Optimization of Damping Treatments for Structure Borne Noise Reduction

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In the automotive industry, all passenger vehicles are treated with damping materials to reduce structure borne noise. The effectiveness of damping treatments depends upon design parameters such as materials, locations and size of the treatment. This article proposes a CAE (Computer Aided Engineering) methodology based on finite element analysis to optimize damping treatments. The developed method uses modal strain-energy information of bare structural panels to identify flexible regions, which in turn facilitates optimization of damping treatments with respect to location and size. The efficacy of the method is demonstrated by optimizing damping treatment for a full-size pickup truck. Moreover, simulated road noise performances of the truck with and without damping treatments are compared, which show the benefits of applying damping treatment. Finally, the optimized damping-treatment configuration is validated by comparing the locations and sizes of the treatment with that of a laser vibrometer test conducted on body-in-white (BIW) structures of the truck. The article also presents a brief discussion of a few other applications in which the proposed method has been applied successfully at DamilerChrysler Corporation in optimizing damping treatments.

Low noise inside the cabins of automobiles is an important product acceptance quality criteria. Therefore, NVH (Noise, Vibration and Harshness) engineers strive to achieve a good acoustic environment that provides total customer satisfaction. In the automotive industry, all vehicles are designed with an acoustic package that consists of various components such as absorbers, barriers, dampers and isolators to improve noise performance. In particular, surface damping materials are effectively used in reducing structure borne noise at frequencies beyond 100 Hz. At such high frequencies, higher order structural panel modes participate in the overall noise level inside the cabin, and hence surface damping treatments play a significant role in attenuating structural panel vibrations.

Typically, damping materials are applied on door, roof, dash, floor and cab back panels of automobiles. The commonly used damping-treatment forms are: (1) bake-on-mastic; (2) spray-on-mastic; (3) partially constrained layer; (4) composite constrained layer; and (5) metal-polymer-metal. In general, the surface damping treatments are categorized into two types. The first type is known as constrained damping treatment, and it has more than two layers. As shown in Figure 1, there is a base layer upon which a viscoelastic damping layer is applied, which in turn is constrained by a third layer. Such an arrangement provides extensional and shear damping. Partially Constrained Layer (PCL) is an example of constrained layer treatment. In the second type, known as unconstrained layer treatment, the viscoelastic damping treatment is applied on the base layer and provides vibration control of the base layer through extensional damping. Bake-on-mastic is an example of unconstrained damping treatment.[1]

Traditionally, experimental techniques are used in optimizing design parameters such as material type, size and location of damping treatments. In particular, laser vibrometer type tests are conducted on BIW structures or full-vehicle prototypes to optimize damping treatment locations and sizes. Specifically, vehicle structures are excited at each suspension or body-mount attachment with a known force input for a range of frequencies using a shaker. The resulting vibrations of various structural panels are mapped using a laser vibrometer, and corresponding velocity contours are generated. This procedure needs to be repeated for all attachment locations. The generated velocity contours are examined manually to develop an optimized damping treatment configuration. While the number of cab mount attachments can vary from 8 to 10 for a typical pickup truck or SUV (Sports Utility Vehicle), it can go up to 30 for cars and minivans depending on the type of suspension. Experimental approaches can become cumbersome, time consuming and prohibitively expensive because the structure needs to be excited for a large frequency range at all noise transfer paths to identify flexible regions on structural panels.

To overcome the drawbacks of experimental approaches, NVH engineers are challenged to find cost-effective analytical approaches that can be employed in early stages of the vehicle design process for optimizing damping treatments. This investigation does just that, proposing a CAE methodology based on finite element modeling and employing modal strain energy information of structural panels. It is also noteworthy to mention that there is no single analytical investigation reported in the open literature that helps to optimize the locations and size of damping treatment. Given this background, we have three objectives:

1. Develop a cost-effective analytical methodology to optimize surface damping treatment of structural panels that can be integrated with current simulation processes to improve noise performance of automotive vehicles. The emphasis is on optimizing the locations and sizes of damping treatments.
2. Demonstrate the applicability of the developed methodology by optimizing damping treatment of a vehicle.
3. Validate the methodology by comparing the damping treatment configuration obtained through simulation with that of an experimental approach.

Optimization of Damping Treatment

The CAE methodology for location and size optimization of damping treatment centers on modal elemental strain energy information of structural panels. It consists of the following steps (see Figure 2):

1. Construct a finite element model of the trim body. Identify body panels such as dash, roof, floor and cab back panels that need to be treated with damping material and request modal strain energy output for those panels only.
2. Perform modal analysis and extract modal strain energy up to a frequency of 400 Hz for the identified panels.
3. Calculate composite strain energy contributions of all the modes for the identified panels by adding elemental modal strain energy, and thereby generate composite strain energy distributions.
4. Plot the composite strain energy contours for the identified panels.
5. Select locations where resultant strain energy is high. Determine the size and thickness of damping treatments based on the composite strain energy contours and production feasibility.
6. Apply damping treatments for the selected areas.
7. Predict road noise and powertrain noise from the full vehicle NVH model.
8. Check if the predicted noise level has been optimized. If

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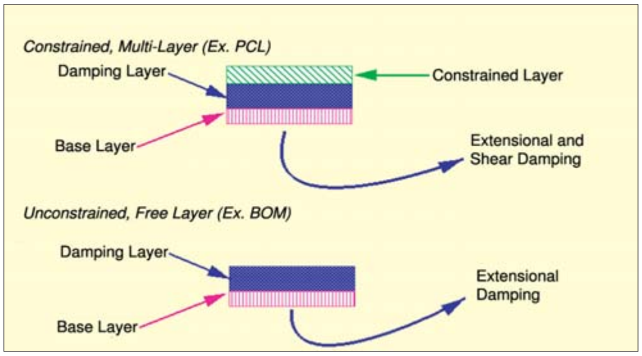

Figure 1. Types of damping treatments.

yes, the damping treatment configuration is optimum.
9. Otherwise, re-plot the composite strain energy contours with a different cut-off range, and extend the area of coverage or increase thicknesses of damping treatments.
10. Repeat Steps 6-9 until an optimum damping treatment configuration is achieved.

Damping Material Simulation. The stiffness and damping characteristics of damping materials exhibit a strong frequency and temperature dependency. Furthermore, these materials significantly alter the stiffness characteristics of the base structure. Therefore, it is important to capture the dynamic characteristics of damping treatments accurately in finite element models. Various, well documented approaches have been developed to simulate damping materials.[2] In this study, the RKU method is used to simulate damping treatment,[2] which, despite its simplicity, works well for both constrained and unconstrained damping treatment. In this method, equivalent properties of damping treatments with base layers are estimated and assigned at the element level. Thus, the effect of a damping treatment is included in terms of increased composite loss factor, increased stiffness and added non-structural mass.

Application

The developed methodology is applied to optimize damping treatment of a full-size pickup truck. In order to identify the locations and size of the treatment, elemental modal strain energy is calculated for the cab structure. The FE model of the cab structure includes details such as steering column, interior panels, seats and doors. Moreover, the cab structure is supported on the cab mounts and all the degrees of freedom are constrained at the frame end of the cab mounts. For this structure, normal modes are calculated up to a frequency of 400 Hz using SOL103 in MSC/NASTRAN.[3] Moreover, the eigenvectors are normalized to unity in the normal mode computations.[4] In this study, the modal strain energy information is requested for the floor and dash panels only. An in-house code was developed to sum the elemental modal strain energy across all the modes of the identified structural panels.

Numerical Results

Figures 3-8 present numerical results on strain energy contours, simulated road noise performances, and velocity contours from experiments. These results are presented in two segments. The first segment (Figures 3-5) shows results corresponding to the full-size pick-up truck. Specifically, Figure 3 shows the composite strain energy distribution for the identified panels. As shown, the panel locations identified in red have elements with high strain energy, whereas the locations identified in blue have elements with low strain energy. This contour plot of strain energy distribution readily indicates the locations and size of the damping treatment that needs to be applied to the panels. Such a damping treatment configuration is shown in Figure 3b. Figure 3c also shows a comparison between the damping treatment configuration identified by the developed methodology and the configuration identified through an experimental approach. A brief discussion of the experimental approach is in order.

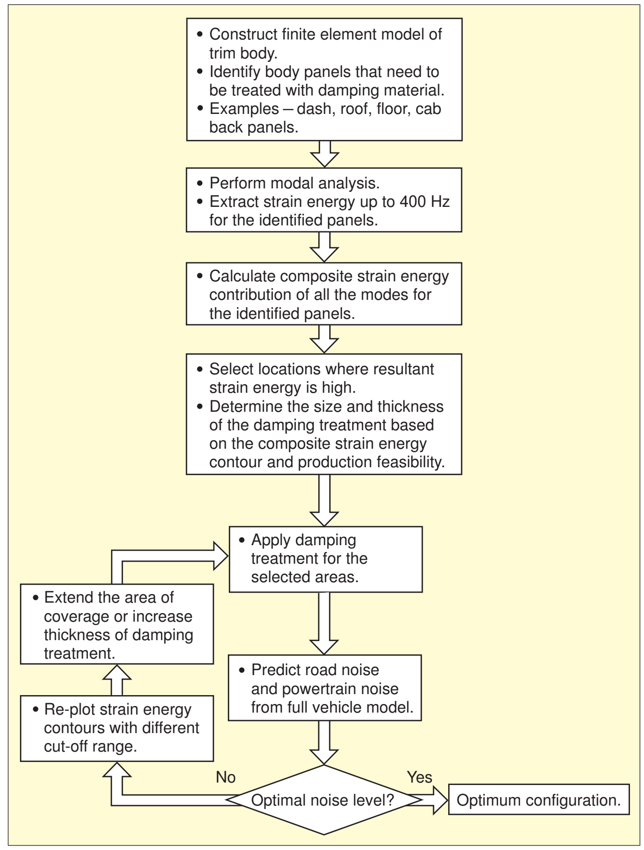

Figure 2. CAE methodology process flow chart.

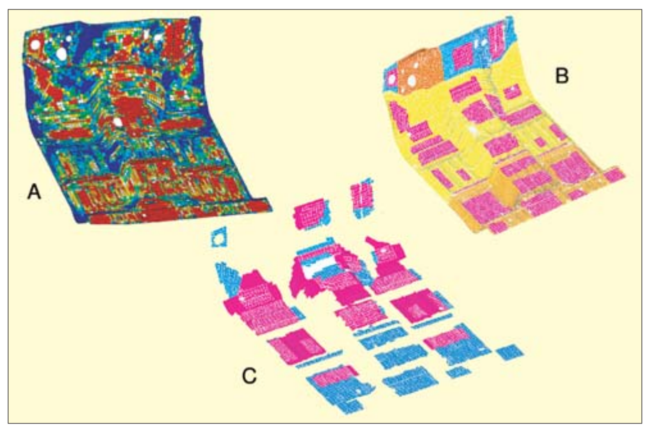

Figure 3. A – composite strain energy contour, B – damping treatment configuration (in magenta) identified by the CAE methodology.

The damping treatment configuration as identified by the experimental approach is based on a laser vibrometer test of the BIW structure of the cab of the pick-up truck. The structure was excited at the FESM (Front End Sheet Metal) mounts using a shaker. The input was a random excitation applied over a large frequency range. The resulting structural vibrations were mapped to obtain velocity contours, and accordingly, the damping-treatment locations and size were identified. Nevertheless, it is interesting to observe that the two damping treatment configurations agree remarkably well. However, there are some differences. In particular, the CAE methodology identifies a few additional locations for damping treatments at the rear of the rear floor panel, which were not identified by the experimental approach. This is perhaps due to the fact that the BIW structure was excited only in the front, which may have failed to excite the rear part of the rear floor panel adequately.

As a part of the validation process, the damping treatment configurations as identified by both the simulation methodology and experimental approach are incorporated in a full vehicle NVH model and the vehicle road noise performances are

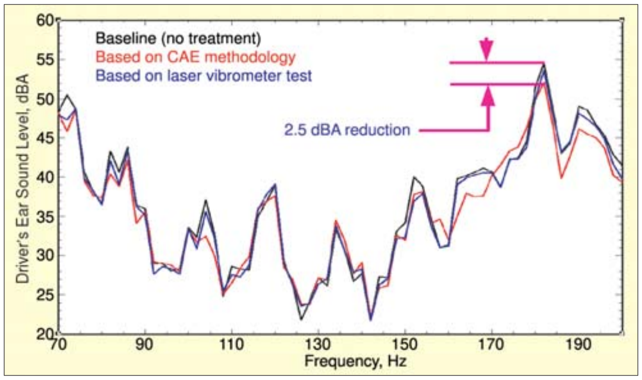

Figure 4. Road noise performance comparison between 70 and 200 Hz.

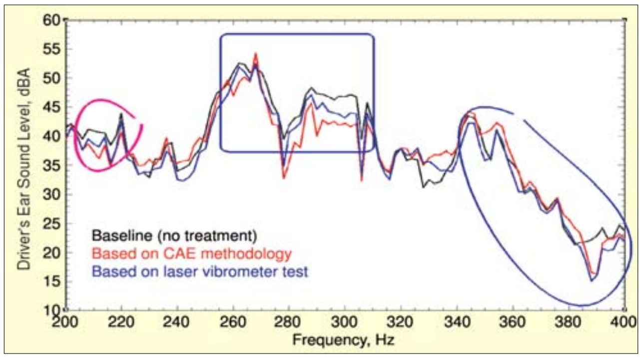

Figure 5. Road noise performance comparison between 200 and 400 Hz.

evaluated. These results are shown in Figures 4 and 5 in which the A-weighted sound level at the driver's ear is plotted in dBA as a function of frequency. In particular, Figure 4 presents the results for the frequency range from 70 Hz to 200 Hz, and Figure 5 presents the results from 200 Hz to 400 Hz. The results are based on three configurations: (1) baseline without damping treatment; (2) with damping treatment based on the CAE methodology; and (3) with damping treatment based on laser vibrometer test. As seen from Figure 4, the peak noise response occurs around 180 Hz where the influence of different damping treatment configurations is evident. In fact, Configuration 2 as identified by the CAE methodology resulted in a 2.5 dBA reduction. However, in Figure 5, the benefits of any one of the two configurations is not clear.

Therefore, we go to Table 1, which presents a summary of road noise results in terms of root mean square pressure ratio (RMSPR) and the corresponding sound level in dBA over the two frequency ranges. In the frequency range 70-200 Hz, Configuration 2 improves the noise level by 16%, but Configuration 3 provides only 8.6% improvement. However, in the frequency range 200-400 Hz, Configurations 2 and 3 provide about the same improvement, i.e. 16% improvement over the baseline without treatment. Table 1 also provides a comparison among different configurations in terms of volume of the damping material used. This comparison shows that Configuration 2 uses nearly 40% more damping material than Configuration 3. This means that Configuration 2 has more surface area coverage of damping material, which contributes to improved noise performance in the 70-200 Hz frequency range. Therefore, a judicious choice of damping material thickness and coverage area is required if there is a constraint on the volume of damping material that can be used for a particular vehicle. However,

Table 1. Summary of road noise performance comparison.

Model	Volume (mm^3)	Road Noise (70-200 Hz) RMSPR	dBA	%Diff.	Road Noise (200-400 Hz) RMSPR	dBA	%Diff.
Baseline	0.0	111.28	40.9	–	144.19	43.2	–
CAE	3.10×10^6	93.55	39.4	–15.9	120.97	41.7	–16
Test	2.21×10^6	101.74	40.1	–8.6	121.12	41.7	–16

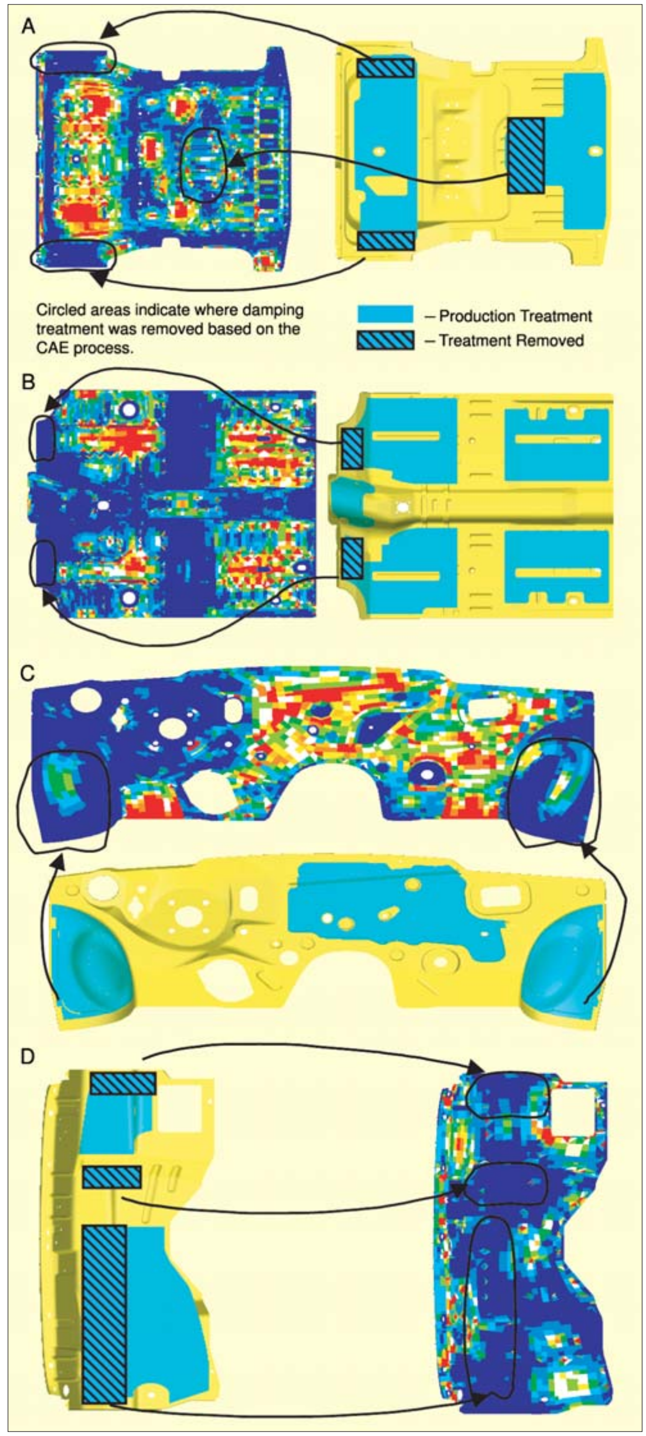

Figure 6. A – damping treatment optimization for front floor panel of a car, B – damping treatment optimization for rear floor panel of a car, C – damping treatment optimization for dash panel of a car, D – damping treatment optimization for dash panel of a car.

since the developed CAE methodology can be used early in the design cycle, the added weight can be adequately accounted for in combination with structural optimization of the panels.

The second segment (Figures 6-8) presents numerical results that demonstrate how the developed CAE methodology has been applied successfully in a few other vehicle programs at DaimlerChrysler. These results primarily focus on generation of composite strain energy contours and how these contours are used in optimizing existing damping treatment configurations. Figure 6 shows the composite strain energy contours for a car. As shown, the areas that are identified in blue are stiff regions of the structural panels. Accordingly, the damping treatments in those locations are removed without any degradation in noise performance of the vehicle. The locations where

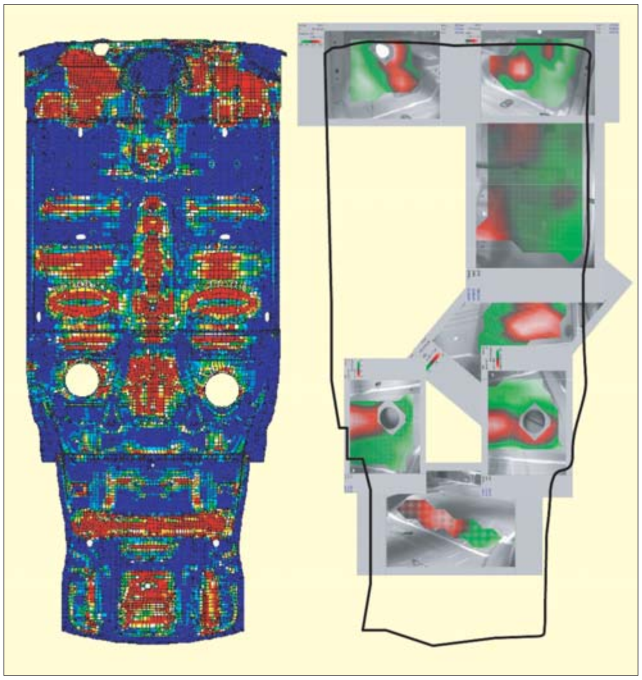

Figure 7. Comparison between strain energy contours (left) and velocity contours (right) for a minivan.

damping treatments were removed are circled in Figure 6. Although not shown, the optimized damping treatment configurations were incorporated in both the full vehicle NVH model and vehicle, and their impact on noise performance was evaluated. The results indicate that the optimized damping treatment configurations, in fact, maintained the same noise performance as that of the baseline damping treatment configuration, but with a significant weight and cost savings.

Similarly, Figure 7 shows composite strain energy contours for a minivan. These contours include modes up to 300 Hz. Figure 7 also shows velocity contours generated by using laser vibrometer tests. These contours are generated by exciting the structure at the front cradle rear attachment and the rear cradle front attachment on the right side of the vehicle only. A comparison between these two contours is revealing. The strain energy contours generated by the CAE methodology agrees remarkably well with that of the test. Based on the strain energy contours, the existing damping treatment configuration for this vehicle was optimized as shown in Figure 8. In particular, Figure 8a shows the floor panel with baseline damping treatment configuration and Figure 8b shows composite strain energy contours. Based on these contours, the damping treatments were optimized, with the resulting configuration shown in Figure 8c.

Concluding Remarks

A CAE methodology based on finite element modeling technique was developed to optimize damping treatments of automotive vehicles. The methodology uses modal strain energy information of structural panels that need to be treated with damping materials. The methodology was validated for vehicles at DaimlerChrysler Corporation. The methodology is simple, reasonably accurate and takes far less computational time than conventional experimental approaches. Furthermore, it can be integrated seamlessly with full vehicle NVH simulation models so that damping treatment optimization can be addressed in the early stages of the design cycle. In summary, the damping treatment optimization methodology offers tremendous potential in saving product development cost and time as well as in improving noise quality.

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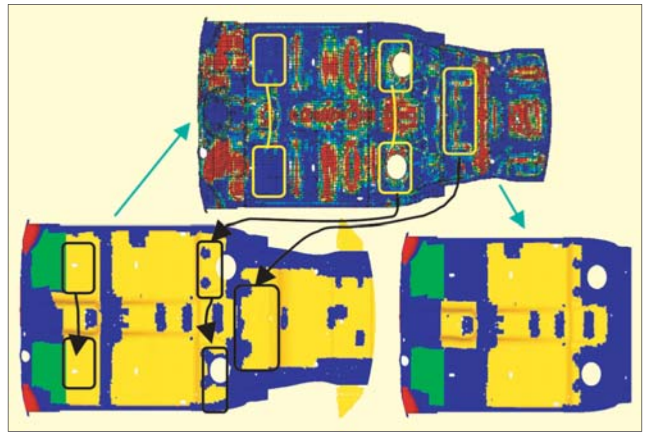

Figure 8. Damping treatment optimization for a minivan.

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