

Optimizing Body Panels for NVH Performance

Murali Balasubramanian and Ahmed Shaik
Chrysler India Automotive Pvt, Ltd.

ABSTRACT

Automotive manufacturers are being challenged to come up with radical solutions to achieve substantial (30-35%) vehicle weight reductions without compromising Safety, Durability, Handling, Aero-thermal or Noise, Vibration and Harshness (NVH) performance. Developing light weight vehicle enablers have assumed foremost priority amongst vehicle engineering teams in order to address the stringent Fuel Economy Performance (FEP) targets while facilitating lower CO₂ emissions, downsizing of engines, lower battery capacities etc. Body sheet metal panels have become prime targets for weight reductions via gage reduction, high strength steel replacement, lighter material applications, lightening holes etc. Many of these panel weight reduction solutions are in sharp conflict with NVH performance requirements. The main challenge for NVH engineers is to recover panel stiffness and mitigate the potentially increased air-borne as well as structure-borne noise transmissibility thru these lighter panels. This is achievable with a systematic approach to optimizing panel geometry and damping treatment upfront during the body structure development process while also incorporating innovative new light weight solutions for acoustic insulation.

This paper describes generic FE based methods to virtually engineer body panels starting with optimization of geometry features followed by efficient damping treatment to meet weight and stiffness targets upfront in the design process. Panel geometry study parameters include panel features such as curvature, form geometry and beads in combination. Mesh morphing to vary panel curvatures coupled with topography methods to optimize features such as bead pattern are used to achieve panel frequencies and FRF targets. Further, a new methodology is introduced to assess panel sensitivities to mid-frequency structure borne noise which is then used to fine tune the panel features for stiffness as well as identification of target panel areas for efficient damping treatment. Also, innovative concepts for weight effective acoustic insulation are referenced.

CITATION: Balasubramanian, M. and Shaik, A., "Optimizing Body Panels for NVH Performance," *SAE Int. J. Passeng. Cars - Mech. Syst.* 8(3):2015, doi:10.4271/2015-01-2265.

INTRODUCTION

As the global spotlight shines on worldwide climate changes affected by the greenhouse gas (GHG) emissions, developed and developing nations alike are scrambling to meet their Y2020 GHG emissions targets discussed /agreed on at the Kyoto protocol (1997) and reinforced at the Copenhagen accord (2007) and beyond. The global glide path strategy is to stabilize atmospheric CO₂ levels to around 450ppm [1]. The global automotive industry (a key participant in this endeavor) is racing to improve vehicle fuel economy, meet stringent environmental regulations in terms of lower CO₂ emissions while minimizing cost impact on consumers. Automakers have a steep challenge to achieve ~40% higher Corporate Average Fuel Economy (CAFE) standards in US by Y2020 as compared to MY2012 levels [2]. Along with hybrid electric vehicles, alternate fuels such as CNG are seeing increasing adoption worldwide, however, the overwhelming focus by car manufacturers currently is still in improving FEP of their conventional engine based vehicle fleets to achieve the ~5mpg CAFE improvement for their MY2016-17 releases [2]. Apart from targeting improved external aerodynamics, higher efficiencies in engine combustion, powertrain torque transfer and thermal management, vehicle engineering teams world over have

the onerous task of achieving ~30% vehicle weight reductions for their new vehicle fleets towards achieving the desired FEP levels. Lighter vehicles also provide the additional opportunity for downsizing of engines, lowering of battery capacity, lowering the fuel consumption and thereby lowering emissions too.

NVH performance requirements are sometimes in sharp conflict with typical vehicle weight reduction concepts. Conventional combustion engine vehicles with increased energy efficiency and lighter body panel stampings are especially vulnerable to higher in-cabin airborne and structure-borne noise levels in the mid and high frequency ranges. Optimum panel mass and stiffness distribution, efficient damping treatment, lighter and multi-layered acoustic insulation applications are the typical strategies used to balance NVH performance against weight reduction goals. A systematic approach upfront in the vehicle concept design phase with the aid of advanced CAE tools and techniques is required to manage multiple and especially such conflicting performance objectives. A typical CAE based process for Body-in-White (BIW) concept development is depicted in [Figure 1](#).

This paper will propose a systematic approach to body sheet metal panel optimization for NVH objectives and is organized as follows. Firstly, the panel curvature effects are highlighted and coupled with bead /form geometry development using a topography method to achieve generic BIW panel modes targets and minimize low frequency driving point FRFs (panel point mobilities). A DOE scheme is used to demonstrate the optimum curvature and bead scheme for a wagon roof panel. This is then followed by evolution of a panel damping optimization scheme using the combined structure-borne panel contributions for powertrain or road noise using the panel Equivalent Radiated Power (ERP) parameter. The application of above methods on a D-sedan dash panel and a B-segment vehicle spare wheel tire tub panel are highlighted.

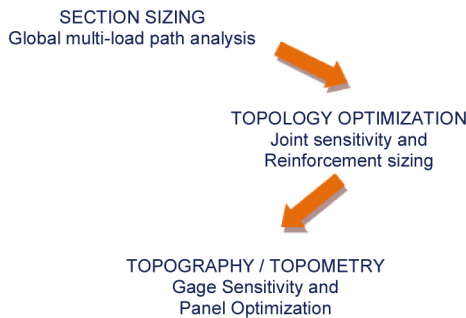


Figure 1. Typical body concept development process using advanced CAE tools and techniques

PANEL STIFFNESS OPTIMIZATION

Curvature Effects on Panel Stiffness

It is well known that curved panels have significantly higher stiffness and panel vibration levels can be reduced with appropriate bi-directional curvature. For better understanding, let us consider a simply supported untrimmed, undamped bare rectangular plate with simple finite bi-directional curvature as shown in [figure 2](#).

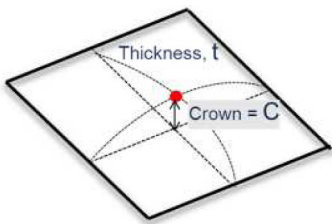


Figure 2. Simply supported rectangular plate of thickness t and small crown C with load applied at center over small area.

Bending stiffness B_c for the simply supported rectangular curved panel with uniform cross-section can be written as,

$$B_c = \frac{(Et^3)}{12(1-\nu^2)} * \alpha_c \quad (1)$$

$$\text{where Stiffness Ratio } (\alpha_c) = \frac{K_c}{K_f} = F(C, t) \quad (2)$$

considering $C \ll$ plate dimensions, small deformations of the plate and where K_c represents curved panel stiffness, K_f the flat plate stiffness for the same material (Young's Modulus E , Poisson's ratio ν) and thickness t . Stiffness ratio α_c is a function of curvature crown C and panel thickness t , represented by $F(C, t)$ and equals 1 for flat plate.

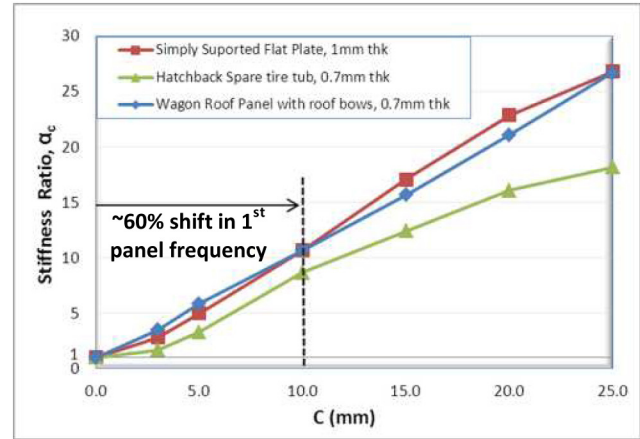


Figure 3. The panel stiffness ratio α_c versus C (in mm)

[Figure 3](#) depicts an almost linear relationship between the stiffness ratio α_c and C for a simply supported flat plate. The stiffness ratios for a spare wheel tire tub panel and a roof panel are also shown in [figure 3](#), calculated with BIW FE models from the driving point FRF responses averaged around the panel center for a low frequency band. Stiffness increase is of an order of magnitude for a 10mm crown. Also, with increased curvature, lower panel modes shift to higher frequencies. Higher mode counts are seen shifting to typically beyond 100 Hz and into the 150-300 Hz range thus significantly reducing or mitigating any potential low frequency booming noise. However, this shift in panel resonances to higher frequencies could potentially affect road noise performance adversely due to increased acoustic power and radiation efficiency for curved /beaded panels relative to flat panels in the aforementioned frequency range. Kato et al recommend the use of soap film geometry [3] for panel curvature to induce uneven panel modes and thereby reducing the radiation efficiency in the higher frequency ranges. Also, increasing lower panel frequencies beyond 100 Hz significantly improves the insulation efficiency of the sound absorption material used in cabin acoustic treatment.

Panel Bead Development using Topography

Beads and embosses like curvature are relatively inexpensive, weight neutral panel features that can be added as part of the sheet metal forming process to tune the panel stiffness for optimum NVH performance. Onsay et al [4] used FEA-SEA methods to study the effects of bead geometry & pattern on panel vibro-acoustic characteristics. They proposed guidelines for simple bead development strategy on panels to balance lower structure borne panel vibrations in low and mid-frequency ranges while cautioning the need to keep the radiation efficiency in check to contain the high frequency radiated noise. However, the presence of acoustic

insulation which always is the case for panels exposed to cabin have significant effect in attenuating the mid-frequency panel vibrations by about an order of magnitude [5] even for “light weight” trim thus more than countering the increased radiation efficiency in mid-frequency ranges. A more desirable strategy to optimize panel stiffness is to use an optimum combination of curvature, beads and embosses to achieve the requisite low frequency vibro-acoustic performance balance.

Methodology

In this section, we showcase a generic method to evolve bead patterns and sizing on body panels based on topography which is applicable on dash, floor, roof, body side panels etc. and demonstrate the method using a dash panel example. Later, in this paper, we explore a DOE scheme to optimize curvature and bead pattern using a roof panel example.

Modal analysis is used to extract the modes up to 500 Hz for the trimmed FE body model and composite mode strain energies are calculated based on linear superposition of the elemental strain energies for the panel sets in question as shown in Figure 4 in line with the methodology used by Subramanian et al [6]. The panel strain energy map is used to define design sub-space for topography as well as locate driving point mobility locations for monitoring. FE sub-model of the plain body (dash) panel is then sub-divided into design and non-design space as shown in Figure 5 and the design space FE mesh is refined to a smaller element size. The fundamental panel mode (dash panel in this example) is the objective function or alternatively a low frequency-band dynamic stiffness parameter can be used. The bead pattern parameter variables are defined in terms of width, height, and draw angle within manufacturing limits and with bi-linear pattern constraint i.e. cross-car and vertical direction beads only for dash panel.

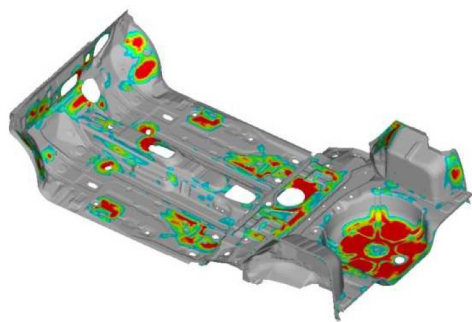


Figure 4. Body panel composite mode strain energy map

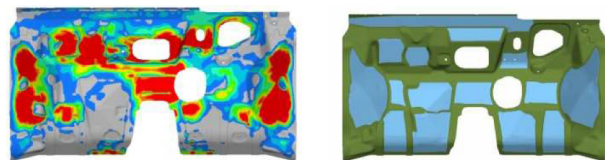


Figure 5. Dash panel composite strain energy map and design space segmentation for topography study (design space for beading in blue)

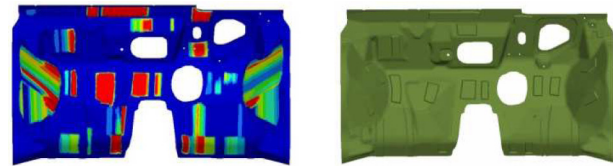


Figure 6. Topography bead contours and bead pattern definition,

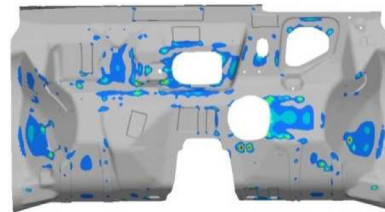


Figure 7. Composite strain energy map post topography optimized bead definition

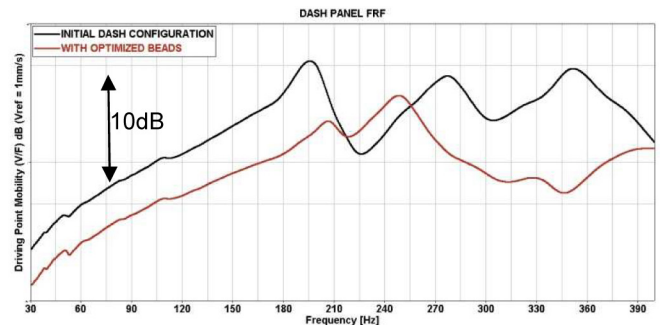


Figure 8. Dash panel velocity comparison between original and topography optimized bead pattern

Figure 6 shows the topography results for bead contours and a preliminary bead pattern definition based on the contour map. The composite strain energy map for the bead-stiffened dash panel and the panel vibration attenuation plot are depicted in Figures 7 and 8 respectively.

PANEL DAMPING TREATMENT OPTIMIZATION

Panel vibration attenuation in the low-mid frequency range (100-500Hz) is achieved thru viscoelastic layered damping treatment in select flat areas on the panel that show high contribution in terms of transmitting structure-borne noise into the cabin. The key step is to identify and optimize the area for treatment on the panel of interest followed by the selection of appropriate weight efficient damping composite for application on the identified optimum area. Area of damping treatment on panel is typically derived based on panel contribution analysis considering the various structure-borne sources transmitting noise into the cabin. Guj et al [5] proposed a CAE based method using NASTRAN for evolving the panel damping layout. They calculated the rms mobility (V/F) at each grid on the panel of interest averaged across the required structure-borne noise load cases for important modes and then normalized the response to capture vibration contour map on the panel in terms of the percentage mobility contribution at each grid w.r.t the overall response. They also account for the effect of trim loss and insulation package on the panel mobilities with the help of FE based implementation of Atilas