Using an NVH Simulator to Develop Power-Train Mounting Systems

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Traditionally, power-train mount development involved the results from computer-aided engineering (CAE), which limits the engineer to producing graphs and animations that represent the tactile point responses. We've pushed this a step further by linking this objective data to subjective feel by using a full-vehicle simulator (FVS). This approach enabled the engineering team to engage senior management in this development process. Attributes important to customers were evaluated in a laboratory-based car that helped select the desired motor mounting specifications. This was useful given the challenges of balancing the attributes of ride comfort and power-train refinement. Bentley's commitment to improving fuel economy and carbon footprint that introduced variable-displacement technology on the new four-liter, V8 engine, posed challenges of developing mounting systems for, effectively, two engines in one car. To strike a successful attribute balance, experiments were conducted to compare the subjective performance of the mounts for all attributes. The FVS technique helped achieve the best balance in dynamic stiffness at different frequency ranges with best damping in a conventional hydramount. Physical simulation has been used on an increasing number of products within Bentley and has become a cost benefit owing to the reduction of vehicle tests in developing mounting systems.

Bentley requires world-class levels of ride comfort and refinement in its products. These attributes complement Bentley's tradition of power, poise and craftsmanship.

The technique was devised between Bentley Motors' Acoustics Department, Sound & Vibration Technology and DJ Fothergill Consulting. The full-vehicle simulator (FVS) technique described in this article supports the delivery of luxury motor cars that offer their drivers class-leading levels of ride comfort and power-train noise, vibration and harshness (NVH) control. These elements of attribute performance were achieved by tuning the power-unit mounting system for an optimal balance between these two opposing attributes. Initially, the technique was augmented by vehicle testing to rank alternative specifications and provide correlation with the simulation process. These tests were also used, following analysis, to confirm the attribute balance between ride comfort (power-unit shake) and power-train NVH.

Once the technique had been established, and confidence in correlation was accumulated, it was possible to optimize the power-unit mounting system with much less physical testing than before. The system model represented a very attractive investment for Bentley, as it would be for any car company, since it reduced the budget required to develop the car. The approach, which made use of both physical test and computer-aided engineering (CAE), has been derived to enable engineers to subjectively assess predicted specification changes without the manufacture of prototype parts. This has enabled Bentley Motors to save development time and cost by reducing the number of hardware test iterations for developing power-train mounting systems. This specification can then be investigated with the supplier for replication through manufacture, with all parties understanding the limits of specification through previous experience.

Full-Vehicle Simulation

The FVS was prepared with two clear attributes in mind: ride comfort and power-train NVH. Owing to the differences in source contribution and data acquisition parameters, it was necessary to approach both attributes with slightly different methods. For this reason, we examined the attributes separately.

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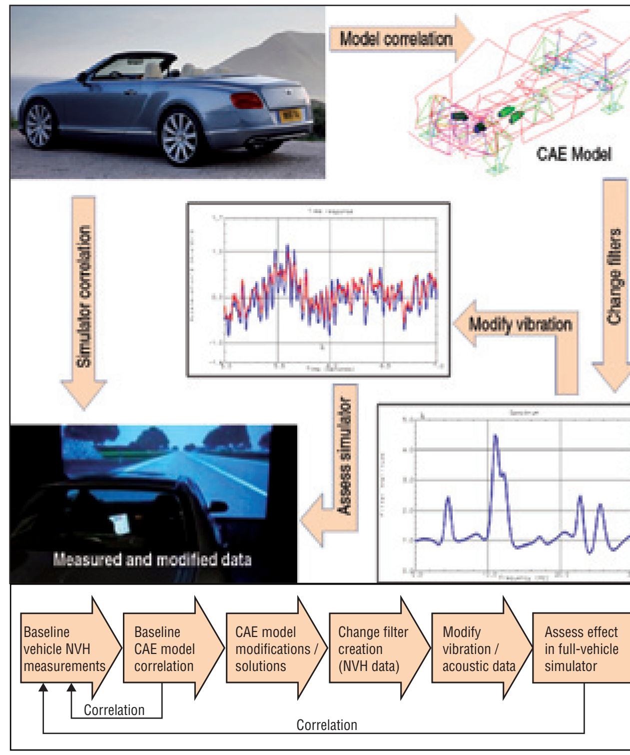

Figure 1. Full-vehicle simulator process.

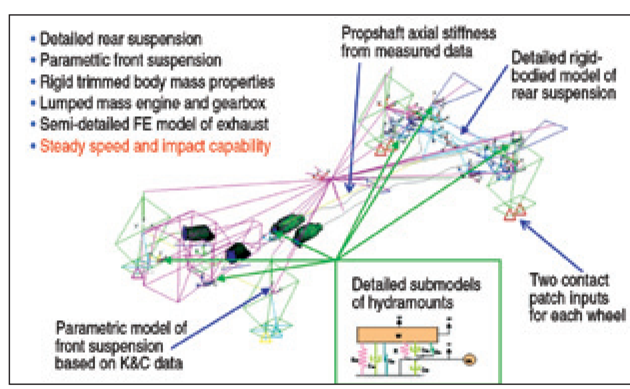

Figure 2. Modeling example.

Ride Comfort Approach. The FVS process for ride comfort can be broken down into simple steps and are illustrated in Figure 1. First, it was necessary to measure the baseline car to provide both a reference for changes to be made and to also provide a correlation reference for the CAE model. Second, the CAE model was created, which allowed a correlation loop back to the reference vehicle and also allowed the modelling of desired changes to the power-unit mounting system. The output of the CAE modelling step was to create a set of change filters that were used to change the response vibration based on the modified mount response. These signals, both reference and modified, could then be played into the physical simulator for subjective evaluation.

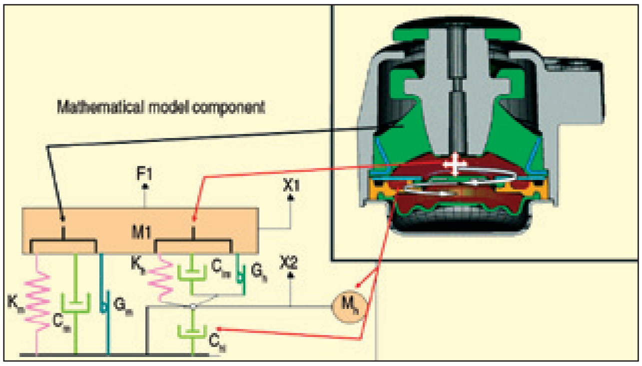

Figure 3. Hydramount model parameters related to physical components.

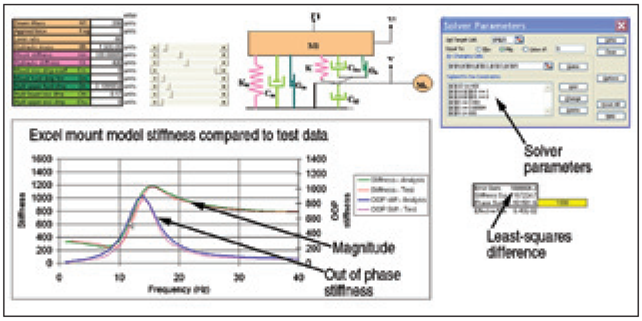

Figure 4. Hydramount submodel solver.

Ride Comfort Vehicle Measurements. The first step in the process, to measure the unmodified vehicle, was carried out with the final step of the process in mind. The tactile points used to replicate ride comfort and impact shake performance in the simulator plus the acoustic signals used to activate the headphones were all measured. These measurements were taken on a range of specified test surfaces for ride comfort responses and power-train NVH maneuvers.

Ride Comfort CAE Model. Following measurement of the baseline vehicle, the CAE model was created to replicate the compliant behavior of the car as shown in Figure 2. This could be done withmore or less detail as required. So, compliant elements that were not modelled in detail were represented parametrically. That is, mathematical constraints were created to reproduce data captured on a kinematics and compliance rig test. Clearly, modelling more compliant elements will give a more accurate prediction of the tactile point response, and it became a decision-making process as to how much detail was included in the model. It was possible, of course, to model only the power-unit mounts, as shown in Figure 3. However, when comparing this basic model to one that includes all compliance elements in the subframes, links, etc., plus the dynamic performance of the body structure, it became clear that evaluating the latter model would yield a subjective response more aligned with the intended result of the whole vehicle.

One of the interesting characteristics of the vehicle model is the way in which the road/tire interaction was modelled. The road surface waves are applied to the ground at the front and back of each tire contact patch. Scalar spring elements were then placed at these locations representing tire stiffness in compression, and (fore and aft) shear. A mathematically rigorous representation of the frequency-dependant phasing of the ground waves between contact patch input points resulted in the wheel being forced in both bounce and roll. This simplification worked well for low frequency shake, where the contact patch only spans less than half a spatial wave. For shorter wavelengths, the contact patch excitation would need to be refined. Had the excitation been applied at only one point per contact patch, then front to rear wheel phasing would have reproduced vehicle bounce and pitch effects, but there would have been no meaningful wheel rolling or fore and aft excitation.

The CAE model was created, and dynamic performance characteristics of the mounting systems were selected based on the kind of surface being modelled. For example, a moderate surface would be modelled using a ±0.1-mm powertrain mounting fixed-displacement curve, while a rough surface or impact events might use a ±0.5-mm fixed-displacement curve. The mathematical model for the hydramount was carefully modified to match the mount performance using the sub-model solver, as seen in Figure 4.

Ride Comfort Change Filter Preparation. The next stage of the process, once the CAE model had been created and correlation was achieved back to the vehicle measurements, was the preparation of change filters. These change filters were calculated from the modified specification and baseline specification. Equation 1 for the filter was calculated:

$$\text{Filter} = \frac{\left\{\underline{\ddot{\mathbf{a}}}(\omega)\text{Modified specifiction/excitation force}\right\}}{\left\{\underline{\ddot{\mathbf{a}}}(\omega)\text{Base specification/excitation force}\right\}} \tag{1}$$

where $\underline{\ddot{\mathbf{a}}}\,(\omega)$ is a matrix of acceleration spectra.

The process that was adopted was quite simple and allowed the engineer to apply the filter to each tactile point in turn. As the excitation force was cancelled out, due to both forcing functions being identical, we were left with a filter that was dimensionless and independent of the forcing function.[1] These filters, for each tactile point, were then applied to the model to produce a 'modified' set of tactile point responses (see Figure 5). This enabled the engineer to evaluate the effect of the specification change in the vehicle by evaluating the baseline specification and then the modified specification. The effect of the modification was then clear when examined in the spectral domain as shown in Figure 6.

Because the ride shake, including impact,[2] measurements were conducted during constant speed test runs, the power-train noise contribution was low. This means that the same power train noise sound track could be used for ride comfort evaluations, which eliminated the psychological effects upon vibration perception. Conversely, some specification comparison runs that had the same power-train noise sound track had actually been criticized for acoustic differences. This highlighted the interdependency of acoustic and vibration subjective assessments and how the vibration differences could affect one's acoustic perception.

One final problem to overcome was to consider the influence of the wheelbase pitching filter. As the front and rear wheels pass across the road surface, it was necessary to consider the influence of the car's pitch behavior. This was most noticeable at transducer locations nearest the car's pitching center. So, as the car pitches in response to the front wheel followed by the rear wheel, a transducer at the pitching center would see a diminished acceleration level compared to a point further forward of the pitching center. This phenomenon, known as the wheelbase pitch filter effect, is characterized in Figure 7.

Power-Train NVH Approach

The FVS process for power-train NVH is separate from ride comfort but can be re-aligned in the subjective assessment stage so that the engineer could evaluate both attributes in the car sequentially – just as in a real car.[3] The approach for data acquisition and CAE modelling was slightly different to that used for ride comfort, which we will detail in the same way.

Power-Train NVH Vehicle Measurements. The approach focused on the requirement to replay the acoustic and vibration data at various throttle openings. The noise and vibration simulation enabled the assessor to evaluate the car in a "free-driving" environment or at various fixed throttle openings in fixed replay mode; e.g. during part throttle, wide open throttle or overrun maneuvers. Initially, the technique was developed for use on a desktop simulator only. This allowed the engineer to evaluate the power-train noise performance through headphones only with a throttle pedal and steering wheel in front of a driving simulation screen. By using the FVS, both the acoustic and vibration behavior could be assessed in the same way as the ride comfort evaluations.

Power-Train NVH CAE Model. The approach for the power-train NVH CAE model worked in parallel with the ride comfort CAE model. The key difference from the ride comfort approach lay with the frequency range of interest. The ride comfort analyses were usually conducted from 0-40 Hz, with some linear extrapolation up to 200 Hz, while power-train NVH analyses required up to 1 kHz and beyond. In fact, the dynamic rating data were usable to

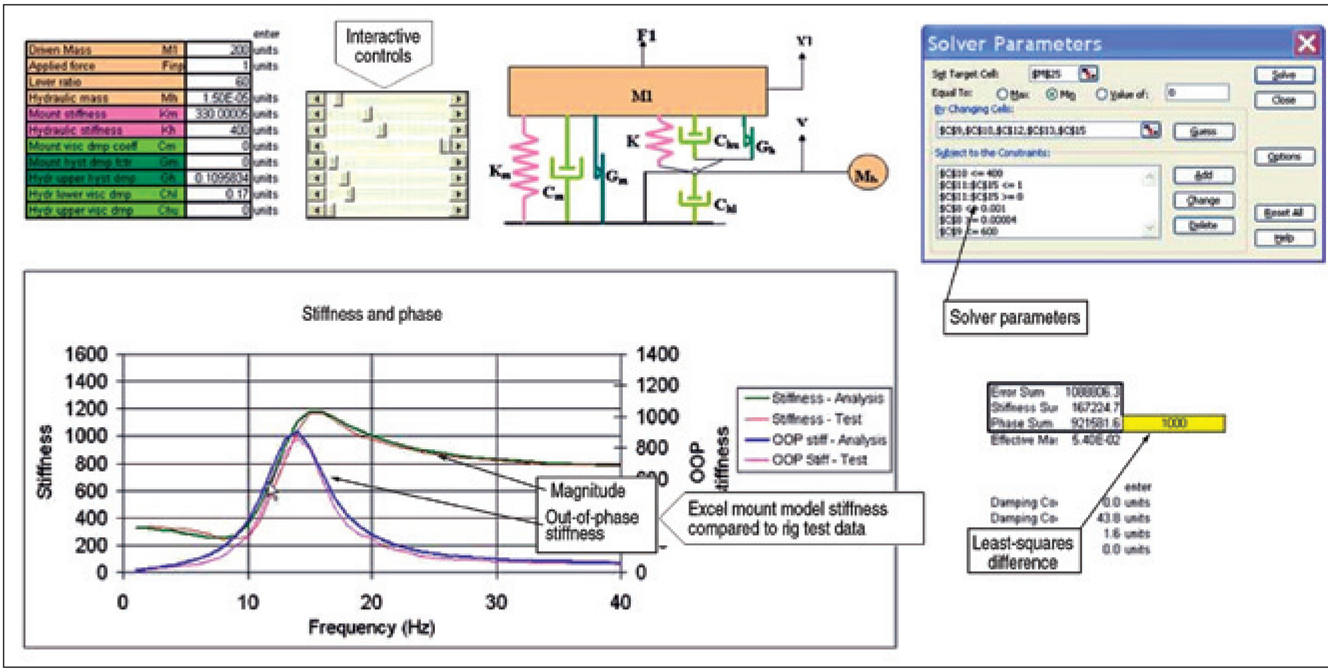

Figure 4. Hydramount submodel solver.

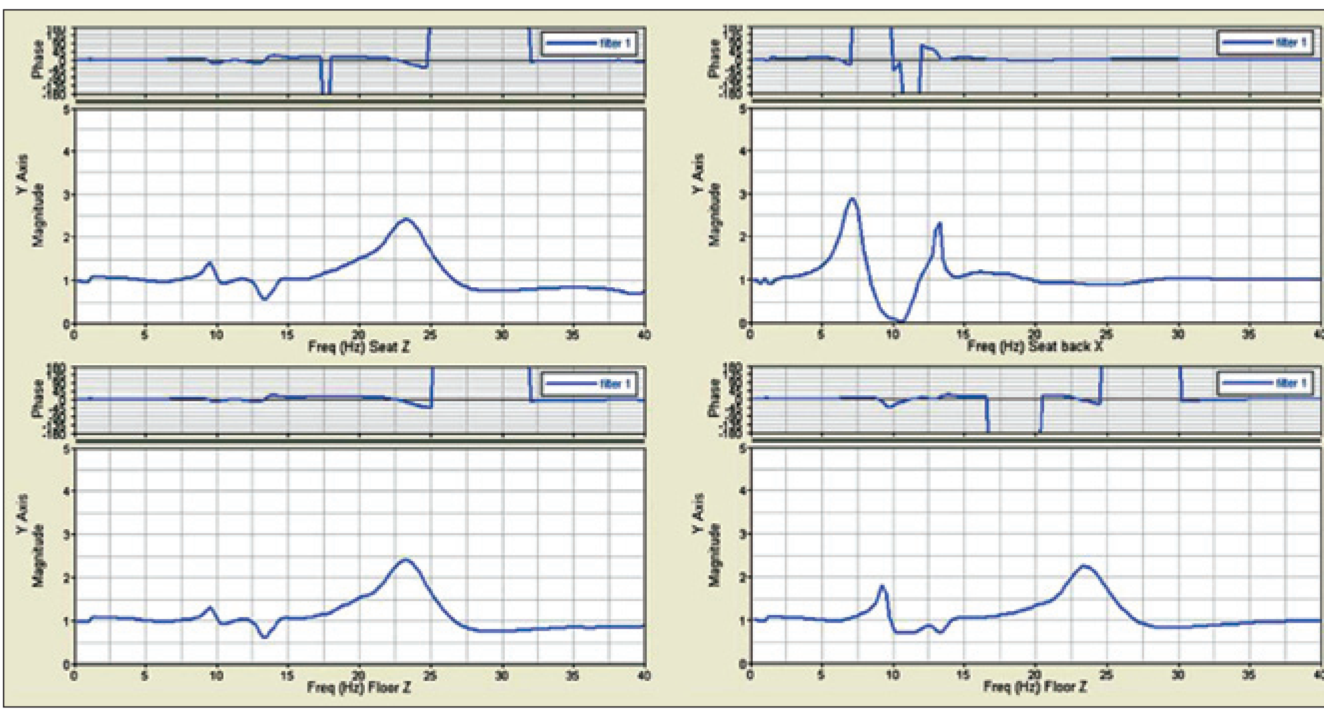

Figure 5. Change filters.

2 kHz. It was also necessary to include the magnitude and phase information for the higher frequency data as the relative phase shifts became significant. So the attribute defined approaches were already divergent.

The power-train NVH model could use a database of test measurements, modified (filtered) test data and CAE predictions. In this case, for the initial model, the vehicle sound and vibration experienced by the vehicle occupants from an existing vehicle were decomposed into the individual harmonic (e.g., NVH from engine mounts, intake and exhaust systems) and nonharmonic (e.g., road noise and wind noise) sound objects. In the NVH simulation, the component sounds were accurately mixed in real time to take into account the vehicle speed and throttle position during the driving simulation. The decomposition approach enabled the data measured on the original test vehicle to be correlated directly with the simulated sound. The method could then be extended to enable evaluations of proposed modifications; e.g., alternative engines measured in a different vehicle, or for vehicle and component target setting, or as in this case, for developing power-train mounting strategy. In the case of power-train mounting system development, other usually dominant paths such as intake and exhaust noise were unchanged so that the influence of the power-train mount was examined in isolation but with the masking effect of the other noise sources.

Specifically for this work, the structure-borne noise and vibration from the power-train mounts was decomposed into mount displacements, mount stiffnesses and noise transfer functions measured from vehicle and laboratory tests. Therefore by manipulation of the mount stiffness curves, the influence of the power-train mounting system on the vehicle's NVH could be assessed. The fixed displacement dynamic stiffness curve selected for ride comfort was a good way to create normalized data across the frequency range. However, this was not so good for the power-train NVH requirement, which was required up to 2 kHz. Clearly, the mount

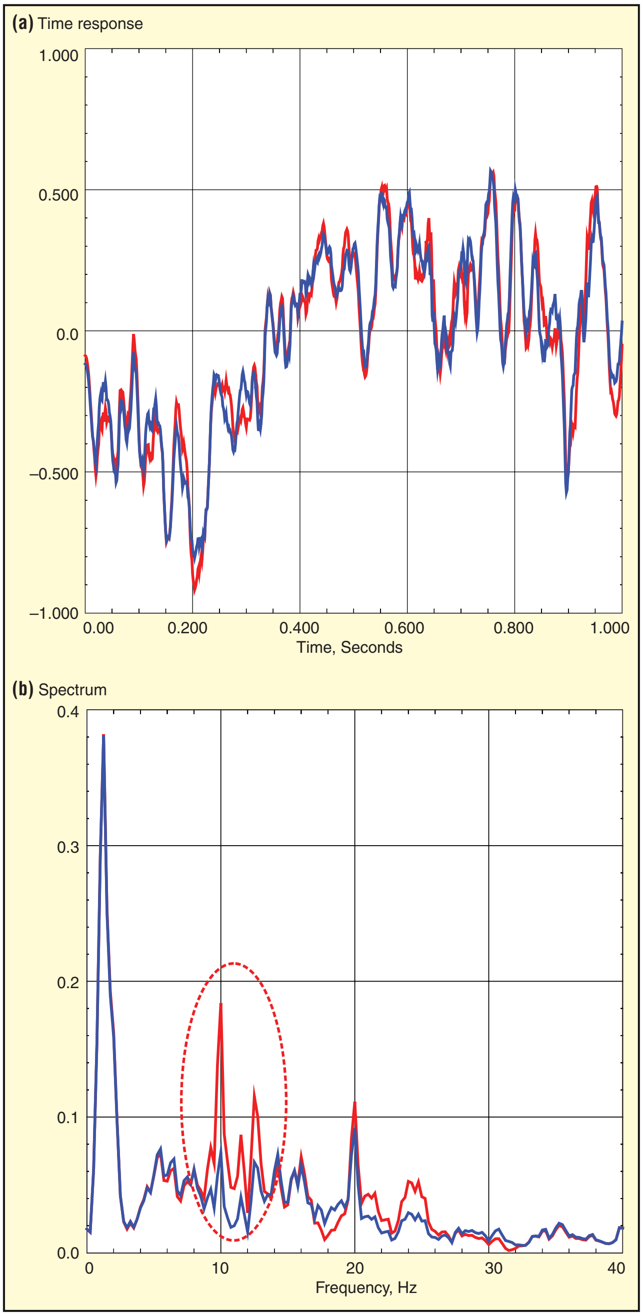

Figure 6. Time response vs. spectral data: (a) very little difference in level between unmodified and modified time histories; (b) clear difference in level between unmodified and modified spectral data (circled).

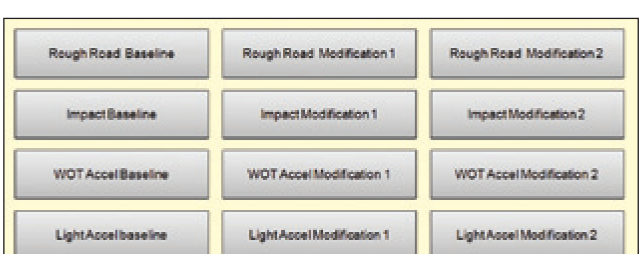

Figure 9. Example of user interface.

would not experience the same displacement at 100 Hz as it did at 1000 Hz, so the approach was changed to fixed acceleration. This allowed the displacement to vary according to the frequency at which the excitation occurred – a more realistic test for high-frequency performance.

As discussed previously, it was not necessary to create a structural model for power-train NVH, which examined the dynamic characteristics of the subframe mounts etc. It was only necessary

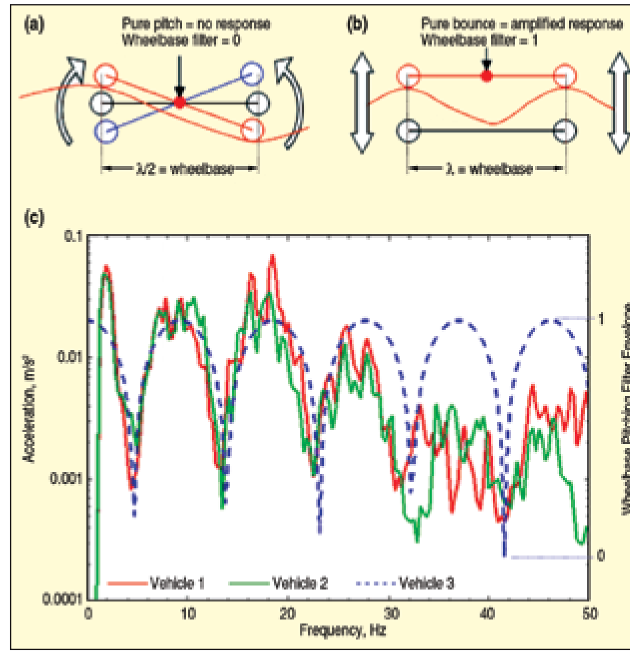

Figure 7. Wheelbase pitch filter effect.

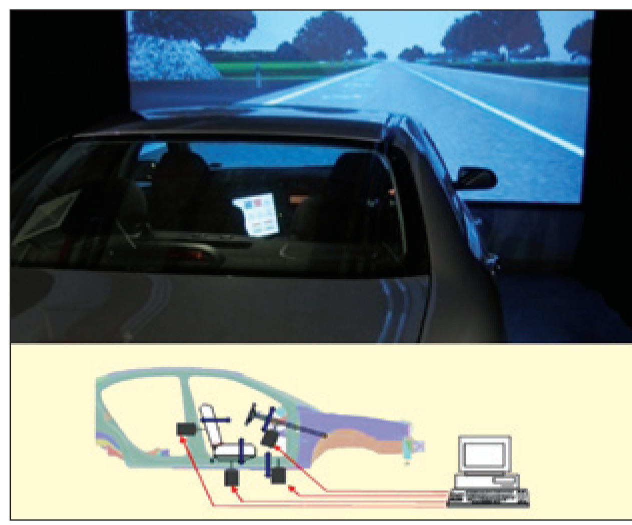

Figure 8. Full vehicle simulator; sound and vibration interactivity and context.

to construct a model that could characterize the source ranking of acoustic sources on the car such as intake, exhaust, power-train mounting system and other sources, for the purposes of this exercise. In the case of acoustic simulations for other purposes, it may be necessary to split "other sources" into road and wind noise, for example, which required a modified approach in data acquisition.

Power-Train NVH Change Filter Preparation. Almost identically to the ride comfort model, it was necessary to carry out change filter calculations to modify both the acoustic and vibration signatures for the various tactile points and ear positions. The change filters used were based on "what-if" studies to gain understanding and production-feasible mount rate curve modifications. However, the increased frequency content of power-train NVH signatures required us to calculate power-train change filters over a higher bandwidth.

Subjective Evaluation

Following all of the stages necessary to prepare the data for all of the tactile point vibration and acoustic signatures for all of the mount specification variants, we could carry out the subjective evaluations. This was carried out in a full vehicle with shakers attached to the floor pan, seat rail and steering column in various axes with high-fidelity, super-aural headphones for the driver[4] (see Figure 8).

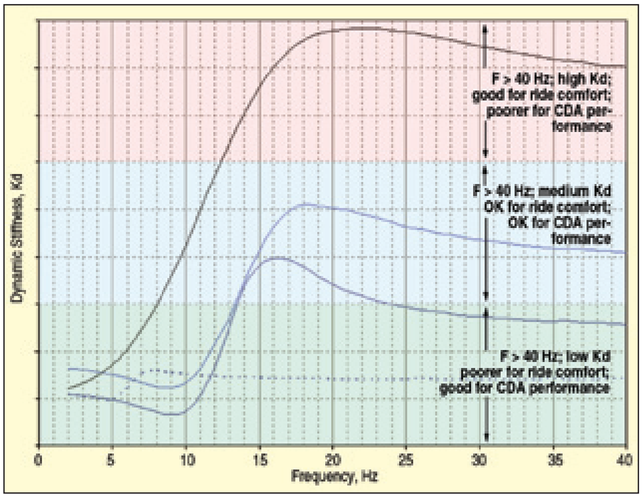

Figure 10. Dynamic stiffness (K_d) for three mounting systems; medium area (blue) is most acceptable attribute balance.

There were visual stimuli in the form of a large video screen to add to the realistic nature of the driving experience. Details such as vehicle gauges were also addressed so that the road/engine speed and even fuel/temperature gauges were active to enhance the driving experience. Once inside the car, the assessor was faced with a computer touch screen on the passenger seat that would display a number of buttons (shown in Figure 9) describing the power-train mount specification and the evaluation conditions (fixed road speed, speed variant schedule etc.).

The vehicle occupants could then select a power-train mount variant on the touch screen, evaluate a selected maneuver (steady-state shake, impact shake, wide-open throttle acceleration, CDA part-load acceleration etc.) and then record their subjective score and notes. Alternatively, the system could be set to record subjective rankings or scores – as you like.

It gave the team the opportunity to examine power-train mount effects on vehicle performance over a range of operating conditions without ever having produced a test part.

Design Changes

Following the subjective evaluations, it was possible to then distill the information gained from the engineer or even a range of assessors and understand what design changes could be made. This was possible with parts that had never been manufactured, and it was the task of the engineering team to determine the best component given the project targets.

Normally it is necessary to balance the attributes of ride comfort and power-train NVH for power-train mounting systems owing to their divergent requirements for dynamic stiffness. Normally, a mount performs well for ride comfort if it has high dynamic stiffness at 10-20 Hz and performs well for power-train NVH if the dynamic stiffness is low, from 40-1000 Hz. This is the perfect attribute balance. However, a subset of the performance requirements for power-train NVH on many of Bentley's latest engines is variable displacement technology.

This fuel saving technology transforms a V8 engine, for example, into a V4 engine by cutting the fuel supply and disabling the valves for the deactivated cylinders. Because the engine was now transformed from a V8 to V4, this changes the predominant engine order content from fourth order to second order. The engine speed range over which variable displacement technology is operable causes a firing pulse from ~37-117 Hz.[5] Additionally, the second engine order forcing is highest at lower engine speeds, so the 37-45 Hz region is particularly important for power-train mount isolation (so, dynamic stiffness is low). Since this is at the upper edge of the region where we manipulate the design to be high dynamic stiffness for ride comfort purposes, there was a very difficult balancing act to perform. We wanted the mount to be stiff below 40 Hz, and we wanted the mount to be soft above 40 Hz. Managing this transition in dynamic stiffness (schematically described in Figure 10) was found to be critical for successful attribute balancing of the power-train mounting system.

Conclusions

The simulation-based approach of reproducing acoustic and vibration performance in a vehicle was a pragmatic approach to obtaining power-unit mount specification. The correlation achieved between the full vehicle simulator based approach and track based measurements has been continually proven over several projects. This means that high confidence could be obtained before reaching the stage of manufacturing physical prototypes of power-unit mounts. This has had the effect of reducing the amount of physical testing per vehicle program, which saved product development time and cost. It also allowed Bentley engineers to develop with confidence to help provide world-class levels of ride comfort and refinement in their products.

The approach is now a staple part of the vehicle development process and delivers confident predictions that can be physically evaluated by all concerned which helps unify the approaches of the engineering development team and the management team.

References

1. D. Fothergill, F. Syred, and P. Kennings, "The Use of System Modelling Techniques to Filter Test Measurements and Drive a Physical Vehicle NVH Simulator," Seventh Altair CAE Tech.Conference, 2011.
2. J. Hargreaves, J. Richards, F. Syred, and R. Johnson, "Interaction of Vibration and Sound Stimuli in Transient Impact Events," EIS Conference, 2009.
3. "Magneti Marelli's Simulator for Development and marketing," Brüel & Kjær, http://www.bksv.com/newsevents/waves/waves_2012_issue8/magneti_marelli.aspx, 2012.
4. R. Williams, F. Henderson, F., M. Allman-Ward, G. Dunne, M. Cockrill, and S. Adlard, "Using an Interactive NVH Simulator for Target Setting and Concept Evaluation in a New Vehicle Programme," SAE 2005-01-2479, 2005.
5. Ian McDevitt, Uday Senapati, and Aaron Hankinson, "Vehicle Refinement Challenges for a Large Displacement Engine with Cylinder Deactivation Capability," SAE Int'l, 2011.

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