inter.noise 2014
MELBOURNE AUSTRALIA
16-19 NOVEMBER

Developing Powertrain Mounting Systems in the Virtual Engineering World Using a Full Vehicle NVH Simulator

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ABSTRACT

At Bentley Motors, the development approach for powertrain mounts has ramped up in recent years, from Hardware based test comparisons in the early years through to our CAE based development with "Physical Simulation". This latest technique, known as the Full Vehicle NVH Simulator (FVS), allows the engineering team to physically evaluate a laboratory based vehicle with high correlation to the vehicle on road – this creates a realistic and useable link between objective data and subjective feel. The tool is powerful in that it allows the team to achieve the required attribute balance between ride comfort and powertrain NVH. Advances made towards Bentley's carbon footprint, like variable displacement engines (selectively reducing the number of active cylinders during operation), adds to the challenges of this attribute balance. This change results in lower firing orders at which mount isolation is required, making spectral tuning of the mount stiffness more challenging owing to the lower frequency nature of this operating range. The physical simulation technique which has been used for powertrain mounts development within Bentley has become a business advantage owing to the reduced number of prototypes required for testing and, therefore, reduced product development costs with accurate predictions ahead of design freeze.

Keywords:
Powertrain Mounting, Full Vehicle Simulator, Attribute Balance.
I-INCE Classification of Subjects Number(s): 13.2.1 Automobiles
Conference Session: D2 Motor vehicle noise vibration and harshness (NVH)

1. INTRODUCTION

To support Bentley Motor's objectives, this technique was devised between Bentley Motors Whole Vehicle Physics department, Sound & Vibration Technology Ltd and DJ Fothergill Consulting.

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

The Full Vehicle NVH Simulator (FVS) technique, described in this paper, supports the delivery of luxury motor cars which offer their drivers class leading levels of ride comfort. This was largely achieved by tuning the powertrain mounting system, on which this paper will focus. Additionally, the body structural performance was developed to achieve good modal separation of the body structure from powertrain rigid body modes and the compliantly mounted subsystems, such as subframes and suspension levers etc., were modelled in order to provide the correct context for our powertrain mounting systems evaluations. In the early stages of use of this approach, the technique was augmented by vehicle testing to rank alternative specifications and provide backward correlation with the simulation process. Once the technique had been established, and confidence in correlation was accumulated, it was possible to optimize the compliantly mounted subsystem shake (including the Powertrain Mounting System) with much less physical testing than before. The system model

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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 makes use of both physical test and CAE, has been derived to enable engineers to subjectively assess predicted specification changes without the manufacture of prototype parts.

The approach has enabled Bentley Motors to save development time and save cost by reducing the number of hardware test iterations for powertrain mounting system development. In this way, the ideal specification can be set during the virtual investigations as there is no requirement to build prototype parts and a desired mounting specification can be derived from the technique.

1.1 Process Overview

Regardless of whether we're examining either side of the attribute balance seesaw {on one side we have ride comfort and on the other side we have powertrain NVH – akin to reference (7)}, similar outline processes can be followed – details of exceptions are captured:

1.1.1 Base Vehicle Measurement

- Measurement of baseline vehicle for specific ride comfort and powertrain NVH maneuvers using tactile points which correlate to FVS excitation points (this includes mount displacements, mount stiffness's, noise and vibration transfer functions to modify response locations for powertrain NVH)

1.1.2 CAE Model Preparation (Ride Comfort Only)

- Kinematics & Compliance test data used to define parameters initially
- Compliant element modelling including dynamic characteristics for hydraulic mounts
- Flexible body shell preparation
- Correlation back to base vehicle

1.1.3 FVS Preparation

- Base vehicle, run from baseline data
- Preparation of modified time data using change filter technique (this approach differs between ride comfort and powertrain NVH)
- Comparison of modified time data including correlation loop back to base vehicle measurement (only possible where modified specifications are measured for correlation purposes)

1.1.4 Making Decisions

- Possibility to rank specifications by technical performance without manufacture
- Balancing of attributes is clear between ride comfort and powertrain NVH

2. THE RIDE COMFORT FULL VEHICLE SIMULATOR (FVS) PROCESS

The use of the process, once completed, is outlined in the following figure and outlined in the below text.

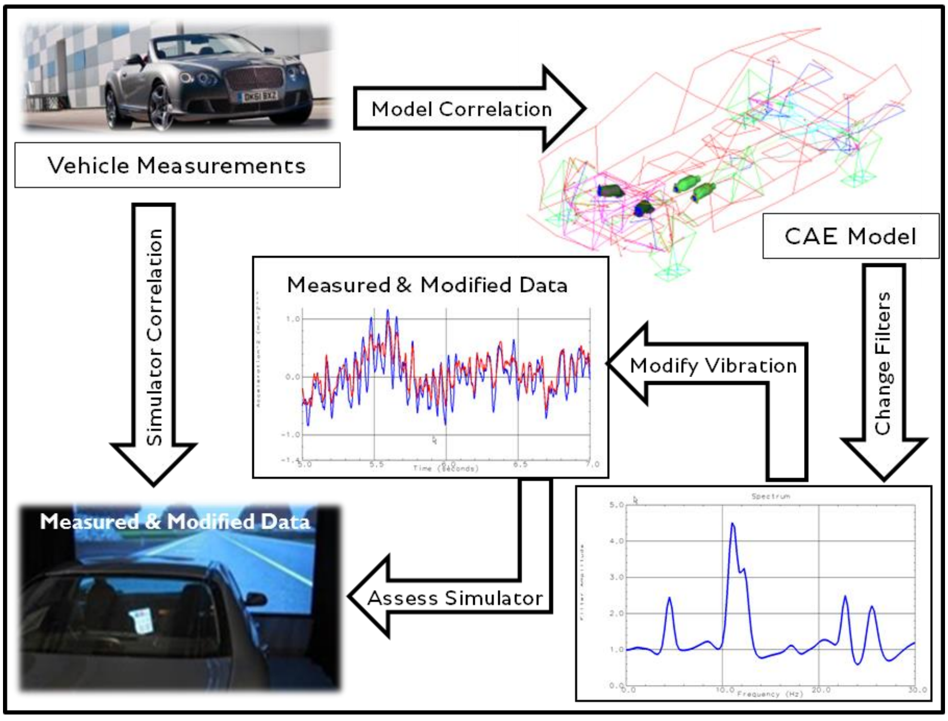

Figure 1 - The Full Vehicle Simulator Process

2.1 Ride Comfort Baseline Vehicle Measurements

The outline process relies, firstly, on the baseline vehicle measurements to provide both a reference for changes to be made and also to provide a correlation reference for the CAE model. The process was reported in references (1, 2 & 3).

The tactile points used to replicate the ride comfort performance in the FVS, plus the acoustic signals used to activate the headphones, are all measured. These measurements are taken upon a range of specified test surfaces to provide ride comfort responses.

2.2 Ride Comfort CAE Model Preparation

Secondly the CAE model is created, which allows a correlation loop back to the reference vehicle but also allows the modelling of desired changes to the vehicle systems. This is the heart of the process for ride comfort and necessitates the CAE model creation correlating to the baseline vehicle. The CAE model is created using the same parameters that exist on the baseline vehicle. This is normally quite straight forward for an existing vehicle as the compliant element data, moment of inertia data, center of gravity data and other co-ordinates are easily available from the engineering design functions – not the case, of course, for a car under development.

When the desired changes to vehicle systems are made, a set of change filters is created which are used to change the selected tactile point response vibration signatures. Both the baseline signals and modified signals can then be played into the physical simulator for subjective evaluation. The baseline signals do not include the change filter part of the process and bypass this straight to simulator assessment. The change filters, to be applied to the baseline time signals to create the modified time signals, are created by dividing the CAE produced modified spectra by the baseline spectra. These change filters are then used to modify the baseline signal to give the resultant modified time response.

These responses can be plotted, in advance of assessing in the FVS, to check that the tactile point responses are behaving as expected. This, of course, needs to be done selectively as it can influence the subjective assessor’s perception of a change.

Often, the dynamic characteristics of the compliant elements for the base vehicle are obtained via experiment. This is done at various amplitudes of excitation with a representative pre-load (as it would be in the car). The dynamic characteristics for representative fixed amplitudes are selected given the amplitudes of bush excitation during each ride comfort maneuver. This stage obviously requires a degree of study as the fixed amplitude at one frequency will differ from that at another frequency. A pragmatic data selection approach is needed in order to best represent the amplitudes experienced by each mounting system during each ride comfort test cycle.

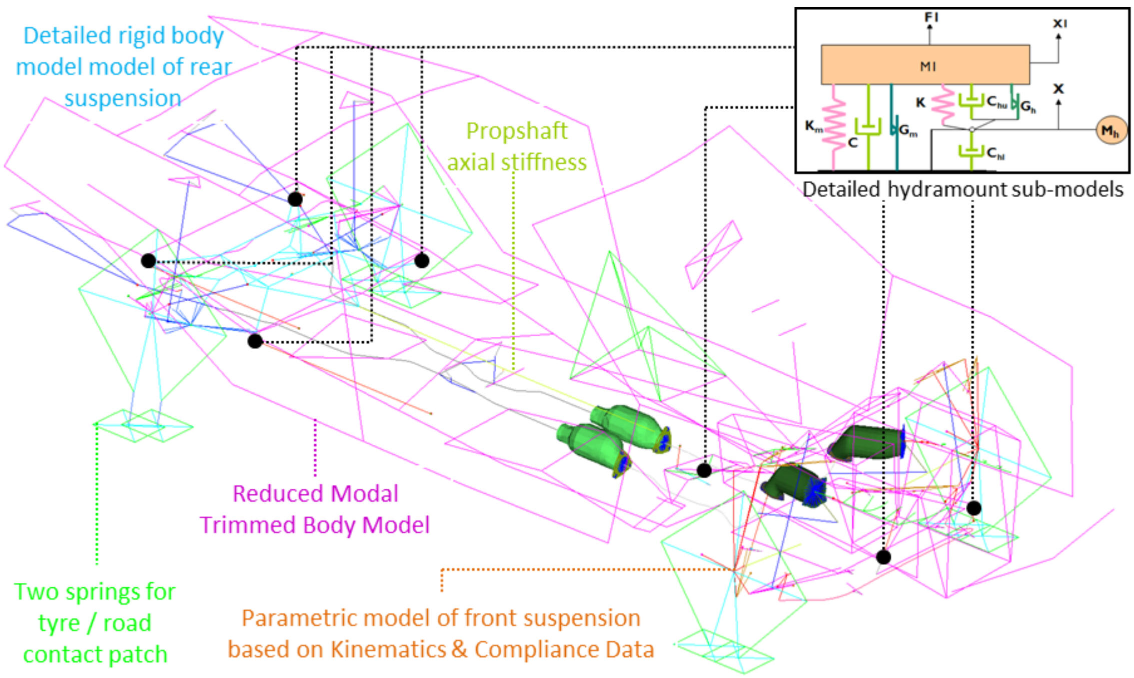

Figure 2 – Modelling Example

The flexible body CAE data preparation takes the full trimmed body shell CAE result and will take a reduced format in terms of geometric locations. The spatial model will be extracted from the trimmed CAE result and only specific points, including those which closely relate to the tactile points, are used in the model. This helps to reduce the system requirements for any analyses.

In the worst case (cabriolet car), the body structural performance is key to the ride comfort performance of the car owing to the spectral positioning of body torsion and body bending modes. It is very important to design to achieve separation between the body's torsion / bending Eigen-frequencies and the compliantly mounted sub-system modes. This is of less concern on limousine cars.

This is normally easier to achieve with the powertrain mounting system than with subframe and lever sub-systems owing to the spectral positioning of these rigid body responses.

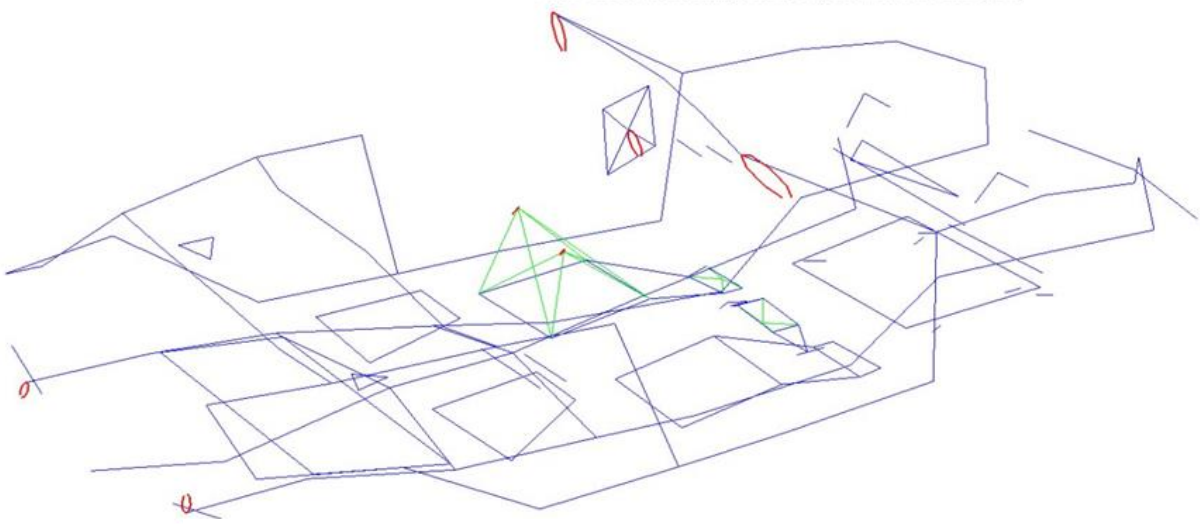

Figure 3 – Body Structure Torsional Response Example

The replication of the compliant behavior of the car can be done to more or less detail as is required. So, either entire ¼ suspension systems can be modelled parametrically; reproducing behaviors measured on a kinematics and compliance test rig, or the individual parts (levers, hubs, bushes etc.) can be modelled in detail. That is, mathematical constraints are 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. A decision making process is adopted to determine how much detail is included in the model. A degree of pragmatism is used to decide upon the value of each included compliant element.

It is possible, of course, to model only the powertrain mounts, for example. However, when comparing this basic model to one which includes all compliance elements in the subframes, links et al plus the dynamic performance of the body structure it becomes clear that evaluating the latter model will yield a subjective response which is more aligned with the intended result of the whole vehicle.

The powertrain mounting system dynamic stiffness properties are selected based upon the kind of road surface being modelled. For example, a moderate surface will be modelled using a powertrain mounting system with a ±0.1mm fixed displacement curve whereas a rough surface or impact events might use a ±0.5mm fixed displacement curve, as related to reference (5).

It's vitally important to retain focus on the divergent attributes of ride comfort and powertrain NVH in this dynamic stiffness property selection. We at Bentley Motors are interested in retaining high dynamic stiffness in the low frequency region (in addition to the damping characteristics afforded by the hydraulic internals of such mounts) to offer good ride comfort whilst retaining low dynamic stiffness, for attenuation, in the high frequency region which predominantly controls powertrain NVH.

In the case of engines with variable displacement technology, a further complication is added into the mix – the region in between the low frequency (Ride Comfort critical) and high frequency (Powertrain NVH critical) is responsible for attenuation of lower order vibration due to half of the engine's cylinders being de-activated. This forces a requirement for the powertrain mounting system to have rapidly decreasing dynamic stiffness between the low and high frequency ranges as shown in figure 5 (highlighted as the 'tuning dichotomy') .

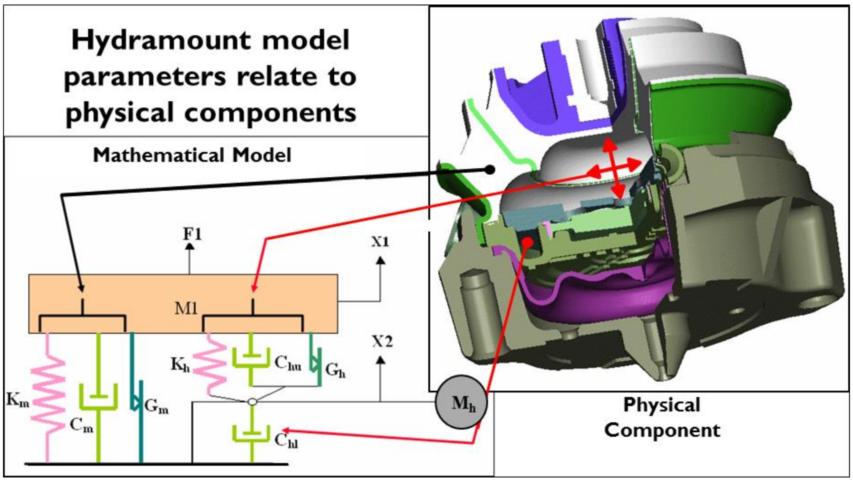

Figure 4 – Detailed Hydramount Model

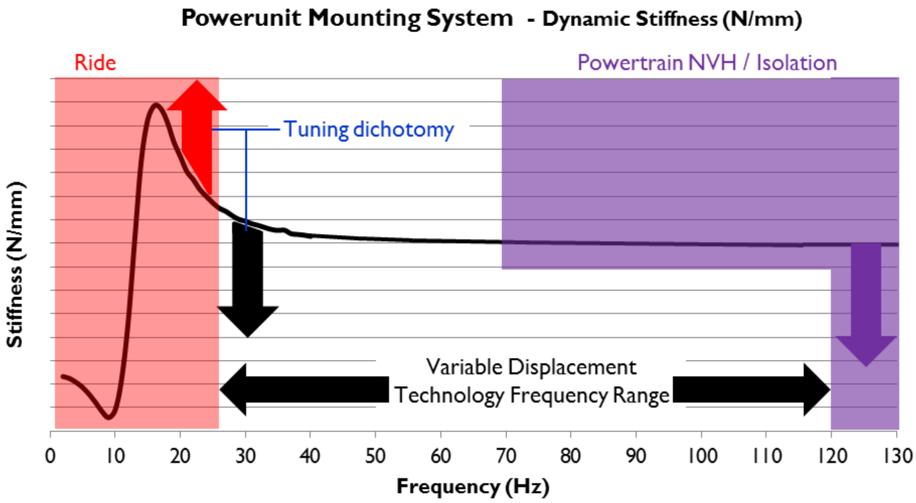

Figure 5 - Attribute critical regions for Powertrain Mounting

Given this tuning dichotomy, the Full Vehicle Simulator process offers an excellent tool for striking this balance by driving the specifications back-to-back and converging on a mount design before parts have even been manufactured.

The mathematical model for the hydramount is carefully modified to match the mount performance [both stiffness & out-of-phase stiffness (OOP)] using the sub-model solver, as seen below. A similar approach is used for chassis mounts.

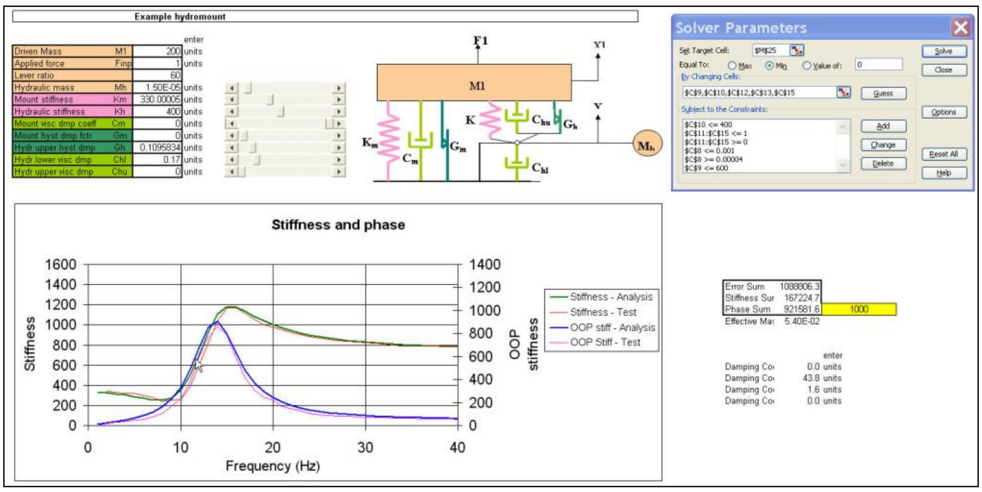

Figure 6 – Hydramount Sub-model Solver

Finally, one of the interesting characteristics of the vehicle model is the way in which the road / tire interaction is modelled. The road surface waves are applied to the ground at the front and back of each tire contact patch. Scalar spring elements are then placed at these locations representing the tire stiffness in compression, and (fore and aft) shear.

A mathematically rigorous representation of the (frequency dependent) phasing of the ground waves between contact patch input points results in the wheel being forced in both bounce and roll.

This simplification works 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 vertically at only one point per tire contact patch, there would have been no meaningful rolling or fore and aft excitation of the wheel.

2.3 Ride Comfort FVS Preparation

The next stage of the process, once the CAE model has been created and correlation is achieved back to the vehicle measurements, is the preparation of change filters. These change filters are calculated from the modified specification and baseline specification.

The filter is calculated as below;

$$\text{Filter} = \frac{(\ddot{a}\ (\omega)\ \text{Modified Specification / excitation force})}{(\ddot{a}\ (\omega)\ \text{Baseline Specification / excitation force})} \qquad (1)$$

where ä (ω) is a matrix of acceleration spectra.

The process that is adopted is quite simple and allows the engineer to apply to each tactile point in turn. As the excitation force is cancelled out, due to both forcing functions being identical, we are left with a filter which is dimensionless and independent of the forcing.

These filters, for each tactile point, are then applied to the model to produce a ‘modified’ set of tactile point responses. This enables the engineer to evaluate the effect of the specification change in the vehicle by comparing the ‘baseline’ and ‘modified’ specifications. The effect of the modification is then clear when carrying out back-to-back evaluations, without the time lag normally associated with changing physical components on the hardware vehicle – 2 hours can be lost when changing engine mounts for example, by which time many assessors have forgotten how the original mount performed.

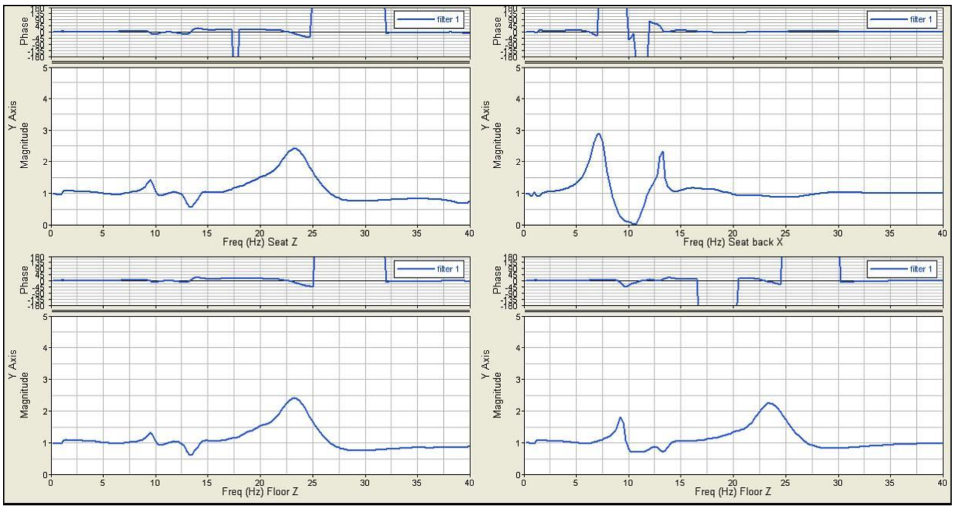

Figure 7 – Change Filters

Because the ride shake measurements are conducted during constant speed test runs, the powertrain noise contribution is low. This means that the same powertrain noise sound track can be used for ride comfort evaluations, which eliminates the psychological effects upon vibration perception. Conversely, some specification comparison runs which had the same powertrain noise sound track, have actually been criticized for acoustic differences. This highlights the interdependency of acoustic and vibration subjective assessments and how the vibration differences can affect one's acoustic perception.

On output from the CAE deck, it's important to note that the tactile point data can often look quite similar if examining in the time domain. This data is often best examined in the spectral domain as the next two figures show, with dimensionless values on y-axes owing to units cancelling.

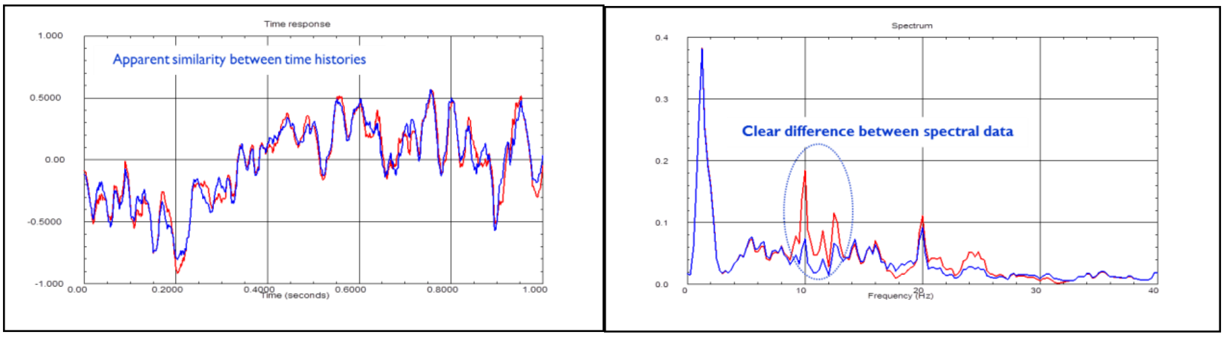

Figure 8 - Typical filter applied to vibration time history (on left) showing Modified and Baseline specifications. Spectral vibration data (on right) shows clearer differences.

Finally, as discussed in 2.2, we need to consider the influence of the wheelbase pitching filter. As the front and rear wheels pass across the road surface, it is necessary to consider the influence of the car's pitch behavior. This is 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 will 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 the following figures.

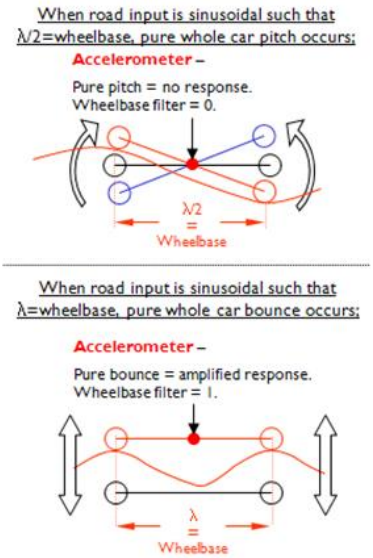

Figure 9a - Wheelbase pitch filter effect

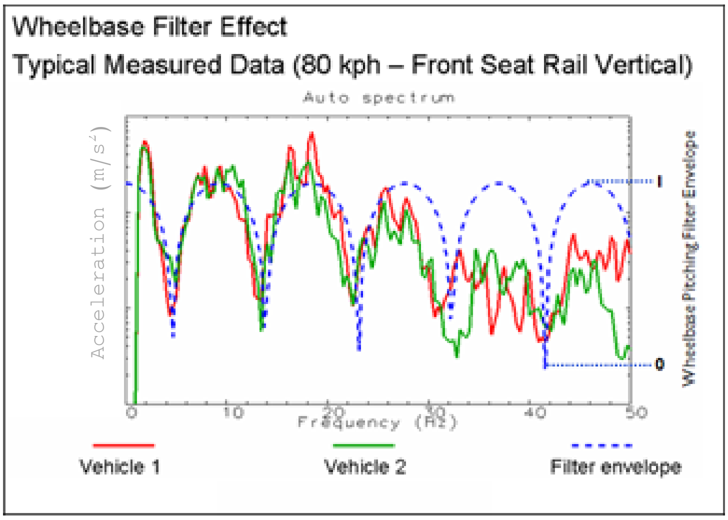

Figure 9b - Wheelbase pitch filter effect upon data

3. THE POWERTRAIN NVH FULL VEHICLE SIMULATOR (FVS) PROCESS

The FVS process for powertrain NVH is separate to ride comfort but can be re-aligned in the subjective assessment stage such that the engineer could evaluate both attributes in the car sequentially – just as in a real car. 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.

3.1 Powertrain NVH Baseline 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 (acoustic simulator similar to FVS but on a desktop computer using headphones only). This allowed the engineer to evaluate the powertrain 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 can now be assessed in the same way as the ride comfort evaluations.

3.2 Powertrain NVH CAE Model Preparation

The approach for the powertrain NVH CAE model worked in parallel with the ride comfort CAE model. The key difference to the ride comfort approach lay with the frequency range of interest – the ride comfort analyses were usually conducted from 0-40Hz with some linear extrapolation up to 200Hz, whereas powertrain NVH analyses required up to 1kHz and beyond. In fact, the dynamic rating data that was used was usable to 2kHz. 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 powertrain 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 non-harmonic (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 the development of the powertrain mounting strategy. In the case of powertrain mounting system development other, usually dominant paths such as intake and exhaust noise, were unchanged so that the influence of the powertrain 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 powertrain mounts was decomposed into mount displacements, mount stiffness's and noise / vibration transfer functions measured from vehicle and laboratory tests as described in reference (4). Thus, by manipulation of the mount stiffness curves the influence of the powertrain 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 powertrain NVH requirement which was measured up to 2kHz. Clearly, the mount would not experience the same displacement at 100Hz as it did at 1000Hz so the dynamic stiffness measurement 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 the high frequency performance. As discussed earlier, it was not necessary to create a structural model for powertrain NVH which examined the dynamic characteristics of the subframe mounts et al. It was only necessary to construct a model which could characterize the source ranking of acoustic sources on the car such as intake, exhaust, powertrain 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 noise and wind noise, which required a modified approach in the data acquisition as in reference (6).

3.3 Powertrain NVH FVS 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 powertrain NVH signatures required us to calculate change filters over a higher bandwidth.

4. DECISION MAKING PROCESS

4.1 Subjective Assessment

Following all of the stages necessary to prepare the data for all of the tactile point vibration and acoustic signatures for all of the specification variants, we can carry out the subjective evaluations. This is carried out in a full vehicle with shakers attached to the floor pan, seat cushion, seat back and steering column in various axes with high fidelity supra-aural headphones for the driver. There are visual stimuli in the form of a large movie screen to add to the realistic nature of the driving experience.

Details such as the vehicle gauges have also been addressed such that the road speed, engine speed, even fuel and temperature gauges are correct to enhance the driving experience.

Once inside the car, the assessor is faced with a computer touch screen which displays a number of buttons describing design changes and evaluation conditions (fixed road speed, speed variance et al).

Figure 10 - Full Vehicle Simulator Cockpit & Visuals with touchscreen (left inset) & shaker schematic (right inset)

The vehicle occupant can then select a specification variant on the touch screen, evaluate a selected maneuver (steady state shake, impact shake, free driving etc.) and then record their subjective score and notes. Alternatively, the system can be set to record subjective rankings or scores – as you like.

It gives the engineer the opportunity to examine powertrain mounting system changes upon the vehicle performance over a range of operating conditions without ever having produced a test part.

4.2 Evaluate design changes without manufacture to arrive at optimum attribute balance

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

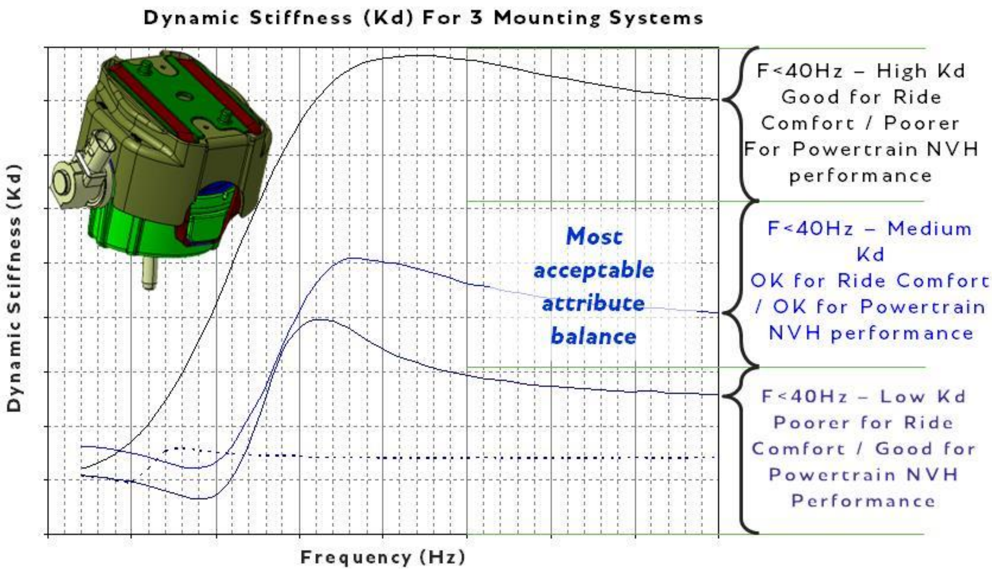

Figure 11: Dynamic Stiffness Attribute Balancing

Normally it is necessary to balance the attributes of ride comfort and powertrain NVH for powertrain mounting systems owing to their divergent requirements for dynamic stiffness. Usually, a mount performs well for ride comfort if it has high dynamic stiffness at 10 – 20Hz and performs well for powertrain NVH if the dynamic stiffness is low from 40 – 1000Hz. This is the perfect attribute balance and is what Bentley strives to achieve. However, a subset of the performance requirements for powertrain 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 is now transformed from a V8 to V4, this changes the predominant engine order content from 4th order to 2nd order. The engine speed range over which variable displacement technology is operable in this example is ~1100 – 3500 RPM. This causes a 2nd order excitation during variable displacement technology operation between ~37 and 117Hz. Additionally, the 2nd engine order forcing is highest at lower engine speeds so the 37 – 45Hz region is particularly important for powertrain mount isolation (so, dynamic stiffness is low). As this is at the upper edge of the region where we manipulate the design to be high dynamic stiffness for ride comfort purposes, there is a very difficult balancing act to perform – we want the mount stiff below 40Hz and we want the mount soft above 40Hz. Managing this transition in dynamic stiffness, as shown in figure 5, is found to be critical for successful attribute balancing of the mounting system.

Finally, the body structural performance content can be modified to confirm the influence of the mode placement upon ride comfort. Placement of the first mode needs to be well separated from the Powertrain rigid body modes. This tool can work in both ways; it can be used to determine the minimum torsion mode acceptable with the given rigid body modes. Alternatively, it can be used to push the design of the body structure by identifying the advantage of improving the torsion mode Eigen-frequency to separate it from these powertrain rigid body modes. This approach can be used to highlight the placement of other body modes such as vertical U bending (1^{st} bending) and vertical Z bending (2^{nd} bending). These modes are normally at a higher frequency than the torsion mode but can also influence the ride comfort behavior given the spatial maxima of body response locations.

5. CONCLUSIONS

The simulation based approach of reproducing acoustic and vibration performance in a vehicle was a pragmatic approach to obtaining powertrain 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 powertrain 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 work with confidence to develop 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 engineering teams involved in development of the car and to understand the performance of their components in a whole vehicle context.

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