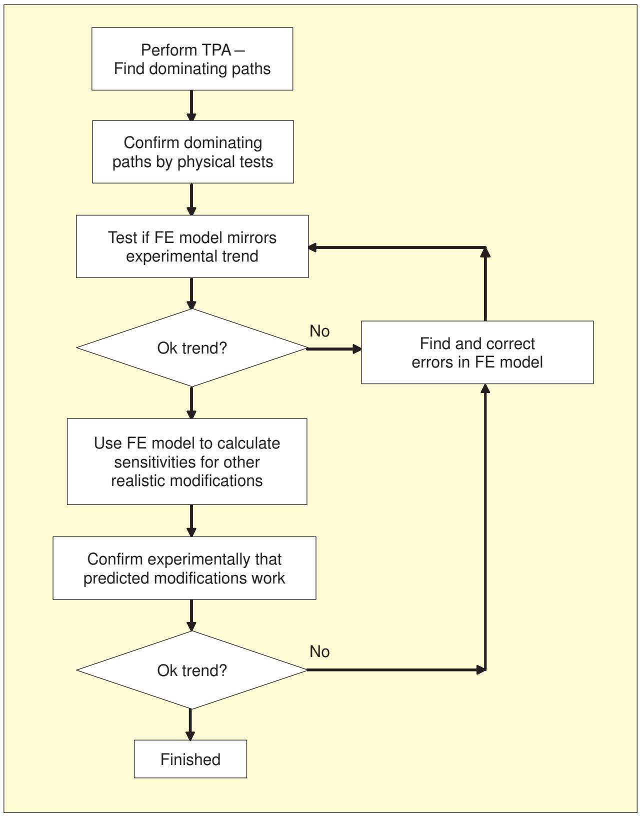

Figure 4. The process of using TPA and CAE to find solution to an idle NVH problem late in a new car development. The philosophy may be used in other similar situations.

Experimental TPA is routinely used to sort out the relative importance between all those rubber mount connections. In this case we had a total of 30 DOFs or possible transfer paths. We will not discuss the practical details of how to perform the measurements involved or the TPA itself, since this is well known and did not present any surprises. The experimental TPA showed that the front-most subframe mounts were responsible for most of the transmission of the second engine order noise and vibrations at idle. This was also confirmed, both by objective measurements and subjective evaluation, by running the car at idle with extremely soft subframe mounts at the front.

Making these mounts very soft was obviously the only direct measure suggested by the experimental TPA that may be realistically implemented this late. Reducing the high noise and vibration sensitivity (NTFs and VTFs) of the trimmed body for excitation at those points was not an option that close to SOP. However, much softer subframe mounts were not an acceptable solution from a handling and steering point of view. Any minor reduction of the front mount stiffness was on the other hand not sufficient for obtaining acceptable NVH results. This seemed again to be the classic attribute conflict situation between idle NVH (soft bushings) and handling (hard bushings) and a compromise would result in mediocre quality for both attributes.

A detailed FE model for the complete car was available. It was therefore suggested that this model should be used to calculate interior noise and vibration sensitivities at idle to any realistic changes in stiffness for all the different rubber mounts that may have an influence. The sensitivity analysis was carried out for the worst-case operational idle condition found in the tests (transmission in drive with front brakes applied). When the sensitivity table was evaluated, however, it suggested that the front-most subframe mounts should be made *stiffer* to reduce idle NVH. This obviously contradicts the experimental evidence and indicates that the FE model was not good enough to be trusted and used for these forced-response calculations.

Eventually, it was found (in this case) that the dynamic boundary conditions introduced for the applied-brakes condition had to be modified to obtain the correct sensitivity trend for the front-most subframe mount modifications. The sensitivity analysis was then repeated with the updated FE model with quite surprising but interesting results. The sensitivity analysis suggested that the largest idle NVH improvement would result when the rear subframe mounts were *made stiffer, not softer*. This is obviously not a modification that could be derived from experimental TPA alone, since any increase in mount stiffness would result in an increase in calculated receiver response levels. On the other hand, it would not have been possible to obtain useful modification proposals with the CAE model without the model update introduced by the results from the experimental TPA.

The reason experimental TPA could not suggest this solution is that it uses a measured 'snapshot' of the operating vibrations at the source side of the bushings. These vibrations are assumed to be independent of bushing changes. The suggested increased stiffness will lead to considerably reduced dynamic motion of the subframe at its front mounts. This results in much lower transmitted forces with the existing mounts. This was obvious when subframe motion was animated with the FE model for the different design cases. The somewhat increased forces through the rear mounts of the subframe were negligible due to much lower noise and vibration sensitivities for those points.

Tests with the stiffer rear-most subframe mounts did indeed result in significantly lower idle noise and vibration levels and an approval for SOP. In addition both the NVH and handling attributes gained from this solution. This became an unusually happy situation without compromises.

The analysis process in this case can be summarized in the flow chart in Figure 4. The total analysis time to obtain the accepted solution was about three weeks, most of which was actually spent to find out what was wrong with the FE model.

Case 2 – High-Frequency TPA. A typical, mid-frequency tonal automotive noise problem illustrates mid- and high-frequency TPA techniques. Interior noise from a power steering pump was transmitted from the hydraulic system to the car body structure through numerous transfer paths, involving resilient rubber elements. Typically, annoying tonal noise components in a frequency range of 200-600 Hz can be present at various engine speeds.

Figure 5 illustrates typical noise-to-force transfer functions (NTFs) measured at one of the connection points for force excitation in three translational directions. The magnitude as well as the phase of the frequency response function varies considerably and rapidly with frequency. When applying traditional TPA procedures, we calculate phases for each path that will vary considerably in length and direction for small changes in frequency. This results in a corresponding variability of calculated total noise level at the receiving location.

To provide meaningful estimations for design purposes, reasonable magnitude and phase stability is needed for the transfer functions. Due to the large modal overlap at medium and high frequencies, we expected the variability of involved FRFs to be quite large between individual cars.[5-7] Traditional TPA results will therefore not represent the ensemble of produced cars. Averaged frequency response functions were therefore calculated over a moving 10% frequency band, as discussed in the previous section. Only the magnitudes of the noise contributions from individual transfer paths are used, while the phases between the components are omitted. Figure 6 illustrates a typical ranking diagram that shows the magnitudes of these individual path contributions.

This mid- and high-frequency TPA was used to exclude a large number of nonrelevant transfer paths and also to 'test' the effect of different modifications for the dominating paths, especially concerning the effectiveness of changes to resilient element properties. This is basically how traditional, low-frequency TPA is used, except that cancellations due to opposite phases between paths are not taken into account and the NTFs are 'smoothed' in frequency.

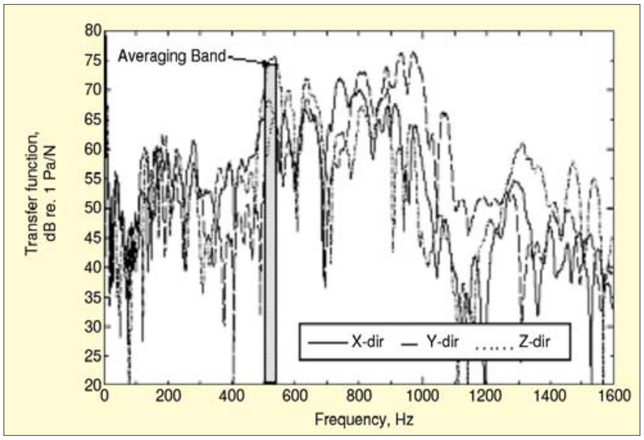

Figure 5. Example of measured noise-to-force transfer functions (dB re. 1 Pa/N). A typical frequency averaging band is illustrated.

Case 3 – Old Fashioned "Physical TPA" Can Still Work. Before any commercial, effective software for systematic experimental TPA analysis was available, actual physical decoupling of certain suspected mounts was used extensively to sort out structure-borne noise paths. It is an important lesson that this "old fashioned" technique can still sometimes be used quite efficiently and may be a tremendous timesaver. In addition, it constitutes a good, independent test method to validate dominating path estimations obtained from a full TPA analysis.

This case history shows how a physical decoupling of an engine mount, based on educated guess and general experience, provided an extremely quick and conclusive identification of the dominating transfer paths for typical mid- and high-frequency noise from transmissions (gear-whine, rattle and clunks). The task was to reduce excessive gear whine, clunks due to lash in the transmission, and also possible gear rattle. These quality-related transmission noises became more and more disturbing while the test cars produced were successively improved with respect to general interior noise levels (interior engine noise, road noise, wind noise, etc.).

One of the first things the task work force had to do was to definitely identify the dominant transfer paths for each of these noises as soon as possible. This enabled us to concentrate all design resources on modification alternatives for those paths only and nothing else, since short lead time was the ultimate priority. Every day of work was invaluable.

The engine suspension was a typical front-wheel-drive pendulum design with the left power unit mount attached to the transmission housing. A reasonable first guess that needed an immediate answer was if the structure-borne transmission through this mount could be the dominant contributor.

Instead of starting with a full experimental TPA involving all the possible structure-borne and airborne sound paths, it was first decided to arrange a simple physical decoupling of the left engine mount from the body while elastically supporting the gearbox by the floor of the chassis dynamometer. It was concluded by subjective evaluation and objective measurements that most of the gear whines and virtually all clunks disappeared when the mount was physically decoupled.

This information was sufficient for immediately starting both a detailed analysis and proposed solutions to reduce mid- and high-frequency transmission via this mount. A subsequent experimental TPA was limited to the engine mount paths (using the above-mentioned mid- and high-frequency method), and it also confirmed that the transmission via the left engine mount indeed dominated strongly. It also sorted out details about the relative ranking between the X- Y- and Z-directions for different noises and driving conditions. See the example in Figure 7 showing a result for part of the gear whine issues.

By quickly identifying the main transmission path(s), based on both expertise and experience, all the task force resources could immediately be concentrated on finding the appropriate solutions instead of waiting for results from a full TPA

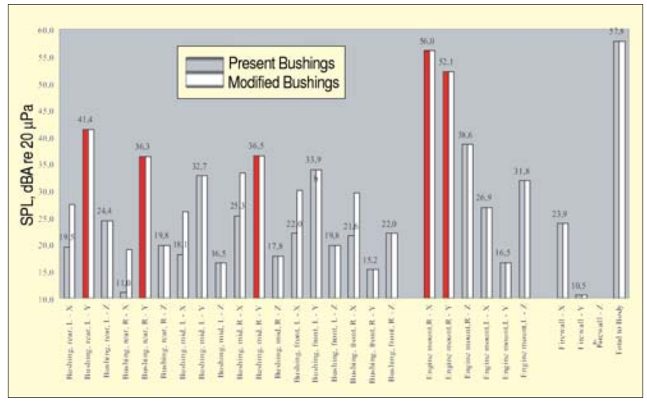

Figure 6. Example of transfer path ranking for one pump blade order at a specific speed.

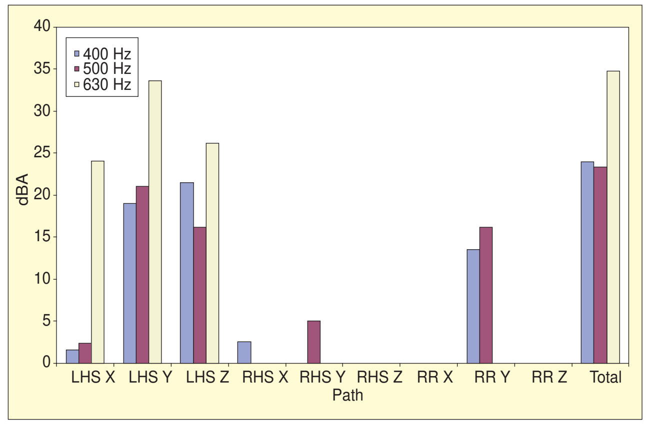

Figure 7. Ranking of transfer paths for structure borne gear whine (1st gear, slow speed)

analysis. The physical decoupling tests were performed and reported in less than one day. A limited TPA needed about three to four days of additional work to evaluate all the different noise components.

Case 4 – A Transfer Path Separation Problem That Needed Special Attention. The last TPA case history shows an especially difficult situation for transfer path contribution separation. The reason for the problem was that several 'source' structure attachment points were situated in close proximity.

The test object in this case was a heavy truck, and the specific task was to separate the structure-borne noise transfer path contributions to the truck frame from the engine suspension and the exhaust system respectively (among all other transfer paths not mentioned here).

The engine is suspended resiliently with a front engine mount on a cross-beam, and rear engine mounts attached to the left and right main frame beams respectively. The exhaust system is rigidly attached to the right main frame beam at a couple of points. Figure 8 illustrates the close proximity of one engine mount attachment location and one of the exhaust system attachments. They are practically mounted to one and the same 'point.'

The TPA situation becomes tricky because one of these attachments is a fixed support without a resilient element. Since full load speed sweeps were included in the test program, decoupling the engine mount was no reasonable option. Obviously the matrix method has to be used to determine the operational forces from the exhaust attachment. It becomes clear that inversion of an ill conditioned FRF matrix, which contains some almost identical rows, will be error prone, and reasonable overdetermination will not help.

We could estimate the force contributions via the engine mount using the resilient mount data. However, if these forces dominate or are of the same order of magnitude as the forces from the exhaust bracket, we still cannot get a reliable estimate of the exhaust system contribution.

Figure 8. Accelerometers attached to the engine mount position and the exhaust system attachment point respectively, illustrating the close proximity.

Additional tests were performed involving physical decoupling to identify the most important transfer paths. Decoupling the exhaust bracket showed that its contribution to the local frame vibration was not dominant. This was also confirmed by the results obtained in the receiver positions. When we applied the matrix inversion method by "brute force" the results indicated erroneously that the exhaust system attachments were important paths, even when it was combined with the mount method for force determination (where appropriate).

Conclusions

Experimental transfer path analysis is generally a powerful tool for the diagnosis of complex transmission of vibration and air-borne sound via multiple paths between energy sources and the car body structures. It is important to understand that experimental TPA is primarily a diagnosis tool and does not result in an experimental model that can be used directly for evaluating various design modifications without some critical considerations.

The dominating paths proposed by an experimental TPA should always be checked with other independent tests such as "old-fashioned" physical decoupling. It should always be remembered that the errors introduced due to practical test environment restrictions or due to the lack of reliable mount data can become too large for the results to be sufficiently reliable.

Reliable, independently confirmed TPA results can be used to test the validity of a CAE model and may initiate an updating of the model or its boundary conditions. This can result in strong synergies in terms of a CAE model being ultimately useful for finding alternative, better or nonobvious modifications. Other problem-solving techniques may require considerable insight combined with unconventional thinking.

TPA should be applied somewhat differently at higher frequencies, taking the natural variability of FRFs into consideration.

TPA can be considerably simplified especially for the analysis of transfer paths for higher frequency noise components. As illustrated, a reverse procedure using physical decoupling before actual measuring all parameters required for a full TPA can save precious time.

Sometimes conditions for straightforward TPA do not exist. In the case with attachment points in close proximity, analytical skill and experience are required to identify and treat such situations.

Acknowledgments

This article presents four real-life practical examples of TPA techniques from projects performed at Ingemansson Technology AB by different consultants for various vehicle manufacturers. I would like to thank Jörgen Svensson and Lars Ivarsson at Ingemansson for providing input to the case histories presented here. I also would like to thank Volvo Car Corporation, Saab Automobile AB, Volvo Truck Corporation, and Scania AB for providing opportunities to use experimental TPA for practical problem solving.

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