

## DESIGN TARGET CASCADING FOR VIBRO-ACOUSTIC CONCEPTUAL DESIGN OF AN AUTOMOBILE SUBFRAME

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### 1. Introduction

Automotive vehicle is considered as a large complex system to design. When designing such a system, it is desirable to decompose it into its constituent subsystems and design each subsystem separately in order to allow concurrent engineering. Thus, we have the capability to predict the effects of design changes on a single component without having to assess the entire system performances. The use of concurrent engineering can reduce the product "design-cycle" time [Kusiak *et al* 1990]. To practice concurrent engineering, it is important to develop a systematic means for propagating the desired system level specifications into performance specifications at the subsystems level [Mahmoud *et al* 2002]. A hierarchy is typically assumed in the synthesis of complex products, and so this (performance) target translation can be seen as a cascading of targets down the hierarchy [Papalambros 1999]. Target cascading in the early product development stages is defined to be a systematic effort to propagate the desired top-level system design targets to appropriate specifications for subsystems and components in a consistent and efficient manner [Kim *et al* 2001]. Under the target cascading concept, product design can be viewed as a four-step process: (i) specify overall product targets; (ii) propagate product targets to system, subsystem and component "sub-targets"; (iii) design system, subsystems and components to achieve their respective sub-targets; and (iv) check that the resulting product meets overall product targets [Michelena *et al* 1999].

In target cascading there is a matter of target consistency. Consistent subsystem targets will ensure that, upon assembly, the different subsystems achieve the desired system-level performances. This is why target cascading must be performed in an "efficient" and "consistent" manner to avoid iterations at the later stages of the product development cycle and to ensure that different subsystems ultimately work together.

Major challenges of target cascading when designing the automotive structure are (1) the high level of interaction between subsystems which is the fact that system performances such as vibro-acoustic performances result fundamentally from the dynamic behavior of coupled subsystem structures and (2) the increasing number of targets at the subsystems level. It is typically the case of the vehicle NVH (Noise, Vibration and Harshness) performances, where the performance targets at the vehicle level are translated into an increasing number of targets at the subsystems levels (car body, chassis, tires, engine, transmission,...). Moreover, these subsystems remain in most of cases dynamically coupled when assembled together. Mahmoud *et al* [Mahmoud *et al* 2002] mentioned that *the presence of coupling, or feedback loops, between the different subsystems implies that the process of setting targets is not guaranteed to work if these feedback loops are not accounted for, i.e. it is possible to meet the different subsystem targets and yet fail to meet the overall system targets*. One way to establish NVH performances of the automotive subsystems is measuring either the Euclidean norm (i.e. 2-norm) or the infinity norm of their Frequency Response transfer Functions (FRF)  $\mathbf{H}_{ij}(f)$  from

each joint input direction  $i$  to each direction of each output joint  $j$ . These input and output directions are further named *dofs*, for degrees of freedom. Further,  $\mathbf{H}(f)$  denotes the whole FRF matrix. FRFs are results of the frequency response analysis which is a method used to compute the structural responses (outputs) to steady-state oscillatory excitations (inputs). In the frequency response analysis, the excitation loads and the responses are explicitly expressed in the frequency domain. Setting targets on the transfer functions between each input dof and each output dof can lead to a dramatic increase in the number of targets at the system and the subsystems levels.

To reduce the number of targets on the subsystem structure level, we use a popular measure in optimal control theory [Gawronski 1998] which is the  $H_2$  norm of the FRF matrix. This aggregated metric (1) reduces the dimensionality of targets on the subsystem level, (2) keeps the design problem tractable, letting it easy to make decisions and carry out an optimization because of the reduced number of constraints and (3) helps engineers to communicate about the results in the conceptual design stage. Avoiding any feedback on design targets is made possible by the proposition of independent targets over subsystems. For that purpose, we apply the approach proposed by Mahmoud *et al* [Mahmoud *et al* 2002] that places bounds on the magnitude of the feedback signals in order to achieve some decoupling conditions.

In the literature, there is few target cascading applications on practical design case studies that apply target reduction and target balancing using norms previously mentioned. In this paper, we use such an approach in the context of the interior road noise performance with an aggregated metric and techniques of fast performance assessment. After presenting this approach, we apply it to the case study of an automotive front subframe. The approach seems to be promising, especially in the conceptual design stage.

## 2. The vehicle road noise example of target cascading

One important automotive vehicle performance is the vehicle NVH (Noise, Vibration and Harshness). The virtual assessment of vehicle NVH performances requires an advanced physical phenomena modeling of the dynamical behavior of the automotive mechanical structure. Besides, there is a higher level of interaction between subsystems structure. One of the scope of vibro-acoustic design of the automotive vehicle is to keep its interior noises, which impact directly the customers satisfaction, to an acceptable desired level when the vehicle is subject to external and internal excitations. Automotive acoustic engineers distinguish different types of vehicle interior noises depending on their sources and their frequency ranges. One of them is the road noise which is caused by the contact between the tire and the road. The road noise (see figure 1) is transmitted through structure-borne and airborne paths. As we focus on the structure-borne path of the road noise, the factors that control this noise path after [Dong *et al* 1999] are: (1) the input level from the excitation source that is here the wheel center displacements due to the road irregularities, (2) the attenuation mismatch efficiency of the rubber silent-block isolators and the influence of their stiffness and their damping and (3) the dynamic structural integrity and noise sensitivity of the body and the chassis at their interfaces. As mentioned in the third factor, during the conceptual design stage a better dynamic structural design of the body in white and the chassis can be of great importance in the reduction of the road noise transmission and can avoid major structural modifications in later design stages that can be expensive and time consuming [Yannou *et al* 2003]. This is why one has to develop conceptual design models to early assess the effects of the design changes on the subsystem level onto the system vibro-acoustic performances while avoiding to assess the performance of the overall system [Ochsner *et al* 1993].

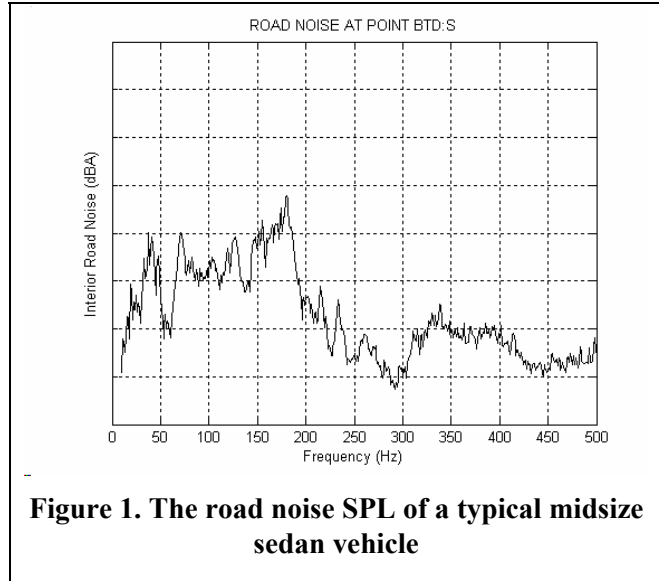
In the automotive industry, the SPL (Sound Pressure Level) target of the interior road noise performance of a vehicle project is set after a benchmarking of the concurrent vehicles. This pure acoustic target can be validated only in later design stages when full detailed virtual prototypes or physical prototypes are available. Before that stage, NVH designers set overall vehicle targets using a numerical vibro-acoustic model that consists in a set of F.E. models (Finite Elements) of the chassis and the body coupled with a three-dimensional acoustic cavity model of the passenger compartment. But, in the early design stages of the body and the chassis, pre-processing and simulation of F.E. vibro-acoustic models with an acoustic cavity is a hard and time consuming task, especially when the topology and the dimensions of the components are not completely fixed and can be the subject of significant further modifications. Then, a structural (physical vibration) response target over the vehicle (the F.E. Model of the body in white and the chassis assembled together) is considered instead

of an acoustic response. A vibration response target is an empirical target that turns out to be robust enough after a number of experiments carried out by the design team on the subject. In the next chapter, we detail this structural target and the way it is propagated onto the subsystem levels.

### 3. Establishing Target Cascading

System decomposition is the first step to establish target cascading. We consider in the example of figure 2 that the system is composed of 4 subsystems:

- Subsystems A and B are, respectively, the front and rear subframes of the vehicle. We further focus on the front subframe as a case study for targets on the subsystem level.
- Subsystem C consists of the bushing joint models. Three simple translational springs have been used to model these coupling joints at each structural component interface.
- Subsystem D is the body in white structure.



**Figure 1. The road noise SPL of a typical midsize sedan vehicle**

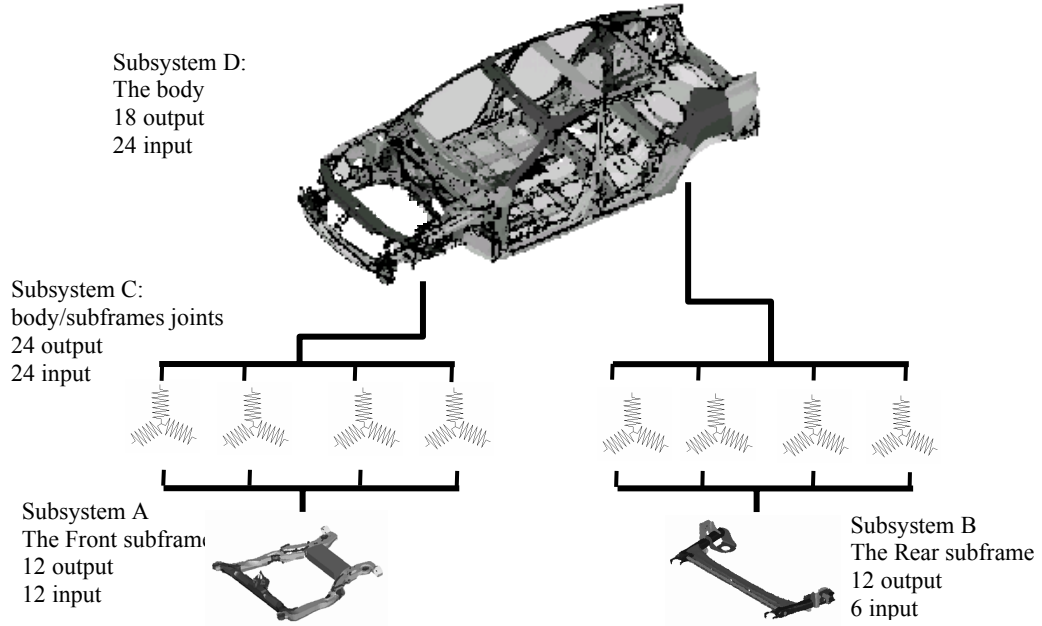
Each subsystem is a Multi-Input-Multi-Output (MIMO) dynamical structure. They constitute, when assembled together, the structure-borne paths (see figure 2) through which the structural vibrations are transmitted from the input joints to the output points. For the overall system, external excitation forces, at the input joints of the subframes, are PSD (Power Spectrum Density) functions that represent the random process of the road noise excitations. Typical excitations of a rough road are modelled by a monotonically decreasing PSD function (relatively to the frequency).

About system output responses, the most direct contributor components of the car body to the acoustic performance in the low and medium frequency range are the body panels. By a simplified analysis, controlling and lessening the low and medium frequency responses of the body in white panels leads to a lesser acoustic response, providing there is no high coupling between structural and acoustic resonance. In order to have a target of the overall system (body and chassis) expressed by an unique real value, we chose the assessment of the 2-norm (eq. 1)<sup>1</sup> of all the structural responses of the body in white panels and contours of these panels.

$$\|H\|_2 = \sqrt{\int_0^{f_{\max}} \text{tr}(\mathbf{H}^*(f)\mathbf{H}(f))df} \quad (1)$$

At the subsystem levels, having targets over each FRF (single input/single output function) leads to  $N \times M$  targets (with  $N$  the number of inputs and  $M$  the number of outputs). In order to lower this important number to an unique real value target, we also use the 2-norm of the FRF matrix as a performance measure at each subsystem level.

<sup>1</sup> In this equation,  $\mathbf{H}^*$  denotes the complex-conjugate transpose of  $\mathbf{H}$ .



**Figure 2. The vibration structure-borne paths of the road noise**

Another important property of the former established targets is the fact that, under the sufficient condition of weak coupling between the structural subsystems (here A/B and D), these targets are independent from the coupling feedback observed when the overall system is assessed. Then, optimizing the subsystem designs to achieve the subsystems targets guarantees to achieve the overall system target. The reader may refer to [Mahmoud *et al* 2002] for a theoretical view.

#### 4. The front subframe case study

Let us take the case study of the conceptual design optimization of a front subframe structure (see figure 3a), a subsystem that is imbedded in the front chassis. We set the subsystem target (see eq. 2), the 2-norm of the subframe FRFs, as a constraint in the optimization problem and the subframe mass as an objective function to minimize. We calculate the subframe FRFs by simulating with MSC/NASTRAN a Finite Element (FE) model. We impose as inputs equivalent road noise excitation forces on the structure of the subframe (in fact on the three translational dofs of the 4 joint interfaces with the arms, see figure 3a). We take as output responses the accelerations of the three translational dofs at the 4 joint interfaces of the subframe with the body. So, in each optimisation loop, 144 FRFs are computed.

$$\|\mathbf{H}(f)\|_2 = \sqrt{\int_0^{f \max} \text{tr}(\mathbf{H}^*(f)\mathbf{H}(f))df} = \sqrt{\int_0^{f \max} \sum_{i=1}^{12} \sum_{j=1}^{12} |\mathbf{H}_{ij}(f)|^2 df} \quad (2)$$

We take as design parameters (a) the thicknesses of the shells that constitute the side-members of the subframe (see figure 3b) and a topological parameter that consists of varying the lateral side-members length ratio (*Alpha*).

**Table 1. Design Parameters and their potential domains**

th1	th2	th 3	th 4	th 5	th 6	th 7	th 8	th 9	th 10	th 11	alpha
[1..3]	[1..3]	[1..3]	[1..3]	[1..3]	[5.6..9]	[1..3]	[1..3]	[1..3]	[1..3]	[1.5..4.5]	[1..1.25]

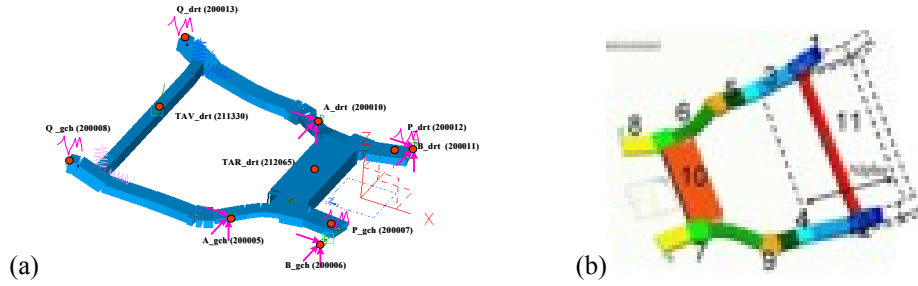


Figure 3. The car subframe: (a) its input and output points, (b) the design parameters

## 5. Performance assessment via metamodels

The assessment of the objective function within every optimisation loop is a time consuming task because of the FE simulation time (more than 20 minutes for computing the 144 FRFs). A serious optimization becomes intractable. To solve this problem, we used a mathematical approximation form, more known in the literature as a metamodel [Simpson *et al* 2001]. Indeed, metamodels are used in engineering as fast tools of performance assessment with an acceptable accuracy especially for conceptual design purposes. To establish a metamodel, three sequential steps are required [Simpson *et al* 2001]: (1) conducting a design of experiments (DOE), (2) choosing the functional form of the metamodel and (3) fitting the metamodel to the DOE data. Figure 4 shows the application of the metamodeling process to our case study (see [Yannou *et al* 2003] for a detailed description of this process). The resulting metamodel assesses the subframe FRFs 2-norm in a shorter time than the original model (lesser than 1 minute). This is why we use this metamodel in the topological optimization of the sub-frame.

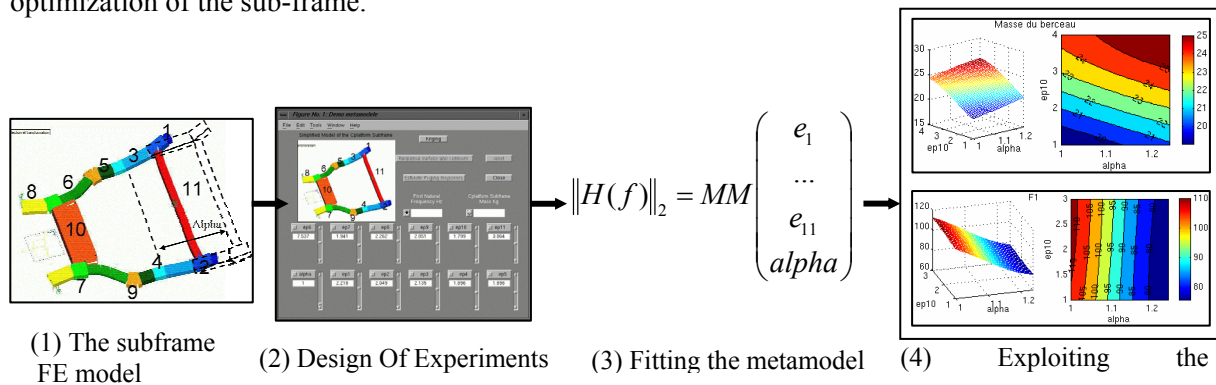


Figure 4. Metamodeling process applied to the subframe case study(see also [Yannou *et al* 2003])

## 6. Results

After optimization of the subframe design, we illustrate here three worthy results. Figure 5 is the plot of the cumulative value of the 2-norm relatively along the frequency. This plot shows three cases:

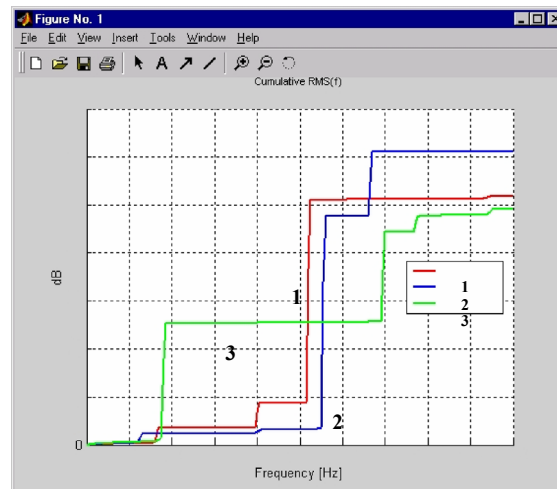
- (1) The optimal design case, the red curve #1. The NVH target constraint is respected with a significant lowering of mass.
- (2) The best mass reduction, the blue curve #2, but NVH target is not respected.
- (3) The best NVH performance, the green curve #3. But the mass has been dramatically deteriorated.

An a posteriori checking has been made that this local subframe optimization has positively contributed to improve the overall NVH target of the road noise performance at the system level.

## 7. Key conclusions

The design purpose considered in this problem is to minimize the interior noise level due to the road noise over a fixed low frequency range. Setting NVH targets over subsystems using norms and avoiding feedback coupling effects are of a great help for designers for being confident that the vibro-acoustic design of their subsystems is possible and will be efficient in a global assembling. There is a

guaranty to achieve the overall NVH system target. This methodology contributes to avoid expensive design modification loops in the detailed design stage.



**Figure 5. The subframe cumulative FRFs 2-norm**

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