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ABSTRACT

This paper documents a joint development process between General Motors and Dow Automotive to improve primary body structure frequencies on the GM family of midsize vans by utilizing cavity-filling structural foam. Optimum foam locations, foam quantity, and foam density within the body structure were determined by employing both math-based modeling and vehicle hardware testing techniques.

Finite element analysis (FEA) simulations of the Body-In-White (BIW) and "trimmed body" were used to predict the global body structure modes and associated resonant frequencies with and without structural foam. The objective of the FEA activity was to quantify frequency improvements to the primary body structure modes of matchboxing, bending, and torsion when using structural foam.

Comprehensive hardware testing on the vehicle was also executed to validate the frequency improvements observed in the FEA results. BIW modal tests were performed before and after the addition of structural foam to confirm the FEA predictions. A production vehicle was also foamed and modal tested to verify that improvements to the BIW were also comprehended at the full vehicle level. Hardware measurements from road response testing were also incorporated into the evaluation matrix. Acceleration measurements (Power Spectral Density or PSD) at the steering wheel and seat track, and sound pressure level (SPL) measurements in the passenger compartment were collected to further evaluate the affect of structural foam on overall vehicle performance.

BIW test results of a current production van yielded a first structural mode frequency increase of 11% using 8pcf (pounds per cubic foot) density foam and 35% using 24pcf density foam. Additional advantages to foaming the vehicle were also observed in terms of improvements to interior sealing and reductions in low frequency interior noise levels.

INTRODUCTION

Reducing the weight of automobiles is a key challenge facing design engineers in the 21st century. Minimizing vehicle mass is essential in meeting increasingly stringent fuel economy and emission requirements. Engineers must discover new methods and technologies to reduce vehicle weight without compromising other engineering design requirements such as structural integrity, safety, ride and handling, and acoustic performance.

A very effective technology to help improve structural performance (locally or globally) and minimize vehicle mass is BETAFOAM® structural foam, a family of two-component polyurethane products manufactured by Dow Automotive. BETAFOAM structural foams are available in a variety of densities and strength properties that can be integrated into a vehicle design to help optimize body structure performance relative to crashworthiness, structural stiffness, and body sealing.

BACKGROUND

The automotive engineering community has dedicated a tremendous amount of resources in the pursuit of enhanced body structure performance. Superior body structure integrity is typically associated with vehicles having high primary structural frequencies.

There are two methods conventionally used to improve structure stiffness performance. The first method is part thickness changes and/or section size optimization. This method is typically used in the early stages of new vehicle product development. The second (and less mass-efficient) method is to add reinforcements and/or doubler plates. This method is used in the late stages of product development or in carryover vehicles that are "locked-in" to particular vehicle architectures due to timing and budgetary constraints. However, new technologies are available that provide mass- and cost-efficient ways to improve body structure performance. An attractive alternative for enhancing structural performance in both new and existing body architectures is by utilizing BETAFOAM® cavity-filling structural foam.

BETAFOAM acoustic and structural foam products are not unfamiliar to the automotive industry, as they have been used since 1995 on several competitive production vehicles. However, recent technological product innovations such as low isocyanate chemistries and fast-reacting foam component materials have made BETAFOAM technologies more attractive from a manufacturing and processing standpoint. Utilizing foam-injection technologies has many potential advantages in terms of enabling section size reductions (improved vehicle styling) and gage/part reductions (minimized vehicle weight) that otherwise would have been very difficult to execute. In addition, current model year structural improvements can be achieved without any modification to the weld sequencing and tooling layouts in the body shop that would be necessary if steel reinforcements were added.

The Pontiac Montana, a member of the GM family of mid-sized vans built off of the same vehicle architecture (Chevrolet Venture, Pontiac Montana, Oldsmobile Silhouette), was chosen for this foam evaluation for several reasons. First of all, FEA models that were previously correlated to hardware test data were readily available. Production body frames, otherwise scheduled for "scrap-out", could be pulled from the plant for little cost. Most importantly, similar to most vans and open architecture-type vehicles, the Pontiac Montana has a low body structure matchboxing mode that is ideally suited for structural enhancement via structural foam.

FEA full body plate models were used to determine optimum foam type (density), placement (location within the structure), and quantity (mass) of structural foam to be injected into the vehicle cavities. The structural mass efficiency of the foam added to the vehicles was measured in terms of foam mass per unit of frequency improvement (kg/Hz). Foam placement in each joint was optimized by evaluating strain energy density plots for each global structure mode shape.

Two different BIW structures, one injected with 8pcf foam and the other with 24pcf foam, were injected in the D-pillar upper and D-pillar lower joints. These vehicles were then modal tested to validate the FEA predictions. In addition, a production vehicle was injected with 24pcf foam in the D-ring (D-pillar upper and lower joints) and then evaluated on the GM Milford Proving Grounds (MPG) test track for rough road shake and noise response.

MATH-BASED SIMULATIONS

MODEL OVERVIEW – The FEA portion of this project was performed on BIW and "trimmed body" models of the Pontiac Montana body (see Figure 1). The "trimmed body" model contains the entire vehicle content except the vibration-isolated components of the vehicle (powertrain, suspension, exhaust) and the door closures.

Conventional two-dimensional plate elements were used to model the BIW sheet metal and glass closure panels. Lumped mass elements were used to represent non-structural vehicle weight (seats, I/P, trim panels, etc). Rigid elements and springs were used respectively to model welds and adhesives. The model used was comprised of approximately 50,000 model elements including plates, springs, and lumped masses. The model used in this analysis also did not include the engine compartment diagonal braces from the motor compartment upper rail to the upper tie bar, which are used to stabilize the front end for twist and lateral deformations.

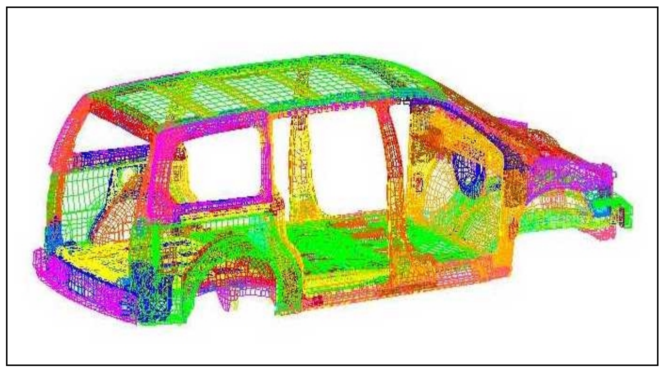

Figure 1: Baseline FEA Body Model

A NASTRAN normal modes analysis was run on the baseline BIW and trimmed body model in order to benchmark the baseline frequency performance of the unfoamed structure. The baseline performance is summarized in Table 1.

Table 1: Baseline FEA Trimmed Body Performance

Mode	BIW	Trimmed
Matchboxing	20.39 Hz	18.11 Hz
Fr end lateral / torsion	28.84 Hz	19.06 Hz
Bending	32.86 Hz	24.51 Hz

FOAM MODELING – Cavity-filling materials were then modeled within the body structure in order to quantify structural improvements when utilizing structural foam.

Three-dimensional solid elements were used in HYPERMESH and NASTRAN to model foam in eighteen body side joint locations. Three different grades of foam (8, 16, and 24pcf densities) were evaluated in order to further comprehend foam performance versus mass tradeoffs. Each foam density has different engineering properties, and foam modulus/stiffness increases with foam density. The material properties of the foam were determined from Oberst bar testing of foam samples and were used as inputs for foam material properties in the FEA models.

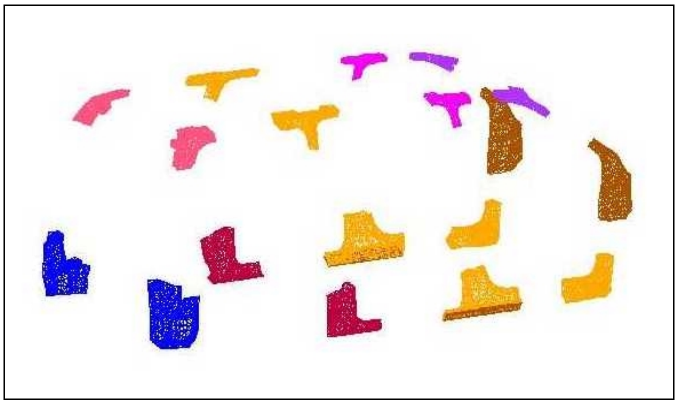

Figure 2: Foam Locations

A schematic of the optimized foam placement in the vehicle is shown in Figure 2 above. There are substantial volume and associated foam mass differences between joints due to the different sizes of body cavities being filled. Foam mass for each joint location considered in the FEA model is summarized in Table 2 below.

Table 2: Foam Mass Distribution for Various Foam Densities

Foam Location	8 pcf	16 pcf	24 pcf
Plenum	0.70 kg	1.40 kg	2.10 kg
A-pillar lower	0.60	1.20	1.80
A-pillar upper	0.20	0.40	0.60
B-pillar lower	0.80	1.60	2.40
B-pillar upper	0.15	0.30	0.45
C-pillar lower	0.60	1.20	1.80
C-pillar upper	0.40	0.80	1.20
D-pillar lower	1.35	2.70	4.05
D-pillar upper	0.35	0.70	1.05
Total mass added	10.3 kg	20.6 kg	30.9 kg

ANALYSIS RESULTS

BIW – Frequency improvements for the BIW structure are summarized in Figure 3 for each foam density. The affect of foam on the matchboxing frequency is nearly twice the improvement displayed by the other modes tracked. Bending benefits as well to a lesser degree, whereas torsion is affected minimally because the mode shape is dominated by front-end lateral motion. The decreasing slope of each curve as foam density increases demonstrates the reduction in foam mass-efficiency with increasing foam density (and BIW mass).

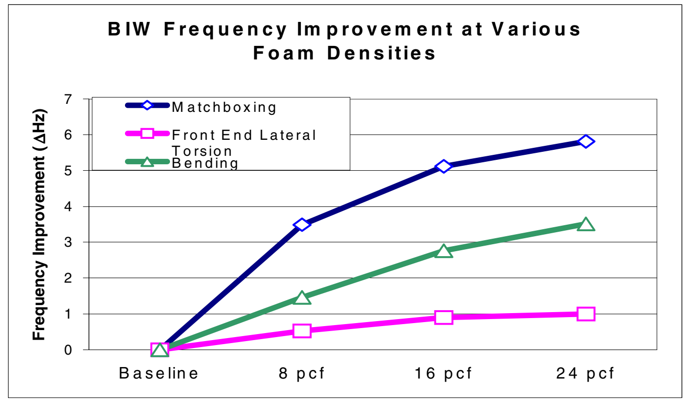

Figure 3: FEA BIW Estimated Frequency Improvements

TRIMMED BODY – Frequency improvements to the trimmed body are lower than for the BIW due to the nonstructural nature of the added interior component masses. The results in Figure 4 verify this phenomenon. The frequency improvements provided by the structural foam at the trimmed body level are very significant but are lower than the improvements at the BIW level. Again, the torsion mode is affected the least by the foam due to the front end-dominated nature of the mode shape.

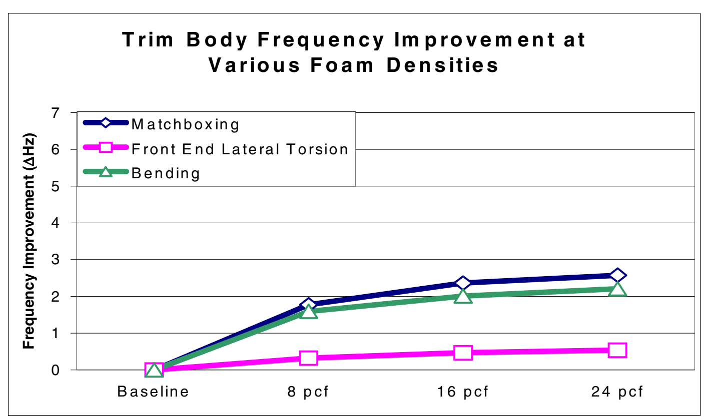

Figure 4: FEA Trimmed Body FEA Frequency Improvements

MASS EFFICIENCY – Although the higher density foam materials (16pcf and 24pcf foam) yield larger frequency improvements, there is a larger associated mass penalty. Table 3 summarizes the mass efficiency of the foam by dividing the added foam mass by the frequency improvement for each mode shape and foam density. This is an effective method to quantify the relative benefits provided by the different densities of foam. The 8pcf foam provides the optimal benefit when considered in terms of foam mass efficiency.

Table 3: BIW Analysis Mass Efficiency Prediction

	Mass efficiency (kg per Hz)		
Mode	**8 pcf**	**16 pcf**	**24 pcf**
Matchboxing	5.8	8.8	12.0
Fr end lateral / Torsion	32.3	44.0	58.5
Bending	6.5	10.3	14.0

D-RING BIW ANALYSIS – Matchboxing frequency improvements were the primary objective of the project and it is well understood that the D-ring heavily influences this mode. Engineering judgement and process limitations drove us to utilize foam were it would provide the most benefit. These factors contributed to our decision to analyze the test hardware and full vehicle only with foam in the D-pillar upper and lower joints.

Additional FEA iterations were performed on the BIW model in which only the D-ring joints were filled with 8pcf and 24pcf foam. Again, these joints were specifically selected to primarily improve the matchboxing mode. As a result, only four locations were filled with foam rather filling eighteen locations when attempting to improve all of the global modes (like what was done in the previous FEA iterations). The D-ring analysis results, summarized in Table 4, indicate a larger improvement in matchboxing for the 24pcf (versus the 8pcf foam) but with lower foam mass-efficiency.

Table 4: BIW FEA Matchboxing Frequency Improvement Predictions, D-Ring Foam Only

Foam Density	Foam Mass	Freq. Impr.	Foam Mass Efficiency
8pcf foam	3.0 kg	+3.0 Hz	+1.0 kg/Hz
24pcf foam	9.0 kg	+5.1 Hz	+1.8 kg/Hz

HARDWARE EVALUATION

BIW MODAL TESTING – Two BIW structures were each injected with 8pcf and 24pcf foam respectively in the D-ring (D-pillar upper and lower joints). The BIW structures were then modal tested to verify the predicted FEA improvements and foam mass efficiencies.

Results for the primary structural modes were tracked for both the 8pcf and 24pcf foam configurations and are summarized in Table 5 and Table 6. As expected, the matchboxing mode displayed the most significant frequency improvement of 2.4 Hz (11%) with 8pcf foam and 6.8 Hz (35%) with 24pcf foam. Torsion, the most closely coupled mode shape with matchboxing, improved to a lesser degree, 0.6 Hz (2%) and 3.5 Hz (12%) respectively. The last primary mode shape, bending (classic 2nd order two-node bending), was essentially unaffected by the addition of foam in the D-ring.

Table 5: BIW Modal Test Summary with 8 pcf Foam

Mode	Base (Hz)	Foam (Hz)	Freq. Impr.	Mass Eff. (kg/Hz)
Matchboxing	20.5	22.9	+2.4 Hz	+0.76
Torsion	30.3	30.9	+0.6 Hz	+4.4
Bending	31.1	31.2	+0.1 Hz	18.3

Table 6: BIW Modal Test Summary with 24 pcf Foam

Mode	Base (Hz)	Foam (Hz)	Freq. Impr.	Mass Eff. (kg/Hz)
Matchboxing	19.6	26.4	+6.8 Hz	+1.2
Torsion	30.1	33.6	+3.5 Hz	+2.3
Bending	32.4	32.6	+0.2 Hz	+39.5

From a correlation standpoint, the FEA results correlated only marginally with the test data. Mass efficiencies of the foam obtained from the test results were less than those predicted in the FEA simulations (1.0 kg/Hz vs. 0.76 kg/Hz with the 8pcf foam, and 2.4 kg/Hz vs. 1.2 kg/Hz with the 24pcf foam). There are a several factors for these differences. Test hardware content was different from FEA model content (such as design updates in D-ring structure). Foam was not injected in the optimal locations within the joints (unlike what was represented the FEA model) because baffles blocked access to some of the critical body cavities.

Other correlation data on foamed vehicles (in which hardware foam content more closely replicates FEA model content) shows very strong correlation between FEA and test results. Despite the noted discrepancies, both the test and FEA results validate foam performance benefits and demonstrate key trends in foam mass-efficiency. Foam, regardless of density, provides a very mass-efficient way to improve performance. Mass efficiency of steel solutions is often significantly smaller than that of BETAFOAM structural foam.

FULL VEHICLE MODAL TESTING – In addition to the BIW modal analyses, a full vehicle modal test was also conducted. The high modal density of a full vehicle test and the corresponding difficulty in extracting body modes influenced our decision concerning which foam material to be used in the test. The 24pcf foam was chosen for evaluation based on the BIW frequency improvements observed and assuming that only a portion of the frequency improvement to the BIW would translate to the full vehicle level. This test scenario was not evaluated at the FEA level because a vehicle level system model was not available. The full vehicle scenario was tested in order to comprehend the flow down in frequency improvements from the BIW to the level that the customer experiences - the vehicle level.

Results for the vehicle primary structure modes are summarized in Table 7. A matchboxing frequency improvement of 3.8 Hz (22%) is less than the BIW test results. Again, this is attributed to the contribution of non-structural mass to full vehicle frequency performance. A torsion frequency improvement of 5.6 Hz (26%) was much larger than that at the BIW level and was an unexpected surprise. This additional increase could be attributed to many factors ranging from vehicle hardware interaction, modal density and mode coupling, and foam location variation. No bending frequency improvement was expected and the 0.6 Hz improvement observed is within the testing margin of error and warrants no further explanation.

The vehicle level frequency improvements observed by utilizing foam are sizable and very difficult to reach in a mass-efficient manner with conventional spot welded sheet metal joint construction methods. The observed frequency improvements are substantial, considering the mass of a full vehicle and the high stiffness increase required for a frequency improvement of just one Hertz.

Table 7: Full Vehicle Modal Test Summary W/24 pcf Foam

Mode	Base (Hz)	Foam (Hz)	Freq. Impr.	Mass Eff. (kg/Hz)
Matchboxing	17.5	21.3	3.8	2.1
Torsion	21.6	27.2	5.6	1.2
Bending	22.7	23.3	0.6	13.3

ON-ROAD EVALUATIONS – Any potential vehicle enhancement must be evaluated holistically to ensure no negative synergies occur with other aspects of vehicle performance. On-road evaluations are typically undertaken to determine if any of these adverse conditions exist. Physical evaluations of a vehicle on a controlled road surface typically occur in both subjective and quantitative steps. The subjective evaluation is performed by an experienced development engineer capable of detecting small variations in vehicle performance. This type of evaluation is quick and reliable although it does not involve any quantitative measurements. Quantitative evaluations may include numerous measurements including, but not limited to, rough road shake and interior noise measurements. Although both methods are reliable evaluation tools, the subjective approach is more customer-focused and introduces the human element to the process (which often carries more weight in the design process).

The subjective evaluation of the vehicle filled with 24pcf FOAM was very favorable. However, as with all vehicle performance-enhancing technologies, supportive quantitative measurements are an added benefit. For the purpose of this study, measurements were kept basic by evaluating only rough road shake and coarse/smooth road noise**.** Figures 5&6 (A-C) display a portion of the shake measurement data taken at the rear outboard seat-track point and noise measurement at three standard seated positions for the foamed and unfoamed vehicle configurations.

Shake Measurements – Power Spectral Density (PSD) measurements were taken on spalled concrete at several standard driver interface points and rear seated positions. Data collection points included driver's front inboard seat track (DFIST), steering column (Col), and second row (SRST) and third row (ROST) seat tracks. Review of the measurement data indicates an increasing level of influence from the structural foam as the test points approach the D-Ring. The rear seat track measurements highlight this difference very well in the primary body frequency range (15-30 Hz).

To report all the seat track data taken would occupy many pages, consequently only the ROST measurements are shown in this paper. The ROST, which is the closest location to the foamed D-ring, does indicate a change in the measured response. Shown in Figure 5(A-C) are the PSD measurements for the ROST in the fore-aft, cross-car, and vertical directions.

The fore-aft direction is essentially unaffected. The lateral direction indicates the de-coupling of the suspension vertical hop and tramp modes from the matchboxing and torsion body modes. This is evident by the response peak shift to slightly lower frequency and higher energy. This occurs at a low enough vibration level, however, so as not to warrant concern. The vertical direction aligns with the road input and is the direction of most of the vibrational energy, as witnessed by the levels in Figure 5C, which exceed –20dB. The vertical response measurements show significant improvement, with measured response levels reduced by 4-5 dB in the suspension mode frequency range and 3-4 dB in the primary body frequency range. Although the vertical measurements do indicate increased response at higher frequencies (40 –100), the vibration level is low enough so as not to warrant concern.

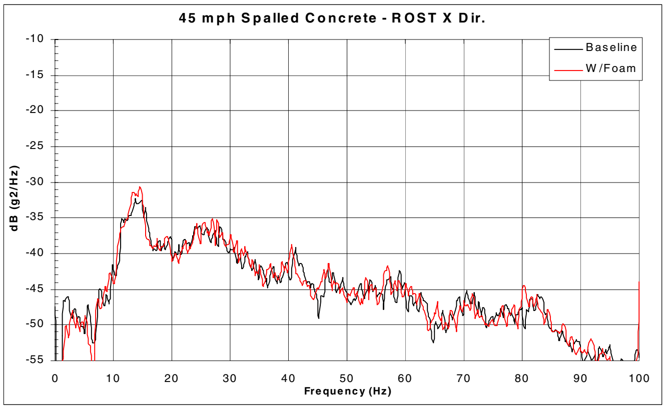

Figure 5A: Rear Outboard Seat Track PSD Fore/Aft

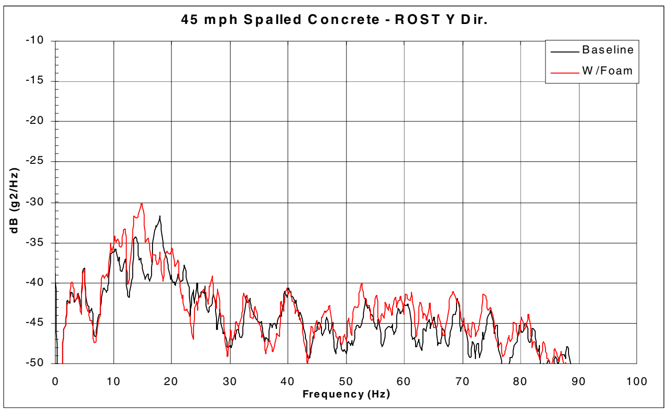

Figure 5B: Rear Outboard Seat Track PSD Lateral

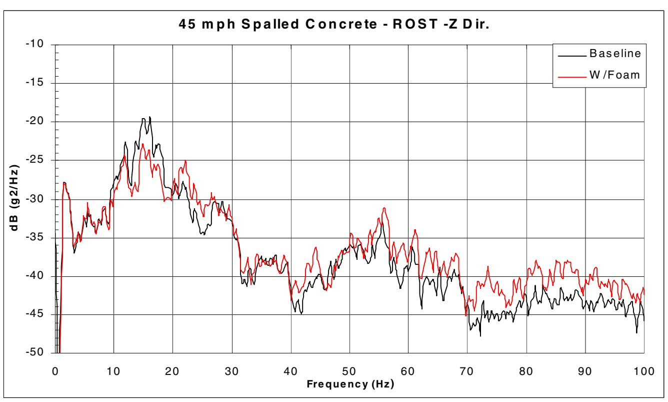

Figure 5C: Rear Outboard Seat Track PSD Vertical

<u>Noise Measurements –</u> Microphone measurements (SPL) were taken on smooth and coarse roads at standard seated positions in each row of seats. Measurement locations were driver's right ear, middle row driver's side, and third row center. Review of the data indicates a consistent trend in sound pressure levels for each seated position and road surface. Again, to report all the data taken would occupy many pages of data. As a result, measurements from only one road surface (smooth asphalt) at all three seated positions are shown in Figure 6(A-C).

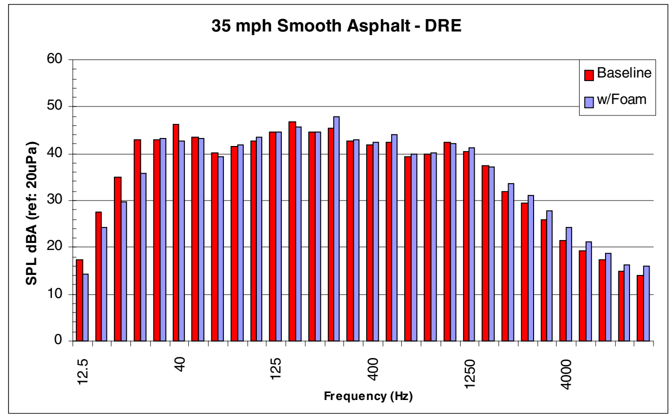

Figure 6A: SPL Noise Measurements at 35MPH on Smooth Asphalt (Drivers right ear (DRE))

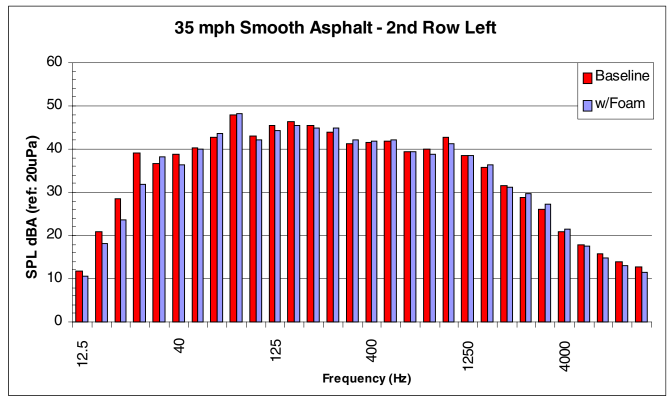

Figure 6B: SPL Noise Measurements at 35MPH on Smooth Asphalt (Second row left)

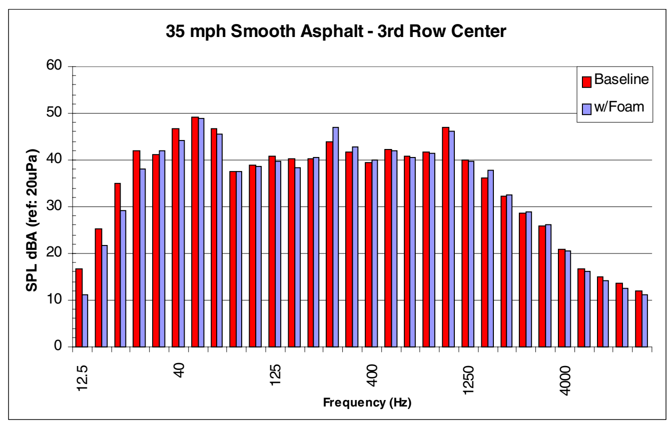

Figure 6C: SPL Noise Measurements at 35MPH on Smooth Asphalt (Third row center)

Figure 6(A-C) summarizes the SPL measurements in 1/3 octave from 0-10kHz for the three locations. Conclusions can be made by considering the frequency bandwidth to be divided into three ranges – low, middle, and high frequency. The low frequency range (less than 100Hz) is most-benefited by lower SPL achieved by utilizing structural foam. This frequency range is often where "boom" and rumbling-type annoyances occur. The middle frequency range (between 100 and 500Hz) is essentially unaffected by the presence of foam in the D-ring. The high frequency range (greater than 500Hz) displays a slight degradation in sound pressure level. A "small" compromise in the higher frequency range may be worth the substantial benefit realized at lower frequency levels. If necessary, high frequency issues could be addressed by body panel or interior trim panel modifications via acoustic material treatments.

The reduction in structure-borne (low frequency range) noise was due to the improvement in body stiffness provided by the structural foam in the D-pillar. Airborne noise levels (middle and high frequency range) were not improved because foam was not added in the areas that are part of the primary noise paths into the vehicle.

CONCLUSION

Very mass-efficient structural frequency improvements are achievable using BETAFOAM® structural foam compared to traditional methods for improving global structure such as adding reinforcements, increasing gage thickness, and enlarging section sizes. Although all three densities of foam evaluated provided significant levels of frequency improvements, 8pcf foam provides the most efficiency from a mass and performance standpoint.

Improved global structure frequency performance was observed in the FEA simulations and modal testing for both the BIW and full vehicle configurations. The foamed BIW had a first mode frequency increase of 30% using 8pcf foam and 42% using 24pcf foam compared to the unfoamed configuration. FEA correlation with test results was marginal due to inherent differences between hardware configurations and finite element model content.

Comprehensive hardware testing also verified improvements to vehicle performance relative to interior acoustics when utilizing structural foam in the D-ring. All seat positions on several different road evaluation tests exhibited reductions in low frequency noise levels. Subjective ride evaluation of the foamed vehicle was very encouraging as the vehicle exhibited a significant improvement in structural "feel".

Although the results of this project were very informative and insightful, additional foam development activities outside the scope of this project are recommended. First, additional test vehicles should be foamed in more joints than just the D-pillar to generate additional modal test data and noise level measurements for additional statistical comparisons. Second, further iterations on FEA models should be undertaken to correlate to test data and to further optimize foam placement with the structure. Finally, further road testing of foamed vehicles should be performed to further validate foam durability performance over the life cycle of a vehicle.

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