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"Digital Prototype" Simulations to Achieve Vehicle Level NVH Targets in the Presence of Uncertainties

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ABSTRACT

"Digital Prototype" simulations have been used at DaimlerChrysler to achieve vehicle level NVH objectives. The effectiveness of these simulations to guide the design when faced with vehicle parameter uncertainties is discussed. These uncertainties include, but are not limited to, material properties, material gauges, damping, structural geometry, loads, boundary conditions and weld integrity. Manufacturing and assembly processes introduce variations in the nominal values of these parameters resulting in a scatter of vehicle level NVH simulation responses. An example of a high frequency NVH concern will be studied and modified to arrive at robust design guidance when faced with uncertainty. The validity of a "deterministic digital prototype" simulation model and its relevant role as a "trend predictor" rather than "absolute predictor" tool in guiding the design is also discussed.

INTRODUCTION

The reduction in program cycle time for new vehicles has made it necessary to use CAE simulations up front in the program. A process to achieve NVH goals using "Digital Prototype" simulations is used at DaimlerChrysler to guide the product design and reduce the number of prototype builds whenever possible (1.). This process is however deterministic in nature. It assumes that the data used to build and run the simulation model is known with precision. In reality, model parameters are uncertain due to manufacturing and assembly process variations. This gives a scatter of response due to all the possible combinations of model parameters. The actual behavior of the system is probabilistic in nature since it lies within this scatter. This is in contrast to the conventional "deterministic" simulation model, which predicts a single response assuming nominal values for model parameters. The uncertainty in the simulation model arises due to variations in parameters such as loads, boundary conditions, material properties, and the geometry. The presence of uncertainties in the prediction capabilities of a simulation model would require a band prediction to predict absolute behavior and assess correlation comparison to a band of test variation.

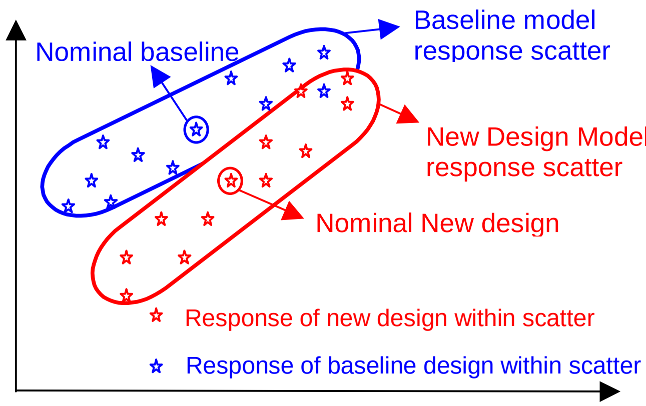

Figure 1: Typical Analytical Model Response Scatter

Robust design changes based on simulation model results should be done with caution in the presence of uncertainty. Firstly, a physical understanding of the problem should be obtained using appropriate diagnostic tools and applying the "Principle of Reasonableness". This step will yield the most dominant (weak) path in the vehicle structure. In the next step, structural design changes to improve this "weak" path should be studied in the presence of uncertainties. This step will give response scatters for "baseline design" and "new design" as shown in Figure 1. Some of the "new design" model responses will not show as much benefit as the nominal "new design" model response. In the extreme case, a combination of parameters can cause some "new design" model responses to be within the "baseline design" scatter. The criteria for evaluating the effectiveness of a design change should account for this scatter. The "robustness" of a design change should be defined based on the improvement of "new design" scatter over "baseline design" scatter.

This paper discusses the issue of using simulation model results to guide the product design in the presence of uncertainties. A specific example of high frequency axle noise simulation is used for the discussion. The "baseline response" of the full vehicle system model measured at driver's ear is shown to be below target. The diagnosis shows the "weak" path responsible for under-target performance to be the rear shock. A bracket at the rear shock along with softened rear shock bushings is proposed to improve the "weak path". The "robustness" of these changes in the presence of uncertainty in shock bracket parameters is discussed.

PROBLEM DEFINITION

MODEL AND LOAD DESCRIPTION

The simulation model used in this paper was a full vehicle representation with the interaction of all subsystems included. Sufficient detail is included in the models for body, suspension, powertrain and acoustics to predict trends in sound response at Driver's ear up to 500 Hz. Figure 2 shows the vehicle model used and has about 500,000 degrees of freedom. The components modeled using shell elements include body structure, body brackets, cross-members, subframes and suspension links. Reduced degree of freedom models derived from detailed FEA representations include powertrain, steering column, IP, tires, seats, and liftgate. Doors were modeled with rigid representation attached to only mass load the adjoining structure. The axles were modeled using lumped mass representation for the differential and carrier housing. The beam representation was used to model axle tubes, half shafts, axle shafts, prop shafts, springs, shocks and stabilizer bars. The acoustic cavity necessary for noise prediction is shown in Figure 3. The powertrain and chassis are shown in Figure 4.

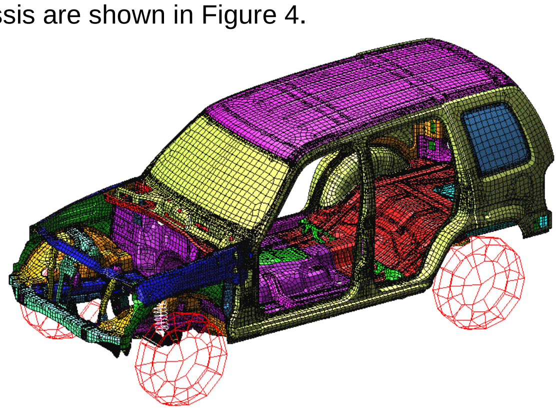

Figure 2: Full Vehicle Simulation Model

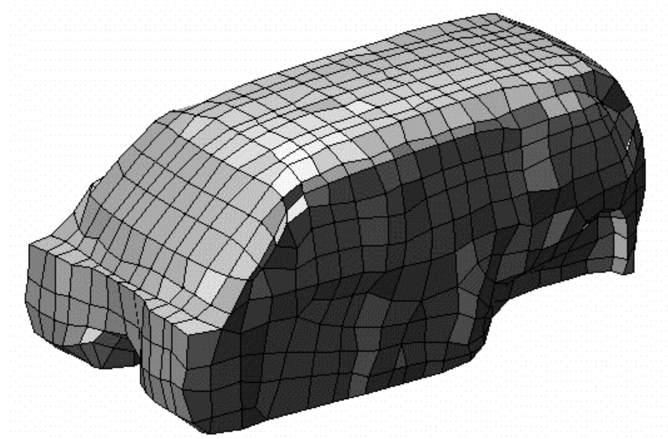

Figure 3: Interior Acoustic Cavity Model

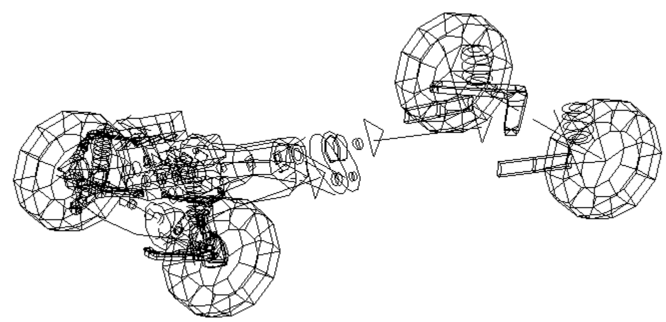

Figure 4: Vehicle Chassis and Powertrain Model

Axle noise simulation is used as a specific example in this paper for the purpose of discussion. The load condition for axle noise is an enforced angular rotation due to ring gear-pinion mesh misalignment. It is applied simultaneously at the front and rear pinion shafts. The response at driver's ear nominal location is used as a model output and is compared relative to the simulation target level.

Figure 5 shows the simulation response of a baseline model with nominal values used for all the model parameters. This "deterministic" model response does not meet target due to a peak in the response at 450 Hz (56 mph). The target was established using simulation results and testing of a reference vehicle along with competitive vehicle testing to arrive at a simulation target for the new model to achieve. This process is described in detail in Reference 1 and was based on using deterministic models. Before embarking on a path of solution search to achieve target, the analyst needs to answer the question: "*With what level of confidence can the simulation model be used as a design guidance tool in the presence of uncertainty*?"

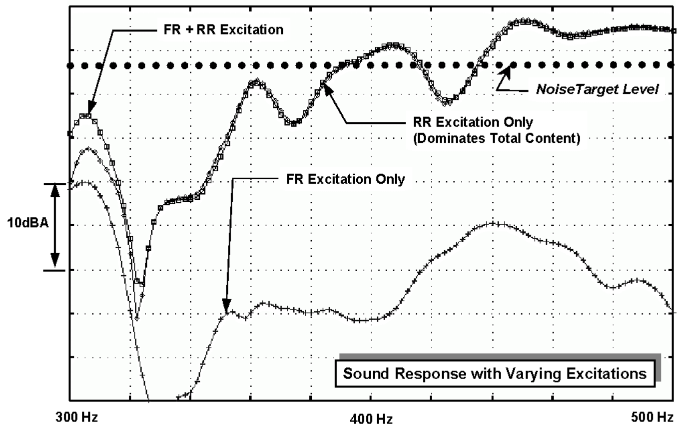

Figure 5: Axle Noise Response at Driver's Ear with Contributions from Front and Rear Axles

MODEL APPLICABILITY IN THE PRESENCE OF UNCERTAINTIES

Now that the model and load are established, it is desired to establish the degree to which the model can represent the actual physical hardware's frequency response. First let us consider the sources and factors contributing to uncertainty. Kompella and Bernhard (2.) measured 99 Rodeo vehicles and established that the structure-borne sound varied in a band as wide as 20 dBs in the frequency range from 150 to 500 Hz. Figure 6 is from their paper has been duplicated here. Freymann and Stryczek (3.) measured several similar operating vehicles in a low frequency range of 0 to 200 Hz and measured a significant noise level scatter. Figure 7 is from their paper and is duplicated here. The plot indicates as much as a 12 dB variation in this range for a much smaller sample size than Kompella. Freymann went further and studied a full vehicle simulation subjected to many random combinations of component and process variations and predicted a sound scatter band of 15 dB's in width in the lower range of 0 to 200 Hz. This is shown in Figure 8 also duplicated from their paper. Thus, Freymann confirmed with simulation roughly what Kompella had measured in hardware for high frequency structure borne noise.

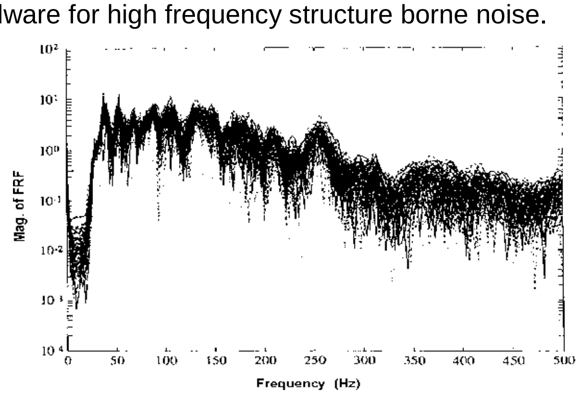

Figure 6: Magnitude of 99 structure-borne FRF's for the RODEO's for the driver microphone
(2.) © 1993 Society of Automotive Engineers, Inc

Clearly, some of the best engineered vehicles in the world are subject to a great deal of uncertainty. Presumably this noise scatter comes from component variation such as material modulus, gage tolerance, shape variation, damping layer effectiveness, and mass density variation. Manufacturing processes can contribute to scatter from poor welding, missed welding, bolting variation, missed attachments, part thinning from forming, and incorrect part substitution.

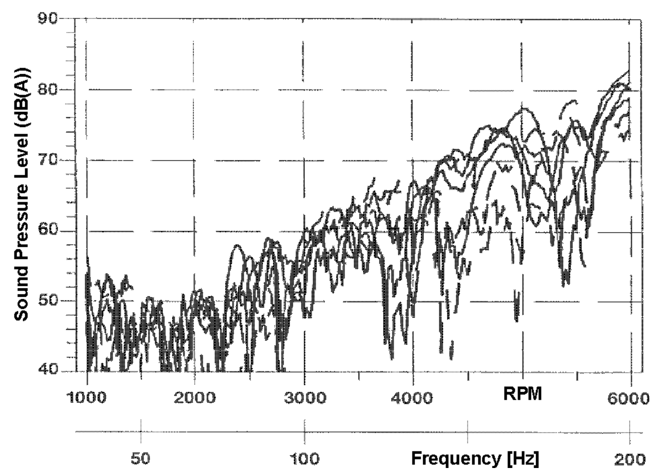

Figure 7: Scatter experimentally detected in the low frequency vibro-acoustic behavior of production vehicles (3.) © 2000 Society of Automotive Engineers, Inc

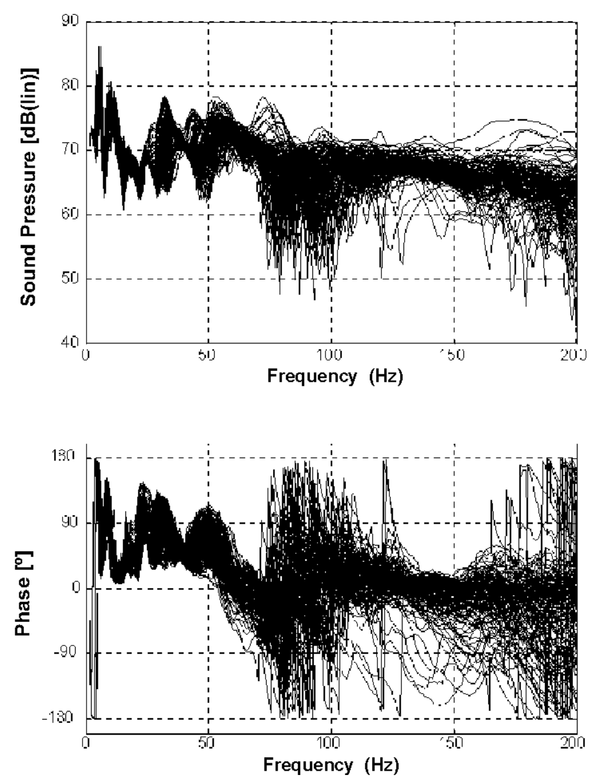

Figure 8: Acoustic scatter numerically determined in the vibro-acoustic behavior of a vehicle due to possible tolerances in the component area and in the production process
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The question of whether a model "correlates" to test must be established with the variation in mind. A reasonable criterion would be to expect the model to fall somewhere within the band of variation of the test data. This would represent a reasonably robust model and still represents a great deal of uncertainty due to the width of the band. It should be noted that for a vehicle program that is to have a short development cycle plan, it is not possible to have enough prototype vehicles to know the full range of variability if hardware were the only means to evaluate performance. This leads to the conclusion that simulation methods are a critical component to enable ever decreasing cycle times because the design decisions will have to be made before hardware is available. It also leads to the conclusion that, in the ideal situation, model correlation with test must be established on existing production vehicles with the purpose of defining the model building practices. Then the model predictions for a new design can be used with confidence to predict performance, correct it, and refine the new vehicle before a prototype is built. There will not be sufficient sample size to confirm correlation during the critical design release phase of the new vehicle program.

If methodology correlation is not established before a program starts, then the design process based on simulation must rely on a widely accepted practice that the model is valid as a trend predictor (1.). Confidence in the model is established from knowledge of historical success at predicting problems, from A-B design comparisons suitably confirmed in hardware, and from simulation predictions that can be shown to have results that follow from well established NVH principles, such as mode separation and isolation.

The model validity discussed in this paper is based on the assumption that the model is an adequate trend predictor (1.). A reference baseline model based on a predecessor vehicle is used as an indicator of existing performance, then a new simulation using the same modeling practice is used to project the relative change in performance of the new design. Knowing the proximity of the reference vehicle's test performance to the desired objective level to be achieved, results in an absolute target to be attained by the simulation (1.).

PROBLEM DIAGNOSIS

With a target and a model performance level established, the task of determining the best possible problem resolution begins. Since the problem is in a very high frequency range from 300 to 500 Hz it is expected that the band of variation is 20 dB's wide indicating a great deal of uncertainty. The basic principle to guide towards goal achievement is based on the "Principle of Reasonableness". Mr. Freymann (3.) made an excellent observation in his conclusions about the uncertainty of sound scatter variation when he stated "As a consequence of this (i.e. "scatter") it can be stated that a thorough vibro-acoustic analysis does require the involvement of a highly skilled and experienced workforce." A high degree of skill and experience is required for and is the essence of the Principle of Reasonableness whether applied by an NVH analyst using a simulation or a test engineer in an NVH Laboratory.

SYSTEM LEVEL DIAGNOSIS

There are great deals of new tools available to the NVH analyst and when used with some degree of experience, good design direction can be established. It was noted from Figure 5 of front to rear contribution that most was coming from the rear. Intuitively, this was expected since the front had a lone differential mounted to the body with halfshafts out to independently suspended front wheels. The rear was a solid axle and inherently harder to isolate from axle noise due to the need for the attachment of control arms, springs, and shocks. Thusly, the first test of the Principle of Reasonableness was passed and the problem was expected at the rear of the vehicle where design modifications to improve performance were focused.

A plot of the baseline axle noise response is shown in Figure 9. Additionally, several disconnect studies are shown, where a suspected key noise path into the body is eliminated. This is a technique that is sometimes used in a lab test, however, in a test there is some difficulty fixing the body side of the suspension bushing to ground. This is easy to do in a simulation and is shown schematically in Figure 10 for a shock disconnect study. It is vital that the bushing be grounded on the body side such that the suspension motion on the suspension side be relatively unaltered. That way the body receives nearly the same input from the remaining connected paths. Removing a suspension component, as is sometimes done in a test for ease of setup, is not desirable since the suspension motion is altered significantly, and potentially contributing resonances of the component are eliminated. Only one path, or pair, is disconnected at a time so that the difference from the base level is easily seen in the plot.

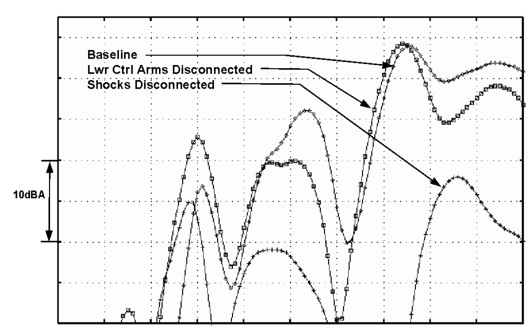


Figure 9: Baseline Vehicle Prediction of Axle Noise and Results of a Disconnect Study of the two dominant Noise Paths from the Rear of the Vehicle

Operating deformed shapes, or forced mode animations at a given frequency where the response reaches a peak, provide a visual image of how the axle system is moving and exciting the body through the suspension links. The animation and the disconnect studies give a good intuitive ranking of the key noise paths.

The operating deformed shape motion exhibited a pitching motion of the solid axle rotating about the ring gear axis. Since the input excitation is a mesh transmission error imposed as an enforced angular rotation between the pinion and ring gear, it would be expected to impart a rotation of the axle assembly about the ring gear axis. Thus, another test of the Principle of Reasonableness was passed. The pitching motion of the axle assembly imparts motion to the body at all suspension control arms as well as the shocks so there was no clear indication of which was the main noise path based only on the animation.

The rear suspension as a stand alone sub-system was studied for suspension link resonances. There were no flexible link resonances in the range of operation. Rigid link resonances of the suspension components on their bushings were in the range but none of these modes aligned with the motion of the axle housing. Thus, no resonance conditions of the suspension links could be seen in the operating deformed shape.

From Figure 9, it is clear that disconnecting both shocks seems to be the most significant effect. The disconnect study is directional but has a great deal of uncertainty in terms of ranking the best path from a feasible point of view since it is not a physically realizable configuration.

To assure that the key path is considered, forced mode animations of the disconnected configurations were studied. It was noted that the largest amplitude of motion was on the floor and concentrated at the rear. This was expected based on Figure 5. At this high frequency, it was noted that there were many nodes and anti-nodes in the floor panel vertical motion. A contour plot of vertical floor pan motion is shown on Figure 11. Light areas indicate higher motion level than the darkened areas. It can be seen that the contours of the baseline and LCA Disconnect are nearly identical while the shock disconnect shows a general lowering of level across the floor as a whole. This points to a reasonable conclusion that the shock is a dominant path and that disconnecting the shock results in reduced sound levels due to a general reduction of input to the acoustic cavity coming from the rear floor motion.

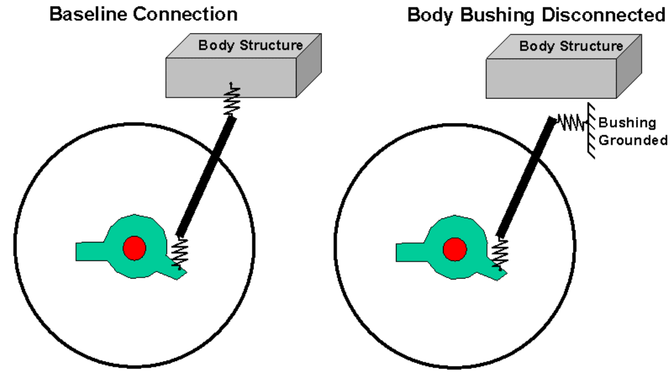


Figure 10: Schematic of the Disconnect Shock Absorber Configuration

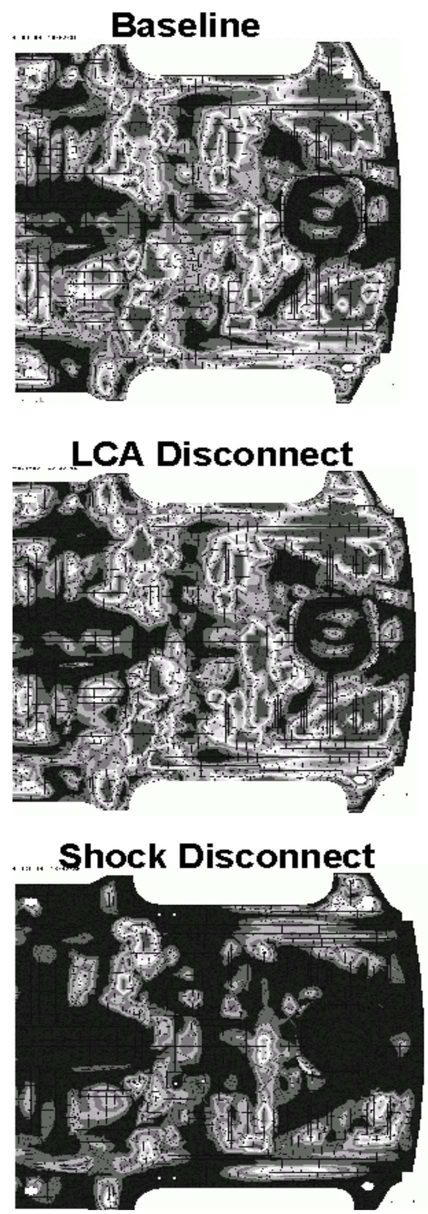


Figure 11: Contour Plots of Rear Floor Pan Vertical Displacement with Key Paths Disconnected

SUB-SYSTEM LEVEL DIAGNOSIS

To further confirm the shock path as the weak link, studies were performed at noise paths including a study of bushing rate design sensitivity analysis.

NOISE PATH SENSITIVITY

Since the shock was a suspected path, mobility and noise sensitivity studies were performed to see how close these parameters were to targets established in a goal setting phase (1.). The Mobility and noise path sensitivity for the right and left shocks axially are shown in Figure 12. This level of path performance is very good and much better than generic goals (see Ref. 1.). This pointed to high force levels as a dominant contribution to the noise. This is not uncommon for a solid axle configuration since the shock properties for NVH isolation must be balanced with those for handling considerations.

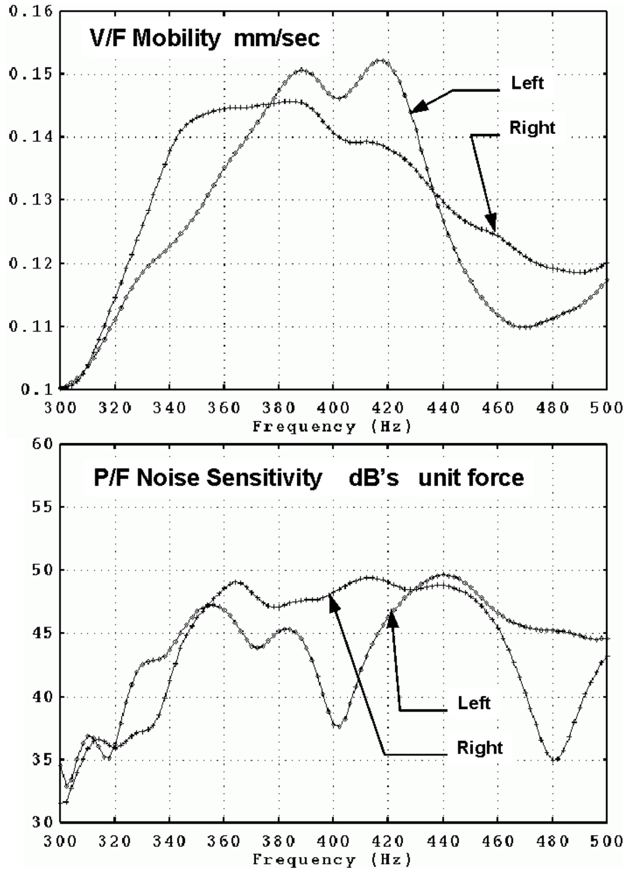


Figure 12: Point Mobility and Noise Sensitivity of the Shock Absorber Paths in the Axial Direction

NOISE PATH ANALYSIS

A complete noise path analysis (NPA) under the axle noise operation condition was conducted using the simulation model (4.). The forces through the body noise paths come from the vehicle model excited by the mesh transmission error input. While noise path (P/F) sensitivities come from exciting the body-acoustic system alone at each individual noise path. The contribution from each path is summed including phase relationship to form the total sound. Then the contribution from each individual path can be compared to the total sound vectorially to rank the highest contributing paths. As a check on the summation, it is compared to the total sound calculated by the original model. Since simulation data was used here and is not subject to testing error, the total sound level from the path summation overlays exactly with the full vehicle prediction. If an overlay is not obtained, this indicates a processing or numerical approximation error in the data.

Figure 13 shows the results of the full NPA. It ranks the rear left shock as highest, followed by rear left lower control arm (LCA), and third the rear right shock. The shocks again appear to be dominant paths. The LCA has positive and negative contribution which would cancel during the disconnect explaining why that path had little effect on Figure 9.

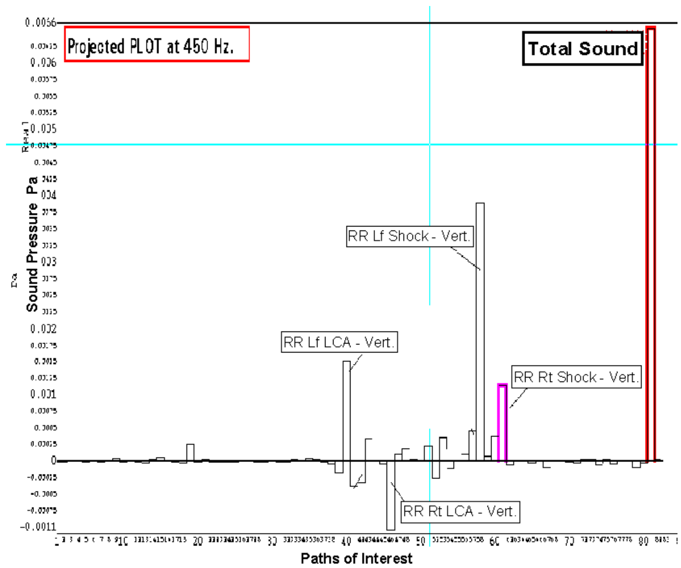


Figure 13: Path Contributions to the Total Sound from a Transfer Path Analysis of the Full Vehicle NVH Simulation Data

NOISE PATH DSA SIMULATION

While an NPA appears to be good at ranking the key path contribution, it is not quantitative in that it can not be used to predict a change in total pressure as a result of an incremental change in the design. However, a widely used approach called Design Sensitivity Analysis (DSA) is available and was applied to this problem, again to confirm the correct ranking of the noise paths (5.). DSA does not rank the contribution of the path to the total sound but instead provides an estimate of how much the total sound level will change quantitatively for a fraction of rate change of the body bushing at the noise path.

$$DSA = DeltaP_t / (DeltaK_i/K_i)$$

Where DSA is the sensitivity quantity and $DeltaP_t$ is the change in total pressure at the response location. Alternately;

$$DeltaP_t = DSA * (DeltaK_i/K_i)$$

This says that a change in total pressure at the response point can be quantitatively estimated from the value DSA and a fraction change in bushing rate. It has been shown that the estimate is accurate as long as the fraction change is limited to approximately 20%.

The results of the DSA simulation indicated that the right shock was the largest contributor and that the left shock was actually beneficial when evaluated at the sound peak. The right shock contribution was twice as large as the left shock benefit. This is in conflict with the NPA analysis, however, they both point to the shocks as a key path.

To determine which indicator was directionally correct, a reanalysis was performed. A reduction of 20% of the shock bushing axial rate was simulated first on the left, then on the right. The results are shown in Figure 14. A reduction of rate on the left side shock bushing has increased the noise level as expected from the DSA projection which indicated that this path was beneficial and not contributory. It can be seen that reducing the rate of the right shock bushing reduced the noise level more than the noise increases due to the left bushing change. This results because the right side bushing rate sensitivity is twice that of the left while opposite in sign. Figure 14 also shows the accuracy of the DSA projections as the % deviation between the projected changes from equation 1 and the results of the full system reanalysis for each change evaluated separately. It can be seen that the worst case projection was accurate to 2.2%. Based on this, it appears that a DSA analysis was a better indicator of design direction than a full NPA summation.

DIRECTION FOR DESIGN IMPROVEMENT

Based on the diagnostic procedures in the previous sections, it was concluded that the shock path was the weak link and the highest contributor to interior noise at the driver's ear. It appears that operating deformed shape animations, disconnect studies, and a DSA ranking analysis can be effective at establishing the reasonableness of the results and determining the weak link for design corrective action.

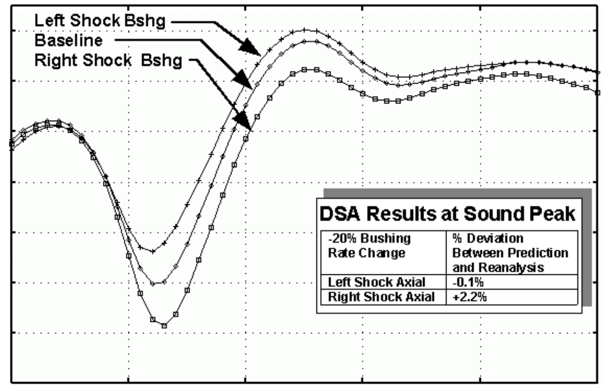
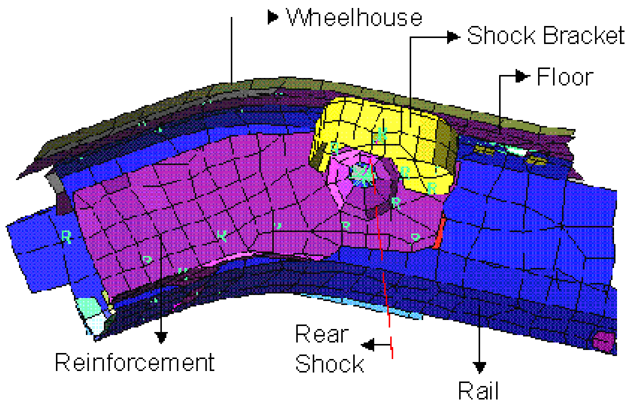


Figure 14: Path Contribution Reanalysis to confirm DSA Projections for Key Noise Path Contributors Based on Full Vehicle Simulation

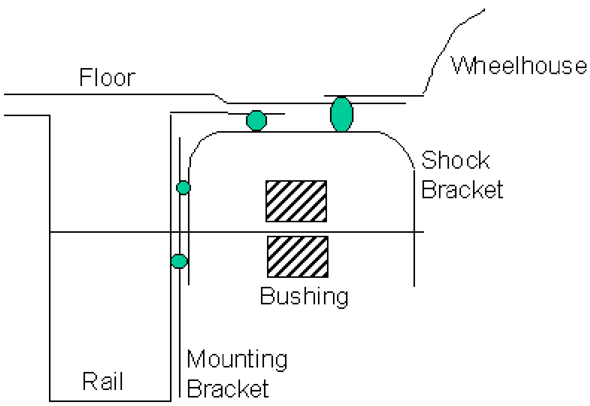
PROBLEM SOLUTION

CORRECTIVE ACTION

The design corrective action for the axle noise problem was based on the diagnosis of the rear shock path as the weak link. High force levels into the body at rear shock and body dynamic stiffness being less than the generic goal of 5-10 times shock bushing stiffness offered an opportunity to soften these bushings. The upper and lower rear shock bushings were softened in all the directions, but more importantly in the vertical direction by 65% at the upper bushing and by 20% in the lower bushing. This design change, however, had to be balanced against the vehicle handling requirement of stiff bushings. In order to meet this conflicting requirement of NVH and handling, an effort was made to increase the local attachment stiffness between rear shock and the body. It was accomplished with a bracket design (Figure 15) that improves the shock attachment and compensates for the loss in stiffness due to softened bushings. The shock bracket was welded to the rail through 2T welds, to the floor and wheelhouse through 3T welds and to the rail and reinforcement through 3T welds. The stiffness improvement due to the shock bracket was seen in the mobility analysis that showed 3 times improvement at the right shock and 1.5 times at the left shock. Therefore, the shock bracket in itself was expected to improve isolation since it strengthened the shock attachment of the body existing in the baseline design.



Configuration of Shock Bracket



Rear View Section of Shock Bracket

Figure 15: Shock Bracket Design

The axle noise response comparison between the baseline model and the Design Model with rear shock path changes is shown in Figure 16. These responses are calculated with "deterministic" model using nominal model parameters. The driver's ear response with only shock bracket change shows more improvement compared to only rear shock bushing changes. This points to the importance of a configuration change to strengthen the rear shock to body attachment. It is necessary even though the mobility performance at rear shock was better than generic goals. The generic goals, therefore, should be used as first principles only. Also, softer bushings are less effective here as the body-to-bushing dynamic stiffness ratio for the baseline model is less than the generic goal of 5-10. However, it is worth noting from Figure 16 that benefits of softer bushings were more evident once body-to-bushing stiffness ratio was improved because of shock bracket, which locally increased the body stiffness (6.). This suggests that configuration changes in the weak link should be investigated first before bushing modification type changes are considered. This is quite intuitive and follows the "Principle of Reasonableness". Now that the improvements are seen in the deterministic models, it is necessary to establish the degree of confidence in design changes in the presence of uncertainty. This will define the robustness of a new design before it is recommended and tested in hardware.

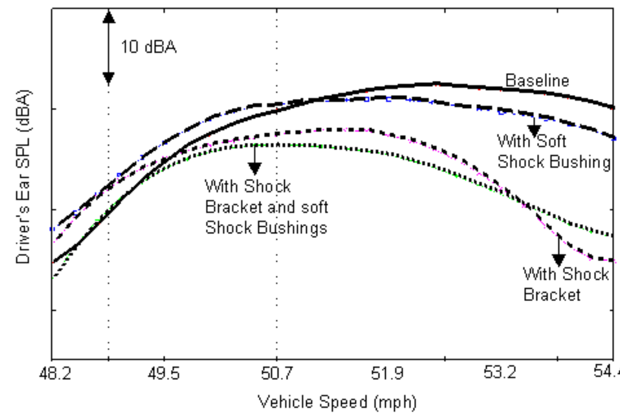


Figure 16: Comparison of Baseline Model Response with Shock bracket and Shock Bushing Proposal

ROBUSTNESS OF CORRECTIVE ACTION

Parameter variations defined in Table 1 are considered in this study to define the robustness of the axle noise solution. The components mentioned in Table 1 are shown in Figure 15. The last two columns of the table indicate parameters considered in the variation study that are unique to the Baseline and Design Models (with shock bracket and soft bushings).

Simulation responses are generated for Baseline and Design model for all parameter combinations shown in Table 1. It includes responses obtained from "deterministic" models with nominal model parameters. A scatter band around the nominal responses (Figure 17) is then obtained by taking an envelope at every frequency of all response curves. The deterministic model response is shown as a 'bullet' at a vehicle speed where the deterministic model response reaches a peak. These scatter bands define the range of variability in output responses due to model parameter variations shown in Table 1. In reality, these scatter bands might be different since not all parameter variation combinations that can happen in a real vehicle are considered here. However, scatter bands shown in Figure 17 have wealth of information to draw important observations.

Table 1 Parameter Variation Details

Parameter	Variation	Baseline Model	Design Model
Gauge for rear rail, rear floor, wheel house and reinforcement	±5%	Yes	Yes
Young's Modulus for rear rail, rear floor, wheel house and reinforcement	-5%	Yes	Yes
Gauge for Shock bracket	±5%	--	Yes
Rail to shock bracket welds	Missed all	--	Yes
Shock bracket to floor to wheelhouse welds	Missed all	--	Yes
Rail to shock bracket welds	Missed some	--	Yes
Shock bracket to floor to wheelhouse welds	Missed some	--	Yes
Shock bracket to floor to wheelhouse and shock bracket to rail welds	Missed some	--	Yes
Shock bracket to floor to wheelhouse and shock bracket to rail welds + Gauge for rear rail, rear floor, wheel house and reinforcement	Missed some & +5% (for gauge)	--	Yes

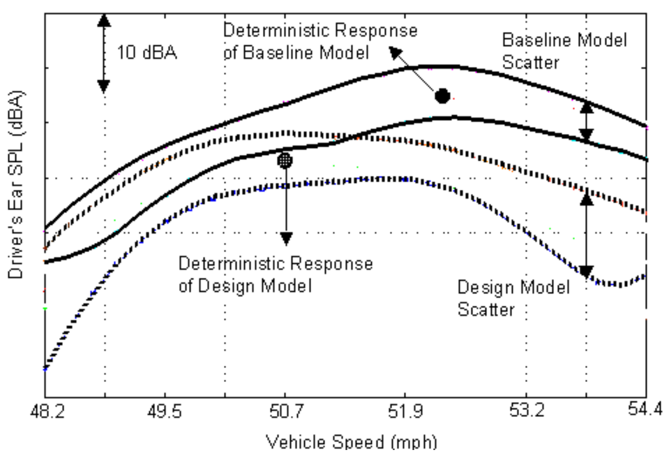


Figure 17: Comparison of Baseline and Design Model Response Scatter

The Design Model response scatter shows improvement over the Baseline Model response scatter around the problem speed range of 52-53 mph. Also, the scatter bands do not overlap in this vehicle speed range. This indicates that even in the presence of parameter variation, there is a high probability that the Design Model will show improvement over the Baseline Model. This gives a higher degree of confidence in the design recommendation of shock bracket. The improvement in response might vary depending on the combination of parameter variations. The recommendation can thus be considered robust in the presence of parameter uncertainty.

It was found that the parameter variation due to missed welds defined the scatter band of design model at a majority of vehicle speeds. The gauge variation, on the other hand, gave a minimal spread around the nominal response curve. This suggests that a design change involving configuration or topology changes such as the shock bracket are more critical than those due to material gauge or material properties. It is also worth noting that the Design Model response scatter is narrow around 52 mph. The scatter widens at vehicle speed below and above 52 mph. Thus, it is seen that when a design change is made that was targeted to affect a specific frequency/speed and based on an appropriate diagnosis, the model response is more robust and less sensitive to variations in the new design.

The deterministic response of Design Model (Figure 17) falls in the center of Design Model Scatter and can be viewed as mean of Design Model responses obtained due to parameter variation. Therefore, the deterministic response represents the response population in a statistical sense and is representative of the response scatter for evaluating design change. This observation is typical of a configuration type change to the design such as adding a shock bracket. During the initial vehicle design phase, most of the corrective actions are such configuration changes needed to improve weak path performance or to better manage vehicle modes. The deterministic models should therefore be relied on during the initial design guidance. Their importance in recommending corrective actions should not be overlooked. An exhaustive parameter variation study can be done at a later stage to define the robustness of the solution. The use of deterministic models should however be limited to trend prediction only since the presence of parameter uncertainty will vary the improvement seen in response due to corrective action (Figure 17). Now that the rear shock path is improved, the problem peak has shifted to a new vehicle speed of 50.7 mph as seen from the Design Model response in Figure 17. Further diagnosis needs to be done to determine the new contributing paths. This is typical of the NVH target achievement process, where, once the most dominant path is improved, the problem shifts to the next highest contributing path. This is an invariant serial problem solving approach whether done by simulation or physical hardware testing. Optimization techniques can improve the serial natural; however, to be effective, they must allow configuration changes similar to the with or without shock bracket case which are difficult to pose for more than one noise path.

The question of "*With what level of confidence can a simulation model be used as a design guidance tool in the presence of uncertainty*?" should be revisited in view of the discussion so far. The diagnostic tools used on a deterministic model were shown to be successful in identifying the rear shock as the weak path for axle noise. The model was also successfully used in guiding the design change for axle noise improvement in the presence of uncertainty. Root cause analysis and the corrective action proposed using a deterministic model were confirmed in prototype tests conducted nearly a year after the design decision to modify the bracket was implemented. The simulation model thus served as a good design guidance tool in the presence of vehicle parameter uncertainty.

CONCLUSIONS

The effectiveness of "Digital Prototype" simulations to recommend design corrective action when faced with vehicle parameter uncertainties has been affirmed. It is shown that the robustness of the corrective action involves using appropriate tools and "Principle of Reasonableness" to diagnose the problem and study the response scatters generated due to parameter variations. Robust solutions generated using simulations can reduce what-if studies done using physical hardware and help reduce prototype costs. The use of a "deterministic" model using nominal model parameters is still valid as a design guidance tool in the early phase of vehicle development when major modifications are configuration type changes. It is also shown that if such changes are made with proper diagnosis, the response improvement is less sensitive to parameter variations. The example of axle noise diagnosis and solution resulted in changes in the design before hardware testing was possible. Subsequent hardware testing affirmed that the model predicted the main noise path correctly.

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