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Design of Folding Seat Tubs for NVH

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ABSTRACT

Recently introduced "Stow-'n-Go" feature in minivans provides the option of folding the 2nd and 3rd row seats flat into a special compartment. These special storage compartments, or "tubs", are designed under many challenging and competing design requirements, one of which is noise and vibration. In this study, both experimental and analytical tools are used to study the NVH performance of seat tubs considering different materials, constructions and damping treatments. The challenge of balancing stiffness and damping in a tight packaging space is augmented by the minimum weight and cost requirements. The details of material selection process for the minivan tubs are presented considering different materials and damping treatments. Various design alternatives considered during the optimization of weight, packaging space, and NVH performance are discussed. Results of the component and the vehicle testing are complemented with SEA modeling.

INTRODUCTION

The minivan seat tubs are designed to enclose the 2nd and 3rd row seats fully in their folded positions. These tubs are designed with just enough volume to contain the seat(s), but not to interfere with ground clearance or the interior storage space. The seats fold flat, level with the floor surface. These requirements place many constraints on the packaging space, structural design of the floor and the seats, the materials and the NVH treatments. In addition to NVH, there are many other factors considered during the design of the seat tubs; such as cost, weight, stiffness, durability, load capacity, packaging, exhaust-heat, crash, etc. All of these attributes are optimized while fitting the tubs in a very tight packaging space. During manufacturing these tubs are installed separately, in the trim-shop, dropped into the cutouts on the floor panel between the rails, and bolted on to the floor and sealed at the perimeter, without using spot-welds.

The design constraints listed above create a formidable task in packaging NVH treatments, such as beads for low frequency mode management [1], damping treatments for mid frequency structure-borne vibration, and carpet system for air-borne noise transmission and absorption. All of these NVH treatments and design features are required to fit in a very tight design space. On the other hand, for NVH, the seat tubs are one of the most important components of the vehicle, since they make up nearly 60% of the floor area (including 2nd and 3rd row tubs). The design of these tubs poses a serious challenge for both air-borne noise transmission and panel vibration. Because of the relatively large cabin space, a minivan is generally susceptible to cavity boom noise. The amplitude of the boom noise is particularly affected by the vibration characteristics of the 3rd row seat tub because of its open-top (cargo storage) design. In addition, first longitudinal cavity mode is sensitive to this boundary excitation. The stiffness of the seat tubs are designed such that its modal alignment with respect to the first few cavity modes is kept well separated.

As mentioned earlier, seat tubs are supported at the boundaries, bolted on to the body with a butyl adhesive applied between the flange and the floor. This arrangement leaves large surface of the tub unsupported creating high modal density at mid-and-high frequencies. These modes readily respond to the structure-borne tire/road noise. Therefore damping treatments are important to help reduce structure-borne noise radiating from the seat tubs. On the other hand, the noise radiating from the rear tire and the exhaust tail-pipe are potential air-borne noise sources which can transmit through the 2nd and 3rd row seat tubs. A high-level of sound transmission loss (STL) at mid-high frequencies is needed to control the air borne tire/road noise. A significant amount of insertion loss is expected from a relatively thin carpet and underlayment system. In order to address all these challenges, the NVH characteristics of the seat tubs need to be analyzed in detail. This is accomplished by employing advanced modeling techniques and hardware tests both at the component level and also at the full vehicle level.

One of the most important design attributes is the material of the tub. In addition to weight, cost, manufacturing, static stiffness, durability, etc…, the choice of a proper material is very important in determining the damping, stiffness and mass-law properties of the tub. For the construction of the tubs, different types of materials and damping treatments were considered, such as regular steel, Metal-Polymer-Metal (MPM) and Sheet Molded Compound (SMC). Since damping loss in regular steel is very low, additional damping treatments are needed to dissipate vibration and also to attenuate noise transmission. The potential damping treatments that can be applied to the surface of the tubs are Liquid Applied Spray Damper (LASD) and Patch-on Constraint Layer (PCL). The stiffness enhancement measures such as geometric formations and beads were also included in the design. The NVH performance of the construction materials was studied by using CAE methods [1-5], and also by testing components and the full vehicle. For high-frequency air-borne noise, an SEA model was used to rank different design alternatives and materials, in the early stages.

At the component level, multiple bench tests were conducted on the tub. Modal analysis was performed on a tub to help layout the mode-management charts. Point mobility was measured to determine the frequency response of the tubs. T60 decay time was measured to determine the average damping loss factor of the tub. The sound transmission loss characteristic of the tub was evaluated by testing the component on a reverberation room window. In the full vehicle tests, both standard chassis-roll dynamometer and road tests were conducted. The details of these tests and some of the results will be presented in the following.

SEAT TUB COMPONENT DESCRIPTION

As summarized in Table 1, several seat tub configurations were evaluated. For the sake of simplicity, the color code of the graph legends in the following figures is listed separately in Table 2. All of the tubs that were tested had beads to enhance panel stiffness. The bead patterns and the depth were slightly different in the tubs made out of SMC material. The nominal panel thickness of the bare sheet metal tub was 1mm. The construction of the MPM tub included a thin layer of polymer sandwiched in-between two layers of steel sheet metal with equal thickness. The most significant properties of the tubs are listed in Table 1.

Tested Seat Tubs Description	Surface Area (m^2)	Weight (Lb)	Damping Material Coverage	Comments
Tub / Bare Sheet Metal	1.5	27	NA	Design in terms of pattern and depth of ribbings are the same, but thickness varies for these tubs.
Tub / MPM Construction	1.5	32	NA	
Tub / Bare Sheet Metal with LASD at Bottom	1.5	30	Full @ Bottom	
Tub / Bare Sheet Metal with LASD at Bottom and Sides	1.5	31	Full @ Bottom, 70% @ Sides	
Tub / Bare Sheet Metal with PCL at Bottom	1.5	32	70% @ Bottom	
Tub / Bare Sheet Metal with PCL at Bottom and Damping Butyl at Sides)	1.5	33	70% @ Bottom, 30% @ Sides	
Tub / SMC Construction	1.5	18	NA	Design is moderately different with other tubs in terms of patten and depth of ribbings

Table 1. The construction of the seat tubs.

MPM Construction
Sheet Metal
Sheet Metal / PCL @ Bottom
Sheet Metal / PCL @ Bottom / Butyl @ Sides
Sheet Metal / LASD @ Bottom
Sheet Metal / LASD @ Bottom and Sides
SMC Construction

Table 2. The legend used in the following graphs.

COMPONENT TESTS

During the component tests, both structural and acoustic performance characteristics of the tubs were evaluated. The component test set-up for structural measurements is shown in Figure 1.

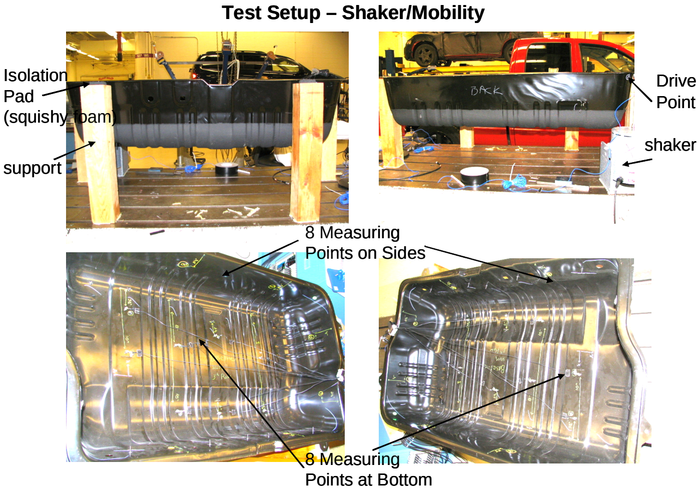

Figure 1. The test setup used during transfer mobility measurements.

The transfer mobility was measured to evaluate frequency response characteristics of the tubs. During these tests, seat tubs were mounted on wood poles with isolation pads to simulate quasi-free boundary condition. This mounting scheme enabled quick change of parts while maintaining a high test-repeatability. A random excitation force was applied by using a shaker, mounted at the corner of the tub. The vibration response was measured by using multiple accelerometers, installed both at the bottom and the sides of the seat tubs. During the analysis, space-averaged frequency response functions (FRF) were evaluated. Considering the frequency range of interest for structure-borne noise in the vehicle, FRFs were measured up to 1 kHz.

The mobility of the tubs, shown in Figures 2-5 (legends summarized in Table-2), demonstrate that the tub with the MPM construction has a lower mobility than the other constructions. Above 50 Hz, there is a clear separation between the MPM and the others, where the inherent high damping property of the MPM is observed to suppress the peaks and reduce the vibration levels. Note that, the tub made out of bare steel with no damping treatment has the highest mobility with sharp peaks. The mobility of the bare steel tub is significantly lowered if damping treatments (PCL or LASD) are applied. Considering the average mobility at the bottom of the tub, PCL has a slightly better performance than the LASD treatment. Additional damping materials applied to the sides of the tubs reduced the average vibration level. Note that, the tubs with add-on surface damping treatments ended up having similar weight as the MPM, but required more packaging space. The mobility of these tubs ranked somewhere in between the MPM and the bare sheet metal tubs. The resonances at low frequencies, observed in Figures 3 and 5, can be a potential concern in controlling of vehicle boom noise. The employment of stiffness enhancement measures are found to be very helpful in tuning the first few modal frequencies of the seat tubs at low frequencies. Considering that beads can be introduced with relatively negligible cost and weight penalty, their shape, depth and distribution are used as design parameters to optimize alignment of first few modal frequencies with respect to the interior cavity modes. During this process average effective stiffness of the tub was constantly checked to maintaining low radiation efficiency at higher frequencies. In other words, air-borne noise transmission characteristics of the tubs were also kept in check while stiffness enhancement measures are introduced. As discussed in detail later, increased stiffness has the adverse effect of lowering the coincidence frequency and degrading the Sound Transmission Loss (STL) characteristics of the tub.

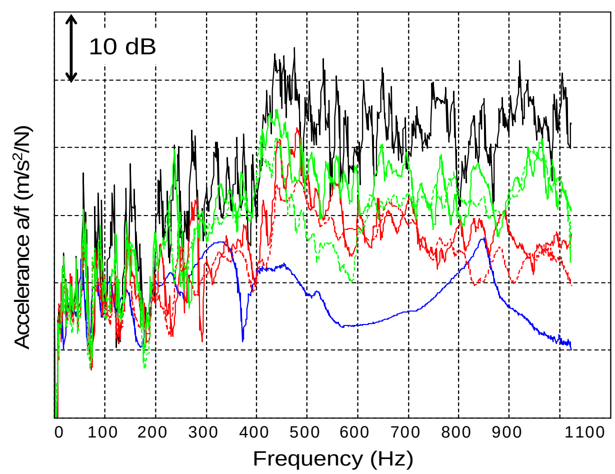

Figure 2. A comparison of space-averaged accelerance at the bottom of the tubs.

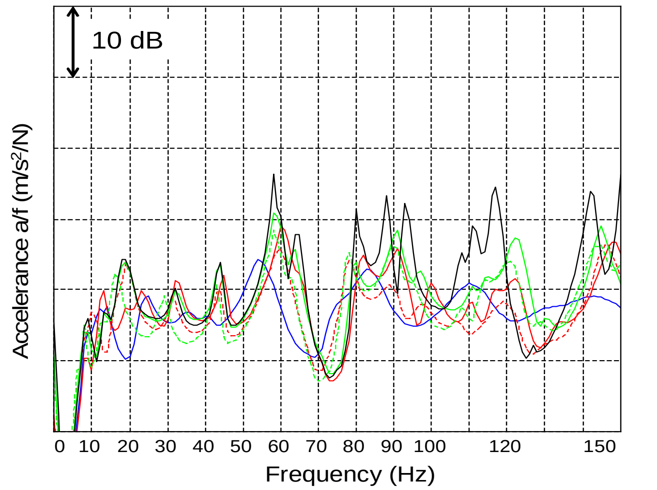

Figure 3. Average acceleration at the bottom of the tub, low frequency range 0 – 150 Hz.

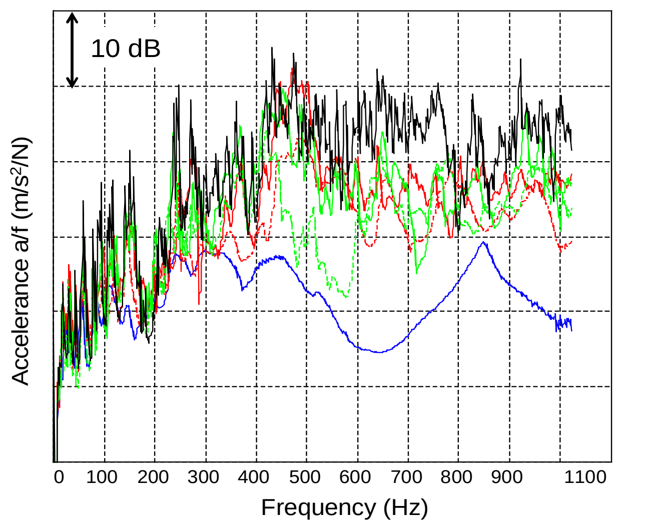

Figure 4. Average acceleration at the sides of the tub.

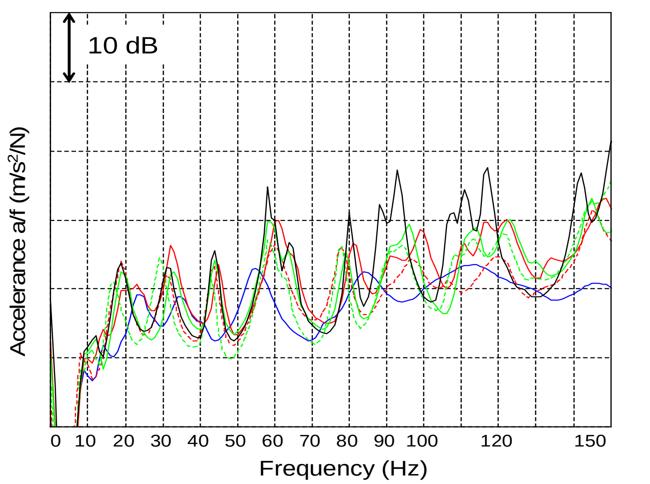

Figure 5. Average acceleration at the sides of the tub, low frequency range 0 – 150 Hz.

In a separate test, the mobility of the tub made out of SMC construction is compared to MPM construction. Note that, in order to gain stiffness, the beads in SMC tubs were slightly different than MPM tub in their pattern, depth and distribution. Therefore, during the tests, transducer locations were slightly different than all tubs that were evaluated earlier in Figures 2 – 5. As expected, due to built-in high damping, the average mobility at the bottom of the tub for MPM is lower than SMC tub throughout the frequency range. In Figures 6 and 7, separation between MPM and SMC is similar to MPM and PCL. In Figure 7, modes of the SMC tub shift to higher frequencies, but since damping and mass of SMC material is low, the mobility remains high.

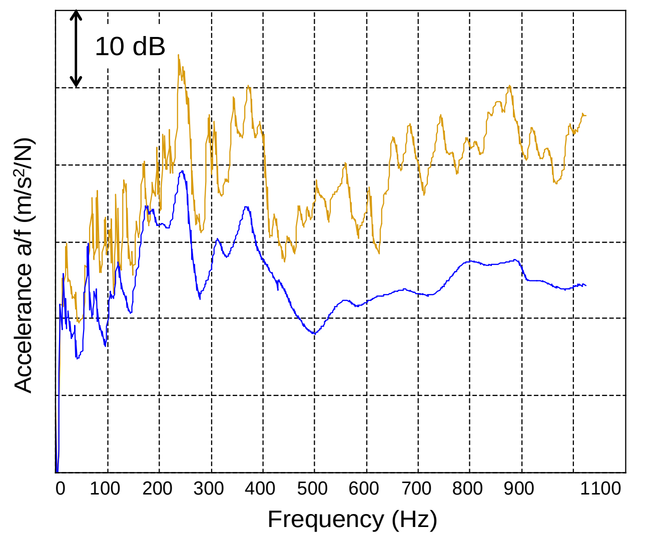

Figure 6. Average acceleration at the bottom of the tubs, SMC vs. MPM, frequency range 0–1 kHz.

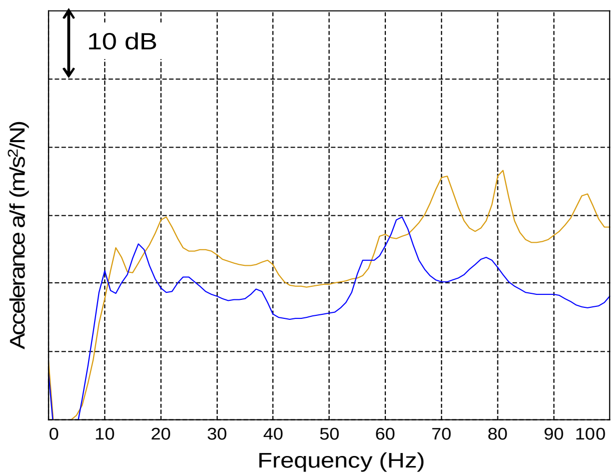

Figure 7. Average acceleration at the bottom of the tubs, SMC vs. MPM, low frequency range 0–1 kHz.

As stated earlier, damping is critical to attenuate structure-borne noise. Damping Loss Factor (DLF) for the tubs with various constructions and added damping materials was evaluated by employing T60 decay test procedure [2]. In order to gain better insight in the material behavior, DLF tests were carried out also on 10"x10" square plates except for SMC material. (Because of material availability, DLF test for SMC material were conducted on tub component). DLF measurements were conducted using an impact hammer and miniature accelerometer. Time trace of the impact points are used to extract damping loss factor applying Schroeder Integration and octave band filtering techniques. Space-averaged component damping loss factor is obtained by conducting decay measurements at multiple response points. The DLF test results are presented in Figure 8. Overall, DLF for MPM construction component is a lot higher than other components.

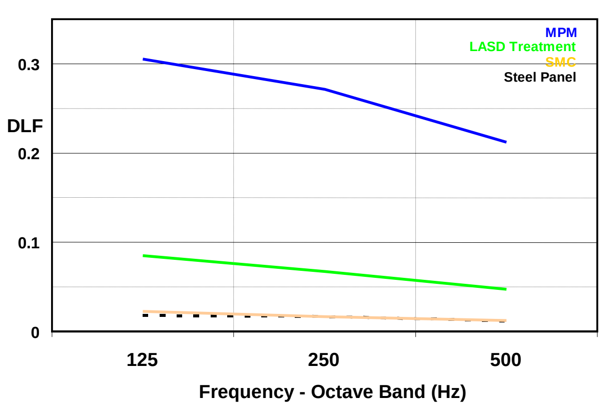

Figure 8. Damping loss factor of tub materials evaluated on 10x10" clamped panels.

Air-borne noise attenuation characteristics of the tubs were measured by conducting Sound Transmission Loss (STL) tests. The seat tubs were mounted on the window between a reverberation room (source side) and a semi-anechoic room (receiver side). The STL is evaluated by measuring the average SPL from a rotating boom microphone in the source room and the average SPL from multiple microphone locations in the receiver room, as shown in Figure 9. As observed from Figure 10, STL performance roughly follows the theoretical mass-law for the corresponding tub, with the exception of the bare steel tub with LASD or PCL treatments.

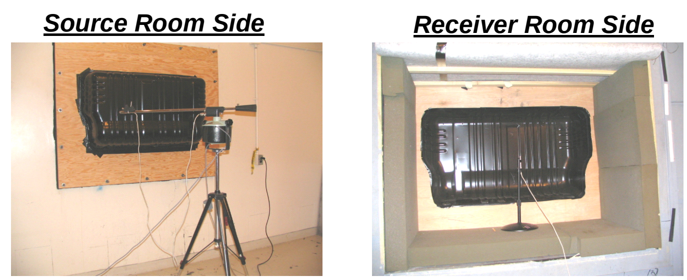

Figure 9, Noise Reduction (NR) measurement setup.

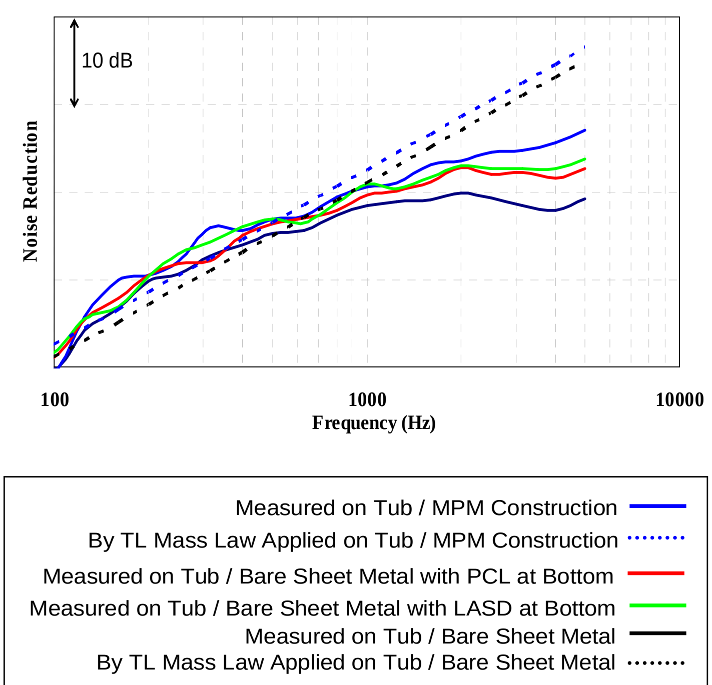

Figure 10. Noise reduction comparison for tubs with different constructions and damping treatments.

The theoretical mass law STL curve for MPM and steel panel (dashed blue and dashed black curves) match those of the tub with MPM construction and bare steel tub (solid blue and solid black curves) up until 1 kHz, respectively. Above 1 kHz because of the lower coincidence frequency of the tubs (~5 kHz), test results fall under the theoretical mass-law STL curve. Therefore the coincidence plateau is lower than the predicted mass-law STL. At this coincidence frequency, the bending wave length of the geometrically stiffened tub matches the wave length of the acoustic waves in the air. Although the tub with PCL treatment (at the bottom only) has more mass than the tub with LASD treatment (at the bottom only), the fact that the PCL tub has partial coverage at the bottom, the remaining area (70% of total) dominates the air-borne noise transmission through the tub. Hence, contrary to the theoretical mass law TL, PCL has lower STL value.

IN-VEHICLE TESTING

Three types of seat tubs were tested during full vehicle evaluations. All components were tested in the same vehicle. The tubs were installed and removed carefully during the iterations. As mentioned previously, the tubs were dropped in, and structurally supported at the perimeter, mainly secured by bolts and adhesives. Since there were no spot welds involved, this attachment method reduced the risk of introducing uncontrolled errors during the removal and installations of the tubs. In order to have minimum effect induced by tub changes in the vehicle, efforts are made to ensure quality mechanical work removing and replacing tub components. In between changes, torque forces were kept at the same level during the re-tightening of the bolts. Application process and cure time for the adhesives were strictly adhered to the application specifications. The tub constructions tested included 1) bare steel construction, 2) bare steel tub with PCL treatment (bottom only) and 3) with MPM construction. For each configuration, both the 2nd and 3rd row seat tubs were installed. The tests were conducted in a chassis-roll dynamometer cell and also on an out-door test track.

1. Outdoor test track testing – For outdoor test track evaluation, the vehicle was cruising at a constant steady speed. SPL was measured at driver's seat and 3rd row seat center locations. In Figures 11 and 12, the SPL results are presented. It is observed that the response has two peaks at 40 and 210 Hz for both driver and 3rd row seat locations. This corresponds to the vehicle and tire cavity resonances, or cavity boom under this operating condition. There is not much SPL difference at the 40 Hz boom between the 3 tub configurations considered. But for 210 Hz boom, tub with MPM construction significantly perform better at the 3rd row seat and not so much at the driver seat. This is consistent with the component level mobility test at Figures 2-5 that the separation of mobility starts around 50 Hz. The 70 Hz SPL peak observed at 3rd seat is not observed at driver seat, thus this peak is not due to cavity resonance. Instead this peak is due to local structural vibration from 3rd row seat tub. At 70 Hz peak in Figure 11, curves with MPM construction and PCL treatment tubs have lower SPL than tub with bare sheet metal construction. This again is consistent with component level tests. Obviously MPM has larger benefits due to lower mobility property. For both 3rd row and driver seat SPL, tub with MPM construction has better performance than the other two types.

2. Dynamometer Cell test – For evaluations in dynamometer cell, results based on the coast-down conditions are shown in Figures 13 and 14. SPL was measured again at the driver and 3rd row seat center locations. The cavity resonance at 210 Hz can be observed from both these graphs. Similar to road tests, performance separation can only be observed from the 3rd row seat, with tub with MPM construction being the best configuration. And tub with MPM material consistently perform better at higher frequency range (> 500 Hz).

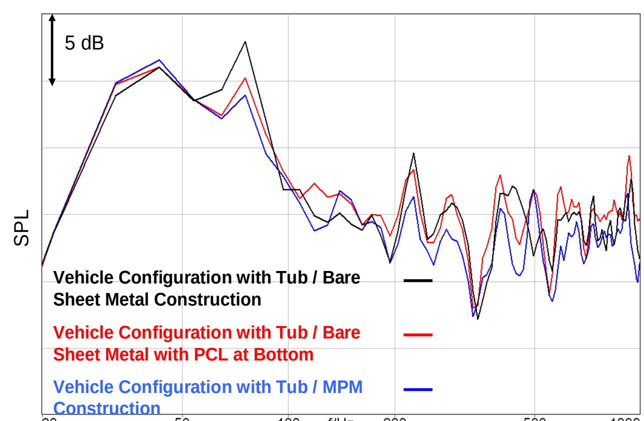

Figure 11. SPL at 3rd row seat with a constant steady speed under outdoor test track testing.

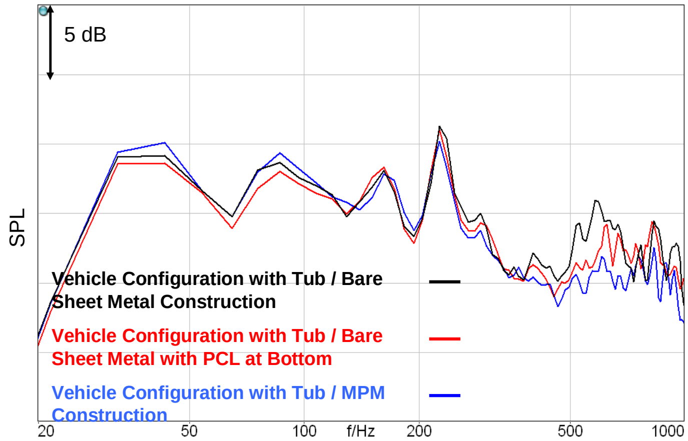

Figure 12. SPL at Driver seat with a constant steady speed, under outdoor test track testing.

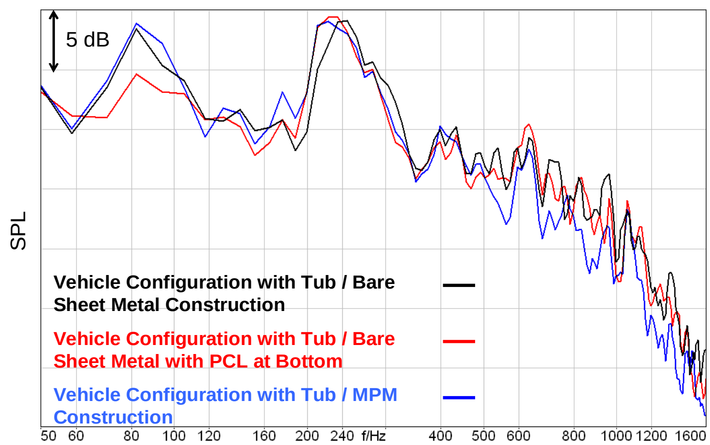

Figure 13. SPL at Driver seat, coast down condition, with rear tire on dyno roll.

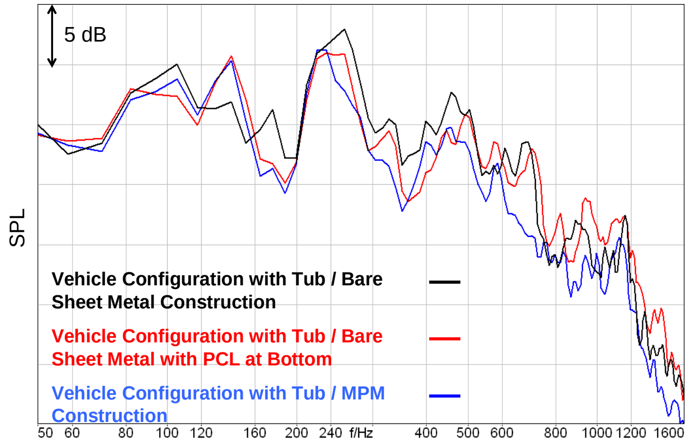

Figure 14. SPL at 3rd row seat, coast down condition, with rear tire on dyno roll.

USE OF SEA MODEL

In an earlier study, the details of constructing a Statistical Energy Analysis (SEA) model and its validation were presented [4, 5]. In order to avoid repetition, only the results of model predictions for air-borne noise transmission and effect of design iterations on the tub are presented here. In the SEA model, both experimental and empirical data were used for the STL of the plastic tub component to simulate airborne characteristics of the tubs. For some tub materials, models were used to predict their STL since early in the design process no hardware was available for all material options. In the model, tire-road noise is used as the source and the response is assessed at the driver's and 2nd row passenger's ear locations. The SEA model is used to predict noise reduction levels for tire/road noise for various tub materials, as shown in Figure 16. The benefit of the SEA model was experienced mostly during the early design process to sort out all the design alternatives and materials to guide engineering team on most valuable choices. The SEA model is kept relatively simple following the main spirit of the theory. This minivan vehicle with large interior acoustic cavity, large tub surfaces and well-defined air-borne noise sources made the SEA approach very applicable, and the model predictions matched the hardware tests results very closely. In addition to tub construction materials, various other design alternatives were studied by using the SEA model, such as the effect of tub covers, carpet construction (mass-back vs. absorptive light-weight), the effects of stiffness enhancements (beads) and damping treatments, etc.

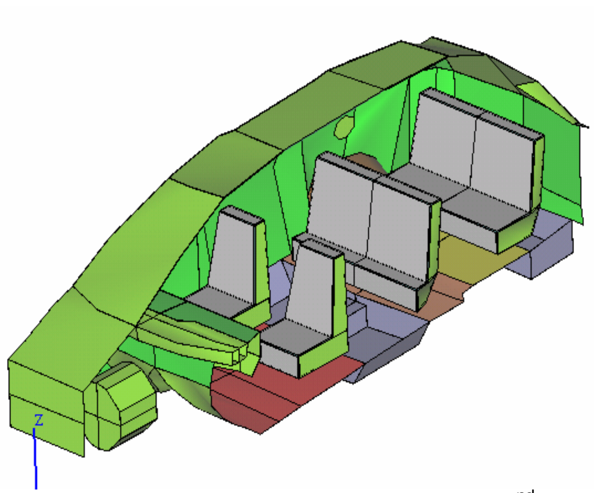

Figure 15: SEA model of the minivan with 2nd and 3rd row tubs.

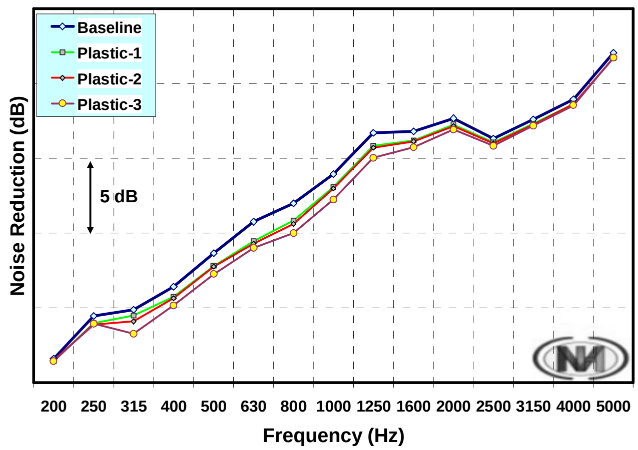

Figure 16: SEA model prediction on tire/road noise reduction to 2nd row head ear space with various tub material options.

CONCLUSION

This study has demonstrated that even in a very tight packaging space, the use of special NVH materials such as MPM, together with geometric enhancements can lead to successful NVH performance. The key point in this study is the engineering process which includes understanding the full set of design constraints, the available tools and the process in which the alternatives are evaluated to find the best optimal solution. The study also demonstrates how the hardware evaluations of component and full vehicle, and modeling can help the decision making process. The NVH performance of the tub gave consistent results between component and full vehicle testing and CAE modeling. For example, although significant separation above 50 Hz between the MPM tub and other tubs was observed in the component level evaluations, full vehicle tests did not show the same degree of separation. This is due to multiple path noise transmission phenomena observed in the vehicle, where the tubs make up only two of these multiple paths. On the other hand, from the road tests both on test track and dynamometer, it was concluded that the MPM was a better choice for NVH for this special tub application, and given all the design constraints. One of the well-known physical properties of an MPM construction is the low bending stiffness, due to splitting the thickness into two layers. In general, this feature may become an issue if modal alignments are not managed properly. The effect of low bending stiffness was evident in the vehicle road tests conducted at constant speed operating conditions. From the test results, a lower cavity resonance at around 40 Hz and a higher resonance at around 210 Hz are observed. However this has not been an issue for this particular application in this vehicle.

There are two major features of SMC construction that creates extra challenges. First, the low mass density which is desirable for weight savings becomes a disadvantage in the mass-law part of STL. Second factor is the low damping loss factor of SMC construction, which affects low frequency resonant response and also the STL near the coincidence frequency. Therefore additional surface damping treatments are needed. On the other hand, the major advantage of SMC is the stiffness which can be tuned by changing molded bead configuration and depth.

In summary, the major constraint in this application has been the extremely tight design space. Otherwise, the conclusions derived in this study can not be generalized to other applications where the alternative constructions may prove more cost effective, such as thicker carpet pile which will increase insertion loss and localized damping treatments using robotically applied LASD which may lead to lower cost. Instead of “One solution fits all” approach, this study emphasizes the value of the engineering process, where understanding of the underlying physical behaviors across the full spectrum, deployment of all available test and analysis tools, detailed layout of all design constraints, and consideration of all materials and constructions leads into successful evaluation, ranking and optimization of the design alternatives.

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