SYNTHESIS REPORT

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PROJECT No : BE-4436

TITLE : DIANA Development and Integration of an Advanced Unified Approach to Structure Borne Noise Analysis

PROJECT COORDINATOR : LMS International - Belgium .

PARTNERS : RNUR - France
MIRA - United Kingdom
CENTRO RICHERCHE FIAT - Italy
FACHHOCHSCHULE BIELEFELD - Germany

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O. Authors

D. De Vis, W. Hendricx	LMS Engineering, Belgium
H. Ghesquière, J.L. Meillier	Renault Research Centre (RNUR-DE), France
G. Toniato, D. Storer	Fiat Research Centre (CRF), Italy
N. Niedbal, M. Hermanski	Technical University Bielefeld, Germany
B. Jones, S. Roberts	Motor Industry Research Association, UK

1. Abstract

The DIANA project aims studying "Development & Integration of a Unified Approach for Structure Borne Noise Analysis". The project was led by LMS Engineering (Belgium) which collaborated with the research centres of FIAT (CRF, Italy) and Renault (RNUR-DE, France). Other members in the consortium were MIRA (UK) and the Technical University of Bielefeld (Germany). Ford Germany acted as sponsor and provided testvehicles.

The first objective of the project included the investigation of advantages, disadvantages, sensitivity to boundary conditions and limits of confidence of several classical techniques in the field of structure borne noise analysis. These techniques were amongst others : single input transfer path analysis, principal component analysis and conventional mount testing.

At the other hand new techniques have also been elaborated. They were related to algorithms for indirect force determination, time domain principal component analysis and advanced mount testing.

Finally, the different techniques have been integrated into two global methodologies : Engine Noise Analysis and Road Noise Analysis. Both analysis methods have been implemented in software code.

2. Introduction

Interior noise in automotive vehicles is generally classified into airborne and structure borne sound. Optimisation of a vehicle with respect to airborne sound generated by powertrain and tires is related to the acoustical isolation, whereas structure borne sound is governed by mount characteristics and vehicle structure dynamics.

The major contributors to structure borne NVH problems are the powertrain inputs and the road inputs. To tackle these two different noise and vibration contributors, specific techniques have been developed for each of them. The" different wheel inputs due to road excitation are non-coherent and are usually treated by statistical techniques such as partial coherence and principal component analysis (PCA). On the contrary, all driveline to body connections" introduce coherent powertrain forces, necessitating a different approach. The contribution to the total inside noise of all of the individual connections is identified by Transfer Path Analysis (TPA).

The analysis of non-coherent signals by statistical techniques' further requires stationary conditions, whereas for powertrain noise typical analysis procedures are based on transient conditions : run up - run down. In order to elaborate further these existing techniques, LMS initiated an international research project : DIANA (Development and Integration of an Advanced unified Approach to structure Borne Noise Analysis).

The first objective of the project is to investigate the advantages, disadvantages and limits of confidence of different techniques in the field of structure borne noise analysis. Next, the confidence levels of the analysis techniques are to be increased by new implementations for troublesome procedures. The second objective is the development of novel techniques to reduce drastically the required time for a noise study.

These novel developments are : algorithms **for indirect** force determination, high quality FRF measurements, road noise analysis in the frequency domain, time domain Principal Component Analysis method for application with transient signals (e.g. driving over a bump in the road), advanced mount testing techniques including rotational degrees of freedom.

LMS is the project leader and collaborated with the research centres of FIAT (CRF, Italy) and Renault (RNUR-DE, France). Other members in the consortium are MIRA' (UK) and the Technical University of Bielefeld (Germany). Ford Germany is the project sponsor and provided testvehicles.

3. Single input Transferpath Analysis (Engine Noise)

Transferpath analysis, is a method which was developed in automotive applications mainly for studying the contribution of the individual powertrain and driveline mounts to the interior noise.

The total inside pressure can be formulated as the sum of partial pressures, each of them induced through specific transferways between the driveline (source) and the cars interior (receiver). These partial contributions can not be separated by coherence or PCA techniques because they are all coherent. The approach to separate them is based on additional dynamic flexibility measurements. The partial pressures can be written" as the product of an input force and a corresponding acoustic transferfunction.

$$p=\sum p_i=\sum\left(\frac{p}{F_i}\right)\cdot F_i$$

with

P : total car interior acoustic pressure

p_i : partial car interior acoustic pressure transmitted through transfer path

F_i : force transmitted through transfer path i

$p_i\ /\ F_i$: acoustic transfer function pressure/force ,

Mathematically, the total interior acoustic pressure can be written as the vector sum of partial pressures which are each transmitted by a different transfer path from the engine to the car interior. This partial pressure can be written as the product of an acoustic transfer function and a quantity representing engine inputs. The acoustic transfer function 'is defined as the acoustic inside pressure per unit external excitation in a specific connection i.

Fig. 1 gives an overview of the measurement and calculation procedure for the single input TPA partial pressure determination, based on mount complex stiffness values.

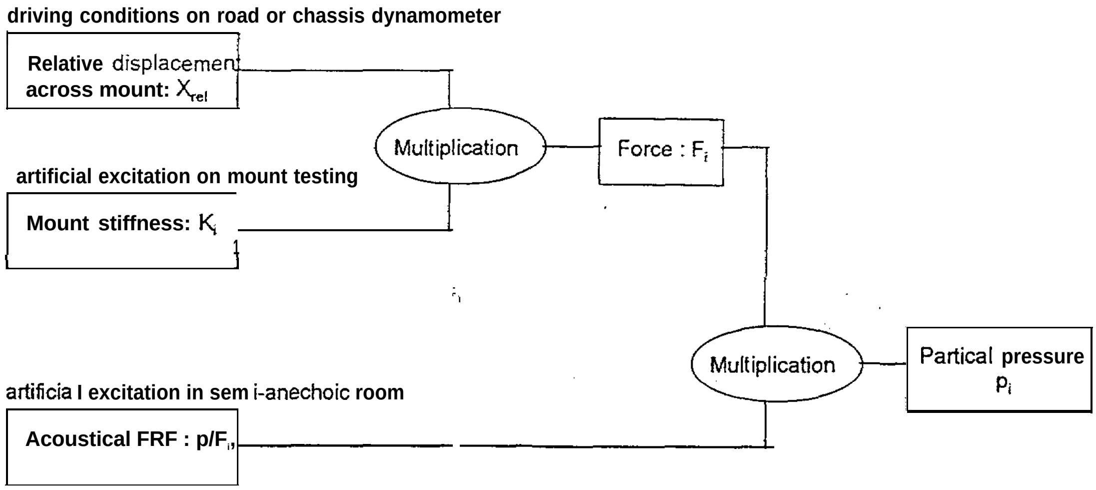


Figure 1 : Single input TPA : mount stiffness ***formulation.*** ‘

The transfer functions are being determined when the engine is not running and the car is being excited artificially by shaker excitation or hammer' impact. A mathematical proof has been given that is really necessary to remove the engine from the car body when those FRF are measured.

The main method to determine engine forces is based upon the complex stiffness matrix of the engine mounts. This method gives the best results when used in a real environment.

$$\{F\}=[K]\ \{X_{rel}\}$$

with

$\{F\}$: transmitted force vector

$[K]$: complex stiffness matrix

$\{X_{rel}\}$: relative displacement between the top and the bottom of engine mount

This approach requires the knowledge of the dynamic characteristics of the mounts (see further).

A second way to extract transmitted forces is based upon the measurement of the direct and cross FRF's between all engine mounts in each direction. Forces are then derived from acceleration measurements under operation, converted to displacements by double integration, and multiplied by the inverse of the flexibility matrix. The flexibility matrix contains the mentioned FRF's.

$$\{F\}=[H]^{-1}\cdot\{x\}$$

with

$\{F\}$: transmitted force vector

$[H]$: matrix of direct FRF's (including cross effect)

$\{x\}$: displacements at the body side of the mounts

This matrix inversion looks very straight forward. However, in practise many problems occur when inverting the FRF matrix.

This TPA procedure has been investigated at scale models of a car. Two complementary models have been built to simulate all fundamental effects of car dynamics. The advantage of model testing is that environmental parameters are much better controlled than during measurements on a real car. More over, results can ‘be verified because they are known on forehand.

These models have been constructed for two purposes. The first one is to validate the functionality of all software code being developed in the project. At the other hand, the accuracy and limitations of different transfer path calculation methods are investigated.

Beside this model testing the different TPA procedures have also been applied to several test vehicles. Not only a straight forward application of the techniques has been carried out. A strong emphasis has been put to the applicability of the procedures in an industrial environment. The effect of practical simplifications upon the accuracy of the results has been investigated in detail. Boundary conditions which have been evaluated are amongst other:

FRF measurements:
- the effect of the presence of powertrain and driveline
- linearity of the complete vehicle
- linearity of the vehicle after removal powertrain and driveline
- sensitivity of microphone position
- mass & volume loading at the cavity of the vehicle
- sensitivity of FRF towards the excitation location
- the effect of the exterior acoustical environment
- the effect of a superimposed low frequency sinus
- the effect of the presence of the engine hood
- hammer versus shaker excitation (indirect force determination).

During the acquisition of operational data at rollerbenches much attention has been paid to the repeatability of the measurement. Also the importance of the structure borne noise has been experimentally determined. In first instance intake and exhaust mufflers have been added in order to reduce airborne noises, However, this will not give a complete cancellation of the airborne noise.

In order to evaluate the contribution of the structure borne noise to the total noise the engine has been completely isolated from the car body. Therefore the engine has been suspended by means of airsprings to the floor of the chassisdynamometer test room. The drive shafts of the front wheel driven car have also been removed.

As a consequence there are no major mechanical connections between the powertrain/drive line and the car body.
Runup tests were performed in neutral gear and the interior noise has been measured. This noise is completely originated by the direct airborne transmission from the engine towards the cars’ cavity.

Special attention **has also been paid to the accuracy of** the force calculation methods. Force transducers have been installed in the scale models as well as in the testvehicle. Measurement results of these force transducers served as a reference **for the evaluation of indirect force estimation techniques.**

In particular the sensitivity of the mount stiffness and matrix inversion methods towards boundary conditions has been evaluated. Amongst others, the exact working point of the rubber mounts during operation has been determined. The position of this working point at the static stiffness curves gives an indication of the linearity of the (dynamical) behaviour of the rubber mount. Also the rotational degrees of freedom and the related moment excitation at an engine mount have been studied. For the matrix inversion method, conditions such as FRF quality, degree of matrix overdetermination, precision of transducer location, inversion method have been evaluated.

The final outcome of the TPA-analysis is a detailed model which indicates the structure borne noise transferpaths. These models serve as a tool to predict the effect of physical modifications of the car. The accuracy of these predictions has also been the object of the DIANA project. Figure 2 presents the total measured interior noise (second order) of a test vehicle. The solid line presents the original vehicle whereas the dotted line stands for a doubled stiffness of the rubber mount at" the dominant transferpath. The dashed line presents the interior noise when this mount is removed. (At this mount the engine was resting at an airspring connected to the floor). This measurement result has been exactly predicted by the TPA-model.

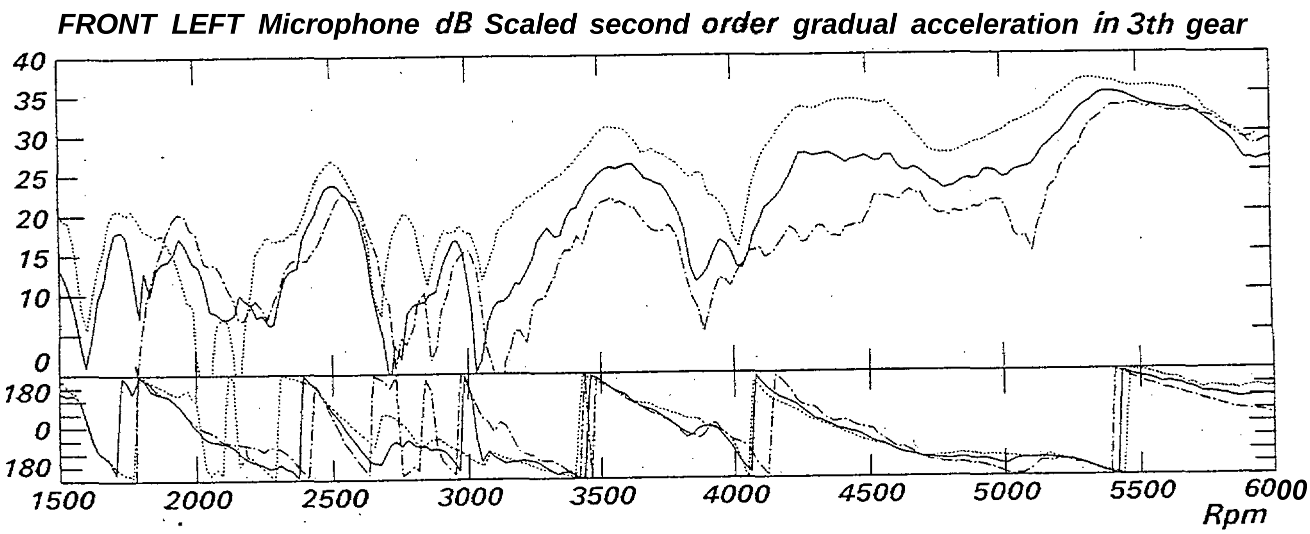


Fig. 2: Engine Noise Analysis. The effect of the variation of the mount stiffness of the dominant transferpaths.
solid line : original vehicle
doffed line : doubled stiffness
dashed line : removed mount (zero stiffness)

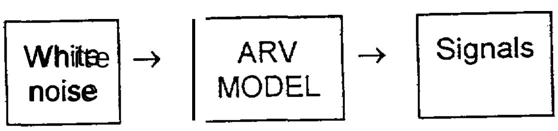
4. Time domain Principal Component Analysis

For application within the time domain a completely new approach has been sought **for. Instead of averaging the measured data one looks to 1 single, data time, series of time samples. This** single series of data are then modelled by autoregressive model (AR) or autoregressive moving average model (ARA). For multiple series autoregressive vector (ARV) and autoregressive moving average vector (ARMAV) model should be used for the analysis of multiple time series. When a statistically adequate model is determined, some' information about the series or system generating the time series can be obtained from the model, such as power spectrum of the time series, modal parameters of the system generating the time series. Furthermore, the model can also be used for the prediction of the future values of the time series, optimal control of the system generating the time series.

The motivation of using time series analysis method for the analysis of operating data is its advantages in spectral analysis over the classic method by FFT. It is well known that FFT will introduce some errors, such as bias and leakage error.

For the time series analysis method, when the model obtained from the time series is adequate, a less-biased with higher resolution and no leakage error result can be obtained. The most important thing is that the time series analysis method is most suitable for the analysis of time series with short length, and can be used for the analysis of transient time series. In the DIANA project the theoretical background of time series analysis for the determination of the number of uncorrelated phenomena in operating data is given. Besides the eigenvalue decomposition of the cross spectral matrix, a new method by the analysis of model residual is also put forward. These procedures have been implemented in software code.

The basic hypothesis of this approach consists in considering the operating data as the output of a system with white noise input. Different models of system can be used, autoregressive model (AR) or autoregressive moving average model (ARMA).



$$U_n \rightarrow \frac{1}{1+\sum_{i=1}^{p} A_i \cdot Z^{-i}} \rightarrow X_n$$

Another approach is the linear prediction of the signal (the future vaiues can be determined by the knowledge of the time series). Basically, an error between real signals and synthesized signals is introduced.

Finally, the computation of parameters is obtained by :

$$Min\left[E\left(\underline{U}_n \cdot \underline{U}_n^T\right)\right]$$

Like in the FFT approach, the cross spectral matrix have to be calculated.
Considering an ARV model :

$$\underline{X}_n = -\sum_{k=1}^{p} \underline{\underline{A}}_k \cdot \underline{X}_{n-k} + \underline{U}_n$$

it could be rewritten like :

$$\underline{\underline{\mathbf{O}}}_{\mathbf{p}} \cdot \underline{\mathbf{X}}_{\mathbf{n}} = \underline{\mathbf{U}}_{\mathbf{n}}$$

A denotes the sampling interval of the data, $\underline{\underline{\sigma}}_U$ the autocorrelation matrix of $\underline{U}_n$ then :

$$\underline{S}(\omega) = \frac{\Delta}{2\pi} \cdot \underline{\underline{O}}_p^{-1}(\omega) \cdot \underline{\underline{\sigma}}_U \cdot \underline{\underline{O}}_p^{-H}(\omega)$$

The time domain principal component analysis has been implemented in software which is based upon following procedure.

1. Estimation of the model order.
2. Calculation of the model parameters.
3. Validity check of the model.
4. Time domain decomposition analysis.
5. Estimation of the cross spectral matrix.
6. Frequency domain eigenvalue decomposition.

The time domain PCA has been evaluated at a scale model of a car. The use of a model for testing is very attractive since different excitations, controlled by the users, can be applied. Multiple non-stationary inputs have been simulated by combining white noise with burst noise and white noise with hammer impacts. In figure 3 the principal components of a white noise and hammer impact are presented. One clearly distinguished 2 principal components. This method shows a lot of potential since only a limited data acquisition time is required. E.g. the data of figure 3 has been acquired in a time span of only 0.6 seconds.

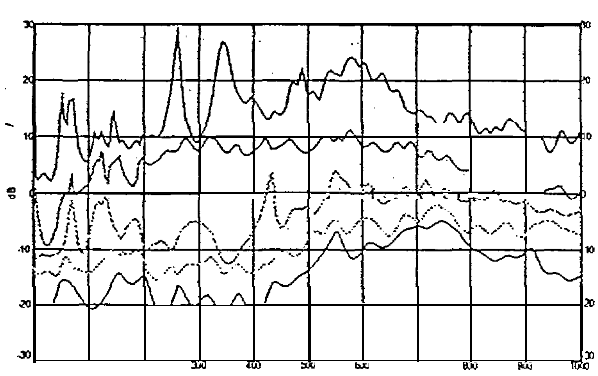


Fig. 3: Time domain PCA : Principal Components white noise and hammer impact (data acquisition period 0.6 seconds).

Nevertheless, one has to be cautious. Although very promising results are obtained, the results also depend very strongly of the choice of the processing parameters. Actually, for this study, the excitations were known and therefore the interpretation of the results is biased. More research is required for an automatic selection of the processing parameters. Nevertheless, time domain PCA appears as an attractive alternative for transient signal analysis.

5. Road Noise Analysis

Road noise deals with multiple independent and mutually uncorrelated excitation inputs due to the rolling of the wheels over the road surface.

The amount of energy transmitted from the wheels to the cars' interior is determined by the dynamic properties of the suspension and the body. This energy can be transmitted through several paths. The main transmission paths being the shock absorbers, suspension triangles, subframe and/or twistbeam connection points, To rank these transmission paths in importance a Transfer Path Analysis is carried out. For a TPA, operational vibrations need to be acquired.

A characteristic of wheel excitation is that several mutually incoherent inputs act simultaneously on the suspension. Due to these multiple uncorrelated inputs, the various phase relationships - between accelerations, between acoustic pressures, and between accelerations and acoustic pressures - are varying continuously. The classic single reference technique used to acquire operational vibrations for powertrain noise applications can not deal with multiple uncorrelated inputs. It is therefore necessary to separate the operational data into sets of 'independent phenomena" by means of a Principal Component Analysis (PCA).

This methodology can be summarised in following picture (figure 4).

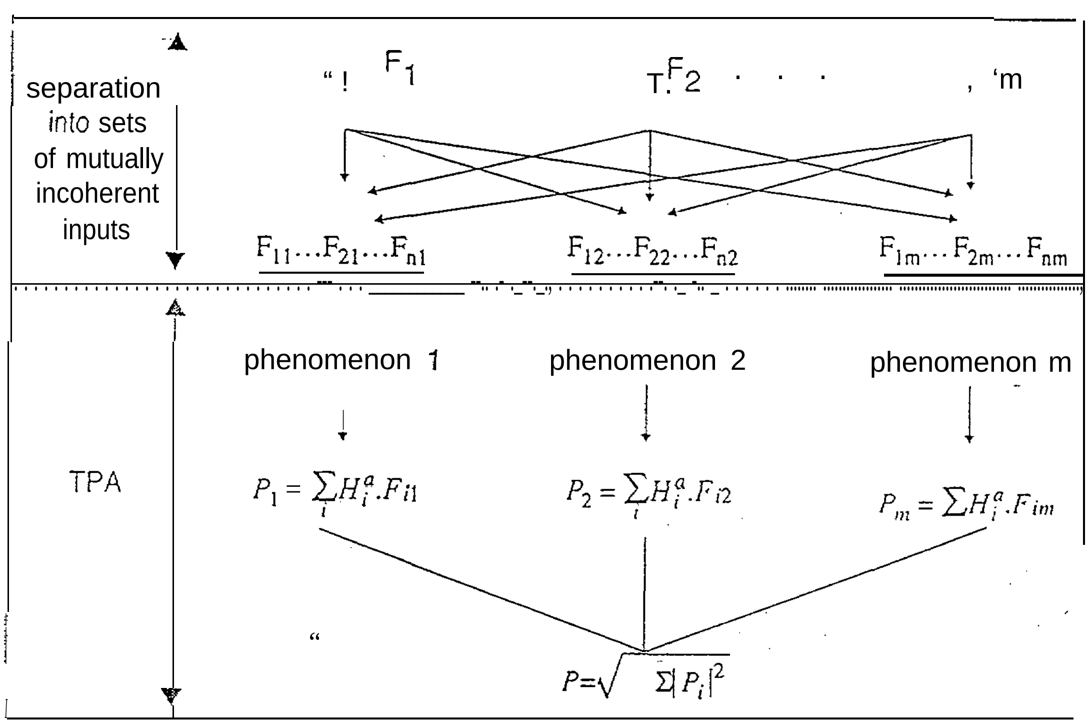


Figure 4: Road Noise Methodology: multiple input PCA.

This methodology has also been implemented in software code.
Principal component analysis aims at finding a set of unit linear combinations $[U]$ of the measured responses, resulting in a new set of uncorrelated signal spectra $[X']$ ·
the principal components :

$$[X(f)]=[U(f)][X'(f)]$$

The crosspower matrix of these uncorrelated signals should be diagonal.

$$[S''_{xx}(f)]=[X'(f)][X'(f)]^h$$

with *h :* hermitian

Diagonalizing the crosspower matrix of the original signals can be performed by computing the eigenvalues and related eigenvectors.

$$[S_{xx}(f)]=[U(f)][S'_{xx}(f)][U(f)]^h$$

$[U(f)]$ **is** the eigenvector matrix of crosspower matrix $[S_{xx}(f)]$ $[S'_{xx}(f)]$ is a diagonal matrix, containing the eigenvalues of $[S_{xx}(f)]$ in descending order. These eigenvalues are real and non-negative because $[S_{xx}(f)]$ **is** Hermetian. They can be considered as the autopower spectra of the principal components $[X'_i(f)]$ which are totally uncorrelated (crosspower spectra are zero).

The number of non-zero eigenvalues, Ns, can be considered as the number of "independent phenomena" interacting on the structure. The acceleration signals $[X'_i(f)]$ can be written as a linear combination of those "independent phenomena".

$$[X(f)]=[U(f)]_{Ns}[X'(f)]_{Ns}$$

The elements of each eigenvector can then be viewed as transmissibility functions from the principal components $[X'_i(f)]$ to the physical locations.

The virtual coherence function is defined as the ordinary coherence between the i-th signal and the j-th principal component.

$$\gamma'_{ij}=\frac{|S'_{ij}|^2}{S_{ii}\cdot S'_{ii}}$$

The virtual crosspower spectrum S'_{ij} (f) is **the crosspower spectrum** between the **th signal** and the j-th principal component.

The virtual coherence function is a helpful tool in finding the importance of the calculated principal components and estimating the number of important phenomena that contributes to the motion or sound pressure **in a specific location.**

If the sum of the first Ns virtual coherence approximates 1, the dimensionality in that location is estimated Ns.
The **sum** of all virtual coherence will equal the multiple coherence for each signal.

In practice a principal component analysis is carried out on the crosspower matrix of the references. For each principal component or “independent phenomenon”, the virtual crosspowers with all response points form the virtual operational deflection shapes. If the output signals are defined by a matrix $[Y(f)]$:

$$[S_{YX'}(f)] = [Y(f)][X'(f)]_{Ns} = [S_{YX}(f)][U(f)]_{Ns}$$

The virtual crosspowers are obtained by multiplying the ordinary crosspower matrix by the eigenvector matrix.

Out of the virtual crosspower, referenced or virtual spectra are calculated,

The number of references equals Ns and for each DOF Ns virtual spectra are obtained.

$$Y_{i,j} = \frac{S'_{ij}}{J \quad - \quad -}$$

Index i in the formulas above could be an acceleration or an acoustic pressure. For a certain principal component j all virtual spectra are fully coherent and spectra referenced to different principal components are uncorrelated.

In order to validate the Road Noise Analysis Methodology several tests at a scale model of a car have been carried out. However, the emphasis in the project was on car testing. The Road Noise Analysis has therefore been applied at 3 different cars. The aim of the road noise analysis is to link the mechanical motion of the wheels to the interior noise, i.e. to determine whether or not there is a causal relationship between the wheel vibrations and interior noise. This can be determined by performing a multiple coherence analysis.

With this objective, the test vehicles were instrumented with 4 microphones in passenger compartment and triaxial accelerometers at each of the wheel hubs. Approximately 120 seconds of operational response data was measured when the car was being driven along different road surfaces at a constant speed. Out of these data a 16 x 16 cross power spectra was determined with 300 averages in order to describe a nominally stationary condition.

The results of the multiple coherence analysis indicated that the interior noise induced could not be allocated to a single one of the 12 wheel vibration components measured : road noise in this test vehicle is caused by the vibration of all wheels in all directions, and all 4 x 3 wheel inputs are therefore required to describe the interior noise. What is important is to notice that the noise measured at different points in the compartment, caused by a complex but incoherent combination of 12 mechanical inputs, is coherent over the lower frequency range.
As regards road surfaces, several types were tested each at a constant speed.
Also the effect of different speeds to the interior noise has been investigated.

The next step in the analysis consists in determining the number of uncorrelated **phenomena (forces) which excite "the structure. Afterwards a set of reference transducers being representative** for all phenomena is selected.

These reference signals serve as a basis for the extraction of the virtual deflection shapes, the number of virtual deflection shapes being equal to the number of phenomena exciting the structure. These virtual spectra also serve as input for the multiple input TPA.
The physical interpretation of these virtual deflection shapes is facilitated by calculating the correlation between modal deformations and virtual operational deformations. Also the results of the multiple input TPA complete the global picture of the cars' dynamical behaviour.

The procedure for Road Noise Analysis can be summarised as follows :

1. Verifying causality wheel vibration and interior noise.
2. Determination of the number of phenomena.
3. Selection of a set of reference signals.
4. Determination of virtual deflection shapes.
5. TPA analysis for road inputs.
6. Physical interpretation of results': correlation virtual deformation/modal deformation/TPA results.

For the 3 testvehicles a clear understanding of the interior noise and its related dynamics has been obtained.

A first method to validate the analysis results of the TPA model is the comparison of the calculated and the directly measured interior noise. For the 3 vehicles the calculation and measurement correspond very well.

However, this first verification is not a guarantee that the contributions of the individual transferpaths are correct. In order to validate the accuracy of the individual transferpath calculations physical modifications have been applied which only affect a few transferpaths.

If the prediction of the modifications corresponds well with the measured effect this validates the importance of the affected transferpaths.

The acoustical contribution of a specific transferpath or partial pressure is written as the product of a car body vibro-acoustical FRF and operational force.

$$p_i = FRF_{P/F_i} \cdot F_i$$

[n order to affect the acoustical contribution of a particular transferpath pi one can alter the vibro-acoustical FRF and/or change the force F_i. At one test vehicle modifications at the car body have been investigated. This means that all forces F_i remained the same before and after the modifications. As a consequence **only the car's acoustical** sensitivity presented by the vibro-acoustical FRF, has been the object of this validation.

At the other vehicles, one did not affect the FRF but evaluated changes in the forces F i .

When only the vibro-acoustic transferpath is affected the modification results can be very accurately predicted. Road measurements proved that these physical modifications at the vehicle were quite successful. They are resulted in a noise reduction as could be expected by the decrease in the vibro-acoustic FRF.

Frequency	Operational Deflection	Transferpath	Modification	Reduction FRF	Reduction Road
28 Hz	hatch opening	rear twist beam mounts	number 1	-18 dB	-10 dB
28 Hz	hatch opening	rear twist beam mounts	number 2	-8 dB	-6 dB
40 i-i'	roof bending	rear twist beam mounts	no effective modification	-	
84 Hz	subframe bending	subframe mounts	number 3	-6 dB	-4 dB

Table 1: Road Noise Investigation: Overview Modification

Figure 5 presents the interior noise of this -test vehicle before and after the modification. One clearly notices a reduction of the low frequent booming of 6 dB.

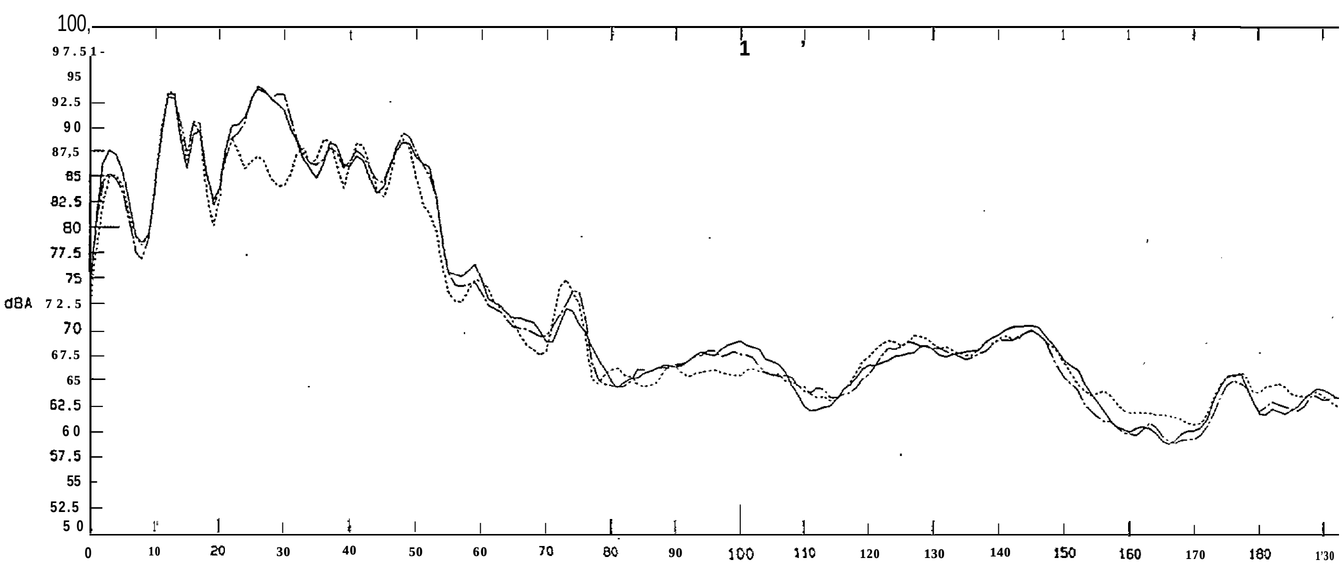


Figure 5: Road Noise Analysis test vehicle 1 : measured interior noise before (solid line) and after modification (dotted line).

At the other hand, when one tries to affect the operational forces one has to be more cautious. In order to check the validity of the TPA model and its application as a predictive tool, it was decided to modify the stiffness of the critical suspension mounts.

When e.g. switching to the nylon bushes an increase of 30 dB would be expected on the partial pressures due to the effected paths. As the sum of the effected paths is approximately equal to the interior noise predicted by the model, a large increase in the interior noise would be expected, and this was the effect predicted by the TPA

In practise, however, the change in the interior noise was quite small, with a deterioration order of a magnitude of 5 dB.

Similarly the change to softer mounts produced a very different effect to that predicted by the TPA. The model suggested that changes to the interior noise would be minimal while the empirical data showed large reductions of up to 18 dB.

As a typical example figure 6 presents the calculated effect of softer mounts and figure 7 presents the measured effect of this modification.

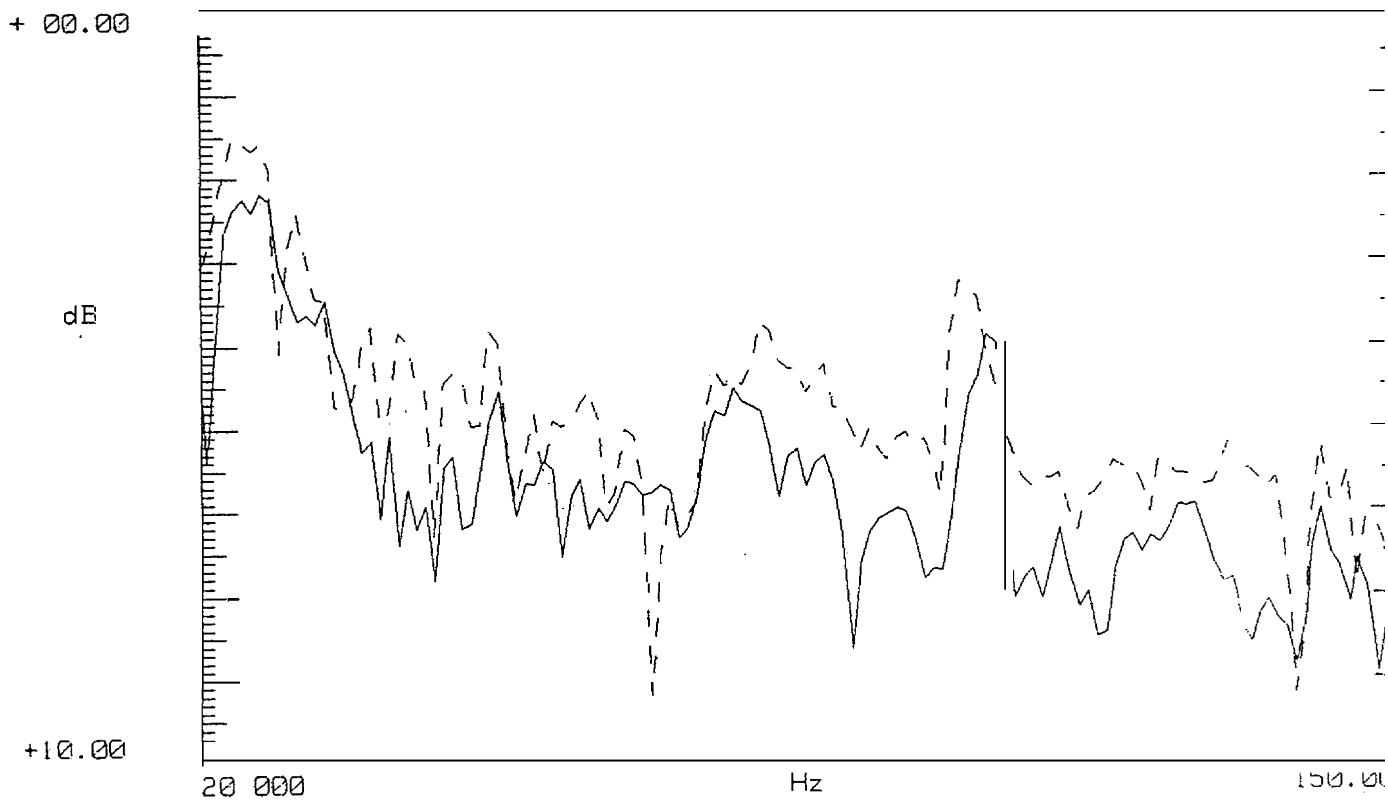


Figure 6 : Road Noise Analysis test vehicle 2 : calculated interior noise before (dashed line) and after (solid line) modification (softer mount).

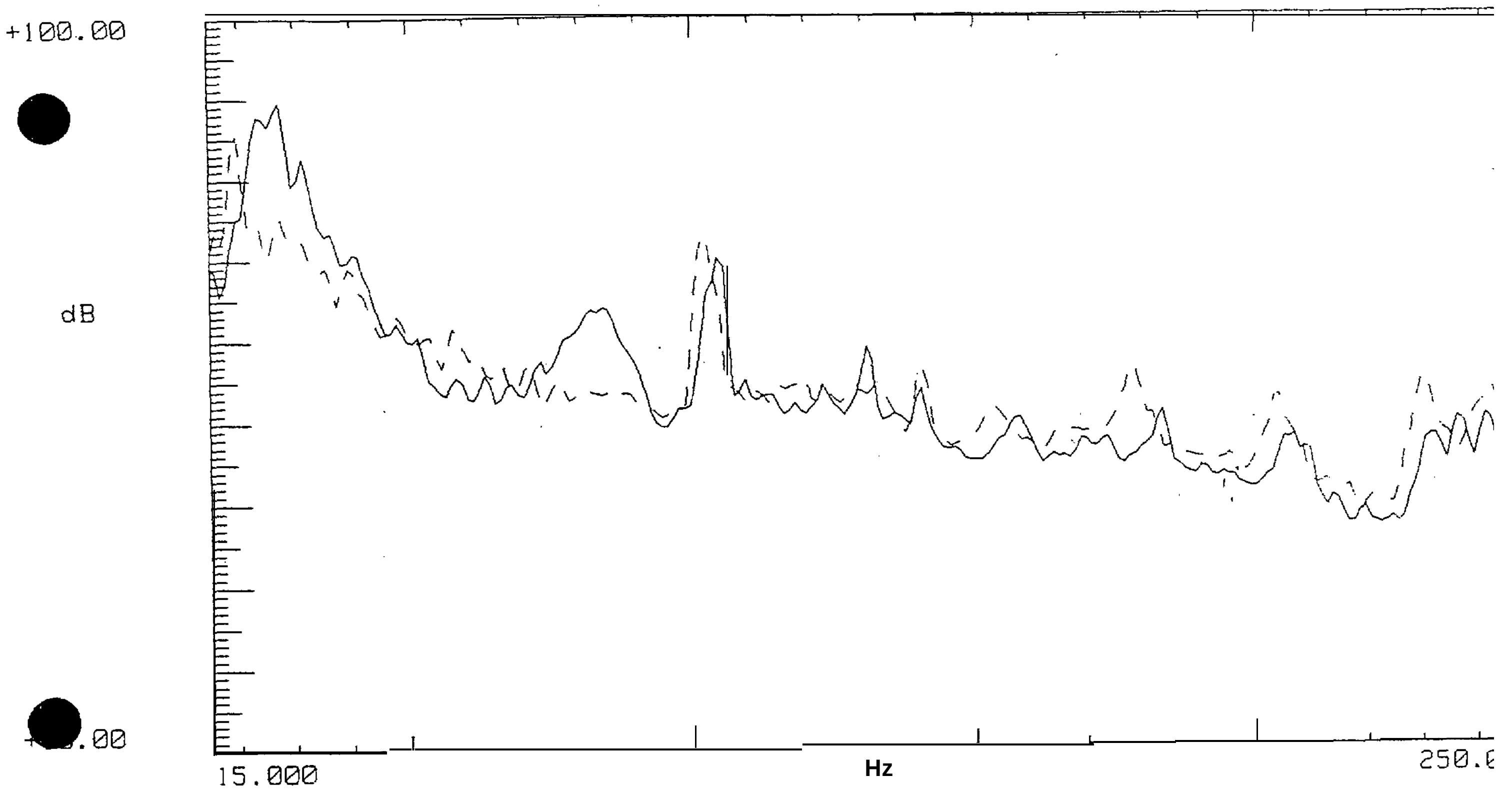


Figure 7: Road Noise analysis test vehicle 2: measured inferior noise before (solid line) and after (dashed line) modification (softer mount).

In both cases the TPA model failed to accurately predict the magnitude of the changes produced on the vehicle, though the trend was as expected. This suggested that there could be errors in the TPA model and further investigations were done to determine the reasons for the discrepancy. These indicated that the problems were due to the underlying assumption of linearity made by the TPA model.

The TPA prediction assumes that the subsystem exciting the vehicle is undergoing freebody vibration where its response is completely mass controlled, with no influence from the isolation bushes. Thus the input forces to the attached vehicle become directly proportional to the dynamic stiffness of the mounts, hence the model is linear.

On changing the rubber mounts to nylon, however, there was a significant change to the displacement across the mount. The acceleration at the body side of the bushes changed by 2 dB while the suspension side of the same mount saw a reduction by 20 dB. The underlying assumption required for prediction using the TPA model were therefore invalid in this context, as the system reacted to the modifications in a non-linear manner,

6. Mount testing technology

A number of environmental variables were identified which affect the dynamic performance of visco-elastic mounts installed in vehicles. These derive from both the instantaneous operating conditions for the mount and the product history. Thus temperature, load rate and multi axis preloads are likely to be the result of operation while material composition, age and static load are affected by the installation history of the product.

The project investigated the effect of following boundary conditions to the dynamic stiffness of rubber mounts :

- Production variability
- Preload magnitude
- Creep
- High stress cycling
- Operating temperature
- Multi-axis preload
- Road load simulation
- Rotational stiffness
- Hydraulic mounts

To support this testing two rigs *were* developed which could be capable of investigating these more complex parameters.

At MIRA an existing test rig was modified to allow both temperature and preload to be accurately controlled. This provided a single axis preload and test capability, but could not cope with the more complex condition of triaxial preload and testing which would more completely describe the in-vehicle situation.

At the Technical University of Bielefeld a more advanced testrig was designed to provide these facilities, and centred around the use of a hydraulic shaker on a moveable cross-head to provide the primary excitation and lateral loading capability (Fig. 8).

This type of actuator has the advantage over an electro-dynamic shaker that the actuator bearings can withstand the high levels of lateral loading associated with multiaxial preloads. The use of a crosshead mounting for the actuator and a bed plate for the mount also provide a low frequency capability missing from the MIRA rig, which uses a dead weight as the reaction load, This was considered vital for the investigation of the roadload simulation and was therefore an important aspect of the design. The lateral excitation was provided by the two electro-dynamic shakers mounted off the bed plate.

The novel nature of this testrig resulted in a unique test facility capable of the simulation of the complex conditions found in vehicles. A great deal of work was required, however, during its development, in order to identify the resonances associated with the rig structure, hydraulic shaker/oil column and mount fixtures. A patent has been applied for this advanced testrig.

In order to evaluate the influence of the rotational degrees of freedom for the sound pressure in the cabin a comprehensive TPA of a complicated Engine mount was carried out. This mount is characterised by the fact that the distances between the three connection points on the body side are unusually large. Therefore it was expected that the rotational degrees of freedom will exert an influence. The results have been compared with a TPA based only on an imaginary central connection point, where the rotational degrees of freedom on the engine side are ignored.

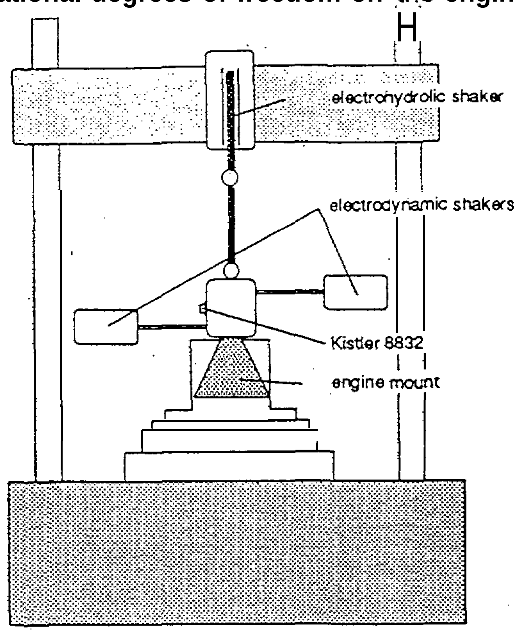


Figure 8: Technical University Bielefeld : Multiaxial rubber mount testrig.

7. Conclusion

The DIANA project aimed studying "Development & Integration of a Unified Approach for Structure borne noise Analysis". In the project 5 partners collaborated (LMS, RNUR, CRF, MIRA, FH Bielefeld) and the sponsor Ford provided testvehicles.

The objective was to develop, implement, evaluate and validate an integrated approach for structure borne noise analysis in cars, This approach was based upon existing techniques and newly to develop techniques.
The goals to be achieved were threefold :

- Evaluation and limitation setting of existing techniques.
- Development of new techniques.
- Integration of different techniques into one global methodology.

The first objective of the project included the investigation of the advantages, sensitivity to boundary conditions and limits of confidence of general classical techniques in the field of structure borne noise analysis at several testvehicles. These techniques were amongst others : single input, transferpath analysis, principal component analysis, mount testing.

The newly developed techniques were related to algorithms for indirect force determination, high quality FRF measurements, time domain principal component analysis and advanced mount testrig.

Finally, the different techniques have been integrated into two global methodologies : Engine Noise Analysis and Road Noise Analysis. This integration resulted in a new software code.

As a general conclusion all partners agree that the project has been carried out as originally planned and that the objectives are reached.

8. Acknowledgements

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