Investigation into Road Rumble in a Light Utility Vehicle

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Vehicle NVH is now a more important component of the vehicle design process than ever. Road noise is one of the key criteria used by potential buyers (albeit subconsciously) to choose what they perceive as the most refined vehicle. LUVs are especially sensitive to low frequency booming noise due to the fundamental acoustic mode that exists in the vehicle cabin at a higher frequency than in sedan type vehicles. An investigation into this booming noise in an LUV at low speed is documented. The noise is identified and quantified after which the source of the noise in the vehicle cabin is identified using NVH techniques such as AMA, EMA and TPA. The cabin's fundamental acoustic mode was at 100 Hz. Finally the source of the vibrations in the vehicle leading to the booming noise in the cabin is identified in the drive shafts, along with its transfer path to the cabin. Solutions for the specific vehicle's booming noise are proposed, two of which are tested with some success.

ACRONYMS

AMA	Acoustic Modal Analysis
dB	Linear decibel
dB(A)	A-weighted decibel
EMA	Experimental Modal Analysis
FRF	Frequency Response Function
LUV	Light Utility Vehicle (a "bakkie")
NVH	Noise, Vibration and Harshness
SPL	Sound Pressure Level
SQ	Sound Quality
TPA	Transfer Path Analysis

Introduction

During the development of an LUV a low frequency road rumble was identified by the motor manufacturing company as a NVH area that needed additional attention. A number of solutions were proposed and tested but only the stiffening of the load box in some areas was implemented in production and the vehicle was launched with this issue still unresolved to a large extent.

In South Africa road noise is of particular concern as road surfaces are especially rough. Road noise inside a vehicle's passenger compartment is caused by many sources. Probably most obvious is tyre rumble generated by contact of the tyres with the road surface. Other causes of especially low frequency noise are the vehicle's engine and power train and structural vibration (caused by vibration of the suspension or vehicle body). Road noise is transmitted through the vehicle into the vehicle cabin via two distinct paths: through the air (airborne noise) and through the vehicle's bodywork (structure-borne noise). These paths are primarily dictated by the frequency of the noise. According to Lilley & Weber (2003) 85 % of the noise entering the vehicle cabin at frequencies lower than 500 Hz enters through structure-borne paths. This changes to approximately 80 % contribution through air-borne paths for frequencies of over 1000 Hz.

Low frequency road rumble is a low frequency "booming" noise generated by the interaction between the vehicle and the road surface. In simpler terms, the wheels and suspension assembly vibrate when the vehicle travels on (especially rough) roads, causing a booming noise in the cabin. It can be both discomforting and irritating for drivers and passengers when travelling in a vehicle (Anon, 2003). As a result, vehicle manufacturers are spending increasing amounts of time improving the SQ experienced by the drivers and passengers.

A vehicle manufacturer's main aim is to sell vehicles at a profit. To accomplish this, prospective buyers must be convinced that their vehicle is superior to the competition's vehicle. SQ and NVH are playing increasingly important roles (albeit sometimes subconsciously) in the buyer's decision making process (van der Auweraer et al., 1997). Consequently, the subjective quality of sounds is a topic of increasing importance in the automotive industry and is becoming a commonly incorporated aspect of the design process.

LUVs have a fundamental noise problem. When noise is generated in a cavity, the acoustic mode with the lowest frequency is placed along the longest path in the cavity. In saloon vehicles, this is generally not a problem as the longest path (from the driver's footwell to the rear window, shown in Figure 1) generates a fundamental mode with its node situated close to the driver's ear. The driver therefore does not hear this fundamental frequency very clearly. In an LUV, however, the longest path stretches from the driver's footwell to the top back corner of the cabin — just behind the driver's head (as seen in Figure 2). The mode's pressure amplitude is therefore at its maximum near the driver's ear. This is the fundamental problem of an LUV with

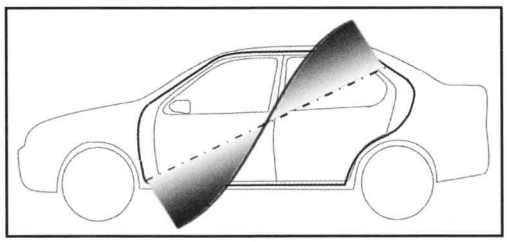

Figure 1: Fundamental acoustic mode of a sedan

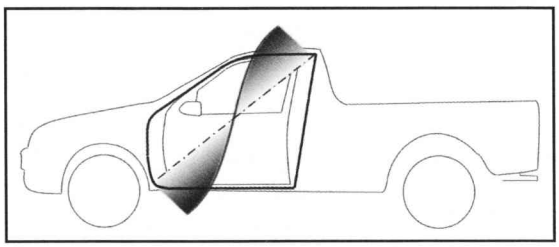

Figure 2: Fundamental acoustic mode of an LUV

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booming noise in the cabin and to reduce its effect, the components in the vehicle that excite this fundamental mode must be identified and modified.

Noise identification and quantification

Run-up test

A run-up test was performed on the vehicle to characterise the booming noise in the cabin and to determine the nature of the vibrations in the vehicle while the booming noise was present. The idling vehicle was free-wheeled from rest down a well-bituminised road of constant gradient. The Sound Pressure Levels were measured at the driver and passenger's ears, while vibrations were recorded at various points on the vehicle, including wheel hubs and body panels. The vehicle was not placed in gear to confirm that the booming noise was in fact induced by contact with the road, and not by the engine's vibrations.

The recorded sound levels at the passenger's inner ear showed a marked resonance at 100 Hz, as well as a "thumb print" shaped area of noise in the region of 85 Hz to 105 Hz between 25 km/h and 40 km/h. Due to the large low frequency content of the signal, it was unclear whether A-weighting would be the correct weighting to apply to the signal as the A-weighting curve would remove these low frequency components. However, after subjective analysis by a panel of listeners, it was decided unanimously that the A-weighted signal represented the "heard" signal the most accurately.

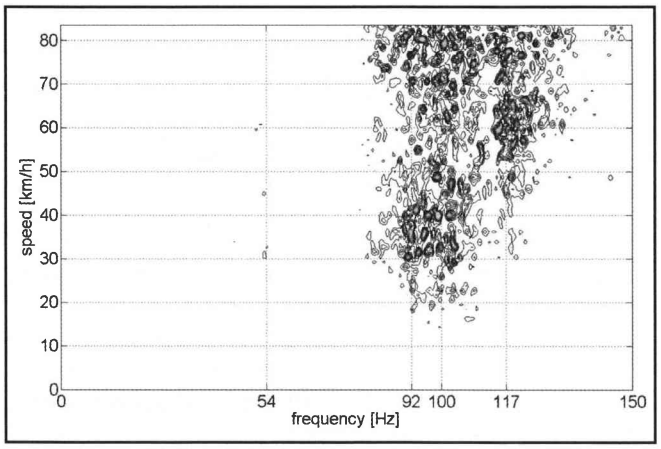

Figure 3: Campbell diagram for noise levels at the passenger's inner ear (A-weighted) (Darker colours are higher noise levels expressed in dB(A))

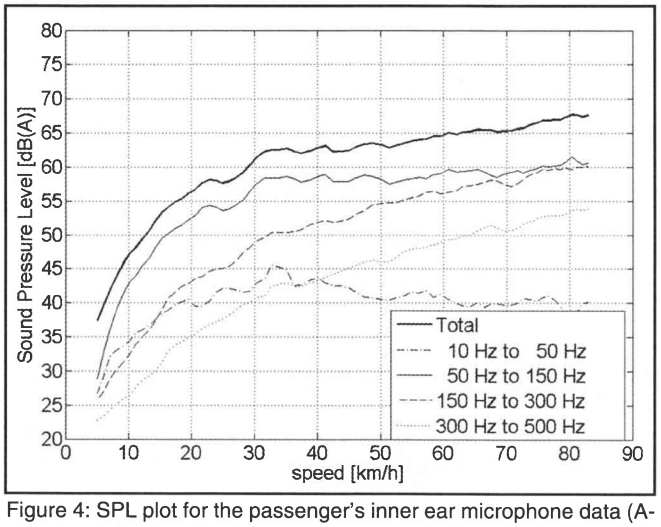

Figure 4: SPL plot for the passenger's inner ear microphone data (A-weighted)

Analysing the frequency content of the noise signals, it was evident that the 50 Hz to 150 Hz frequency band dominated the signal (Figure 4). A sharp rise in the SPL in this frequency band from 25 km/h to 30 km/h was also seen, confirming the 85 Hz to 105 Hz noise to be the LUV's booming noise.

Analysis of the vibrations at the body panels showed some panels to also exhibit the "thumb-print" of excitation in the region specified above. The panel that showed the most vibration in this region was directly behind the cabin seats at the end of the load box. More interesting though were the vibrations measured at the front wheel hubs. The vibrations at the hubs also contained the "thumb-print" of excitation between 25 km/h and 40 km/h (Figure 5).

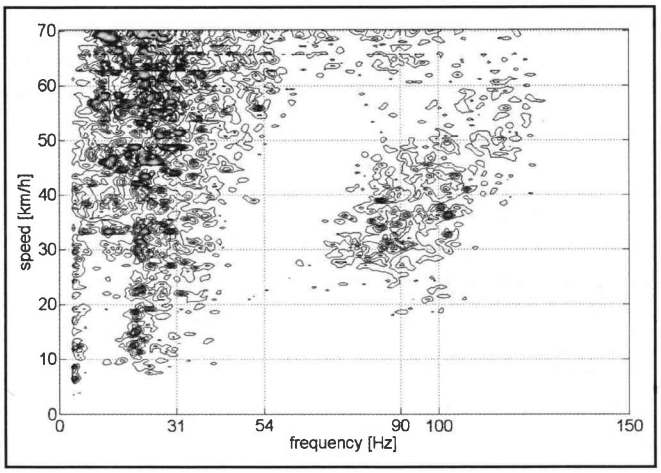

Figure 5: Campbell diagram of the vibration level of the front right wheel hub during the Run-up test (Darker colours are higher noise levels expressed in dB(A))

Acoustic Modal Analysis

The cabin's fundamental mode was determined both analytically and experimentally.

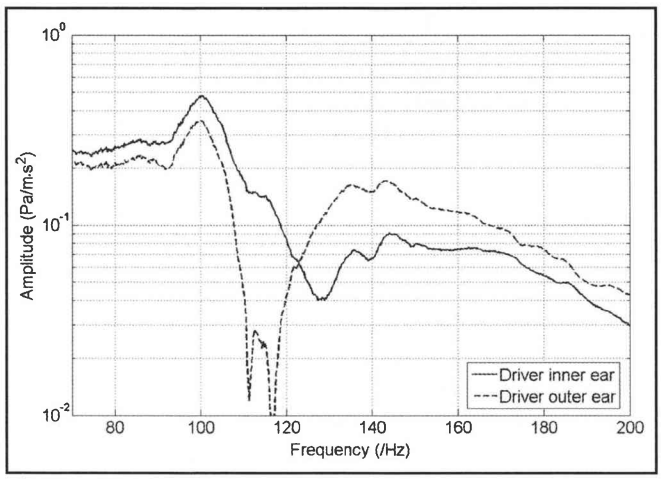

Figure 6: FRF for the driver inner and outer ear microphones during the experimental AMA test

Experimental AMA

The vehicle cabin was instrumented with a binaural recording head on the driver's seat and a microphone at the passenger's inner ear. A binaural head is a device that replicates the head and torso impedance of a human with two microphones at each ear's position. Measurements taken with a binaural head are typically used for sound quality assessments. The cabin was excited

using white noise with a maximum frequency of 500 Hz played by a 10 inch dual-cone speaker located in the driver's foot-well. The sound input into the cabin was measured by attaching an accelerometer to the speaker's inner cone. Coherence levels of almost unity were achieved for frequencies above 60 Hz. Hendricx et al. (1997), among others, also used this speaker-accelerometer setup successfully. The speaker was placed on a soft sponge to minimise direct vibration inputs into the cabin's floor panels.

The resulting FRFs for the driver's inner and outer ears (Figure 6) showed an acoustic resonance at about 100 Hz which was in the same range as the identified booming noise.

Analytical AMA

Analytical AMA was performed using a Finite Element software package (MSC.Nastran 2004) with MSC. Patran as pre- and post processor. The cabin interior was modelled using rigid boundaries and non-absorbent materials. The entire cavity was then filled with a fluid material having air's properties at 25°C and 1 atmosphere (101,3 kPa).

The results of the analytical analysis showed the first mode to be a longitudinal mode, running from the cabin footwells to the top back corner of the cabin (behind the driver's head) at 103 Hz, as seen in Figure 7. The asymmetrical mode shape in the two footwells is due to the asymmetrical dash panel configuration; the driver's dash panel, together with the steering wheel, protrudes much further into the cabin than the passenger's dash panel.

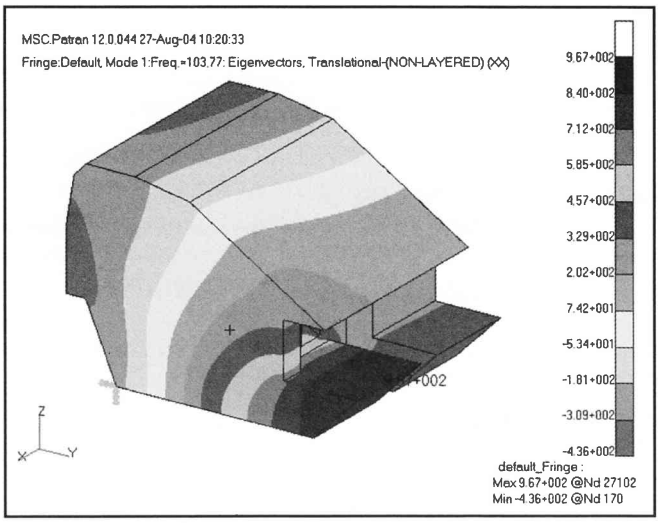

Figure 7: Predicted fundamental mode of the vehicle's cabin using analytical AMA

The analytical modes were very similar to the experimental results, as shown in Table 1.

Frequency [Hz]		
Mode	Experimental	Analytical
1	100.00	103.77
2	112.00	107.75
3	135.00	144.47
4	144.00	153.56

Table 1: Comparison of cabin's experimental and analytical acoustic modes

Experimental Modal Analysis

Since the booming "thumb print" included noise from 85 Hz to 105 Hz, the source of the 85 Hz to 90 Hz noise was required. EMA was performed on the panel directly behind the cabin seats at the end of the load box (named the "Back Panel").

The Back Panel was instrumented with a grid of 6 accelerometers and struck with a modal hammer. A fundamental mode was found at 90 Hz. Using a structural dynamics software package, SD Tools v5.1, the mode was seen to be a "speaker mode" where the panel vibrates such that all points on the panel move in phase with each other. When vibrating in this manner, the panel will cause a large displacement of the air around it, consequently becoming an excellent noise source. The mode was also quite damped (Figure 8) meaning that the panel's displacement from about 80 Hz to 100 Hz would be large.

The Back Panel was therefore identified as a major source in the cabin of noise between 80 Hz and 100 Hz.

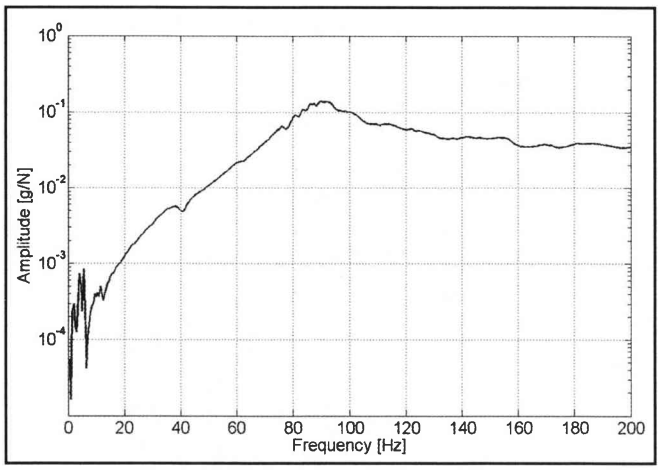

Figure 8: A typical FRF of the Back Panel obtained with an impact test

Transfer Path Analysis

The data from the Run-up test showed the front wheel hubs to vibrate with a similar pattern to the internal booming noise. To determine the extent of the coupling of vibrations at all wheels to the internal cabin noise, a basic TPA was completed.

The wheel hubs were each vibrated in turn by a modal shaker with a random signal and the resulting cabin noise measured at the passenger's inner ear. The resultant FRFs showed that the front wheels contributed more to the internal cabin noise between 85 Hz and 120 Hz, while the right front wheel contributed more in that frequency band than the left front wheel.

Problem identification

The identification of noise in a vehicle cabin can be simplified into finding the "sources" that generate the vibrations in the vehicle, the "amplifiers" that amplify these vibrations, and the "speakers" that convert the vibrations into the identified problem sound. The problem identification stage of the project therefore dealt with finding firstly the speakers in the vehicle and secondly the amplifiers and vibration source.

Back Panel tests

The Back Panel's "speaker" mode at around 92 Hz together with the cabin's primary acoustic mode at 100 Hz was seen to generate and amplify the noise in the cabin in the booming frequency range. Two tests were performed to see whether modifications of the Back Panel would lead to a reduction in the booming noise

in the cabin. In the first test sandbags were pushed against the Back Panel to stop any vibration of the panel. In the second test, commercial "sound deadening pads" were affixed to the panel to see what effect a more practical fix would have on the booming noise.

The sandbag test resulted in a linearisation of the SPL at the passenger's inner ear between 25 km/h and 40 km/h, but little reduction in the noise levels (Figure 9). A reduction of up to 3 dB was achieved, considered to be "just perceptible" (van Niekerk, 2002) to the human ear. This showed that a redesign of the Back Panel, which should accomplish an even more significant change in the mass and stiffness properties of this panel, could have a significant influence on the booming noise in the cabin.

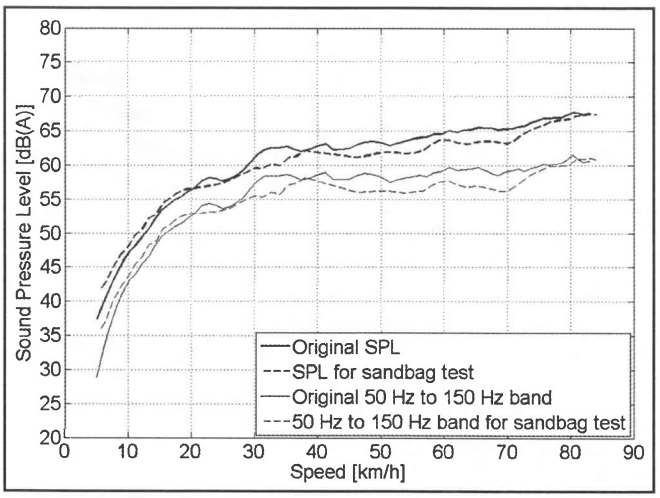

Figure 9: SPL levels at the passenger inner ear during the Sandbag test

The addition of the sound deadening pads to the Back Panel, however, had very little effect on the cabin's interior booming noise.

Drive shaft assembly tests

During the Run-up test it was seen that the front wheel vibrated unusually and showed the same vibration "thumb-print" as the vehicle's booming noise. This showed that something vibrated enough between 25 km/h and 40 km/h to generate vibrations in the body of the vehicle and excite the speaker. In addition to this, the transient nature of the vibrations suggested that the responsible component would have to change relative to the vehicle's speed i.e. it must be a rotating component. The TPA test had also established that vibrations at the front right wheel hub coupled better with the booming noise in the cabin than vibrations at the left front wheel hub.

The only difference between the left and right front wheel assemblies was the drive shaft assemblies: the right drive shaft assembly had an extension shaft running in a bearing connected through a bracket to the gearbox to connect the standard length drive shaft to the gear box (the gear box was not in the middle of the vehicle as the engine was transversely mounted). A series of tests were performed on initially the right side drive shaft assembly and subsequently on both drive shaft assemblies.

EMA of the right side drive shaft assembly

The right side drive shaft assembly was instrumented with three accelerometers on each shaft and shaken with a modal shaker. The resulting FRFs showed a significant, well-damped mode at around 83 Hz (Figure 10). The mode's shape was an almost rigid body vibration in the vertical plane as there is very little bending of the drive shafts. The point that experienced the greatest displacement was of the bearing in the bracket that connected the Extension Shaft to the engine housing i.e. at the midpoint of the shaft assembly, as seen in Figure 11. The displacement at the hub is also large. This meant that a large amount of energy would enter into the vehicle chassis at around 83 Hz if the shaft assembly were to be excited at that frequency.

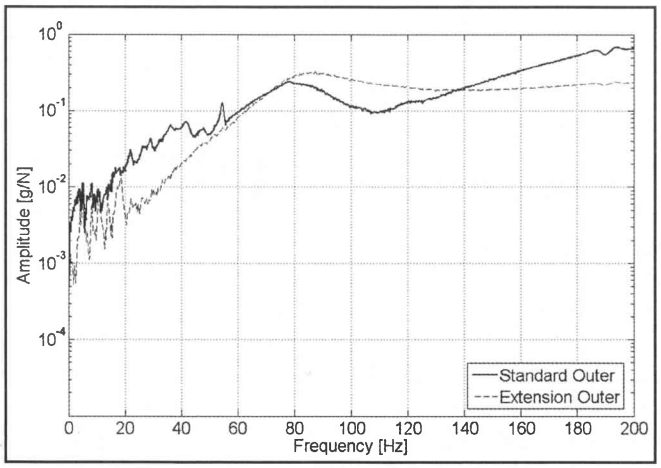

Figure 10: FRFs for the right hand side drive shaft assembly (Solid line response at 1Z+, dashed line response at 4Z+)

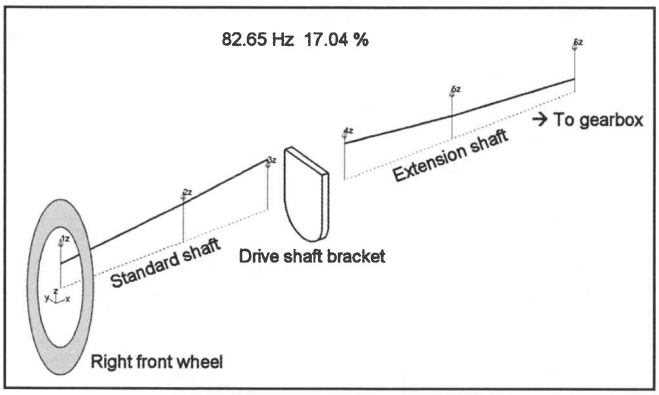

Figure 11: Schematic diagram of the right hand side drive shaft assembly, showing the first mode's shape

Operational test

To conclusively determine the contribution of the drive shafts to the "booming" noise, an operational test was performed

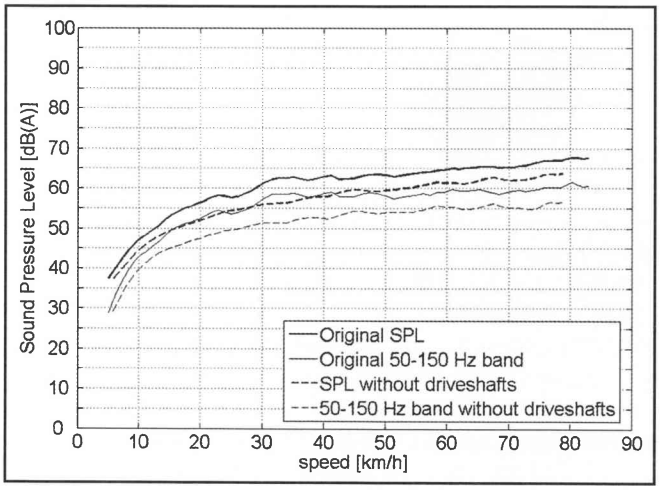

Figure 12: SPL plot for the vehicle with and without drive shafts

where the drive shafts were entirely removed from the vehicle. The vehicle was allowed to accelerated from rest down the same road used for all other operational tests with the engine idling. The internal SPL was measured at the passenger's inner ear and then compared to the internal ear SPL recorded during the Run-up test performed with the drive-shafts fitted.

The results (Figure 12) showed that the booming noise had largely been removed. The new SPL line was much smoother, increased much more linearly with speed (especially above 15 km/h) and was about 7 dB lower in the critical speed range of 30 km/h to 35 km/h. Keeping in mind that a difference of 5 dB is clearly discernable and a reduction of 10 dB sounds half as loud, this was a major improvement.

This proved that the drive shafts were at best responsible, and at least very influential in the generation of the booming noise in the vehicle cabin.

Operational TPA

An operational TPA was carried out to determine the paths along which the vibration passed through from the drive shafts to the vehicle cabin. Three main paths were identified and instrumented with accelerometers, namely through the engine and engine mounts, through the spring and damper, and through the wishbone suspension assembly. Again using the standard stretch of road, the vehicle accelerated from rest with the engine idling.

Results showed the primary vibration path to be through the wishbone suspension assembly (Figure 13). Measurements at the hub, at the inner and outer wishbone bushings, and on the chassis floor where the wishbones were bolted all showed the tell-tale "thumbprint" of vibration around 85 Hz to 105 Hz and 25 km/h to 40 km/h.

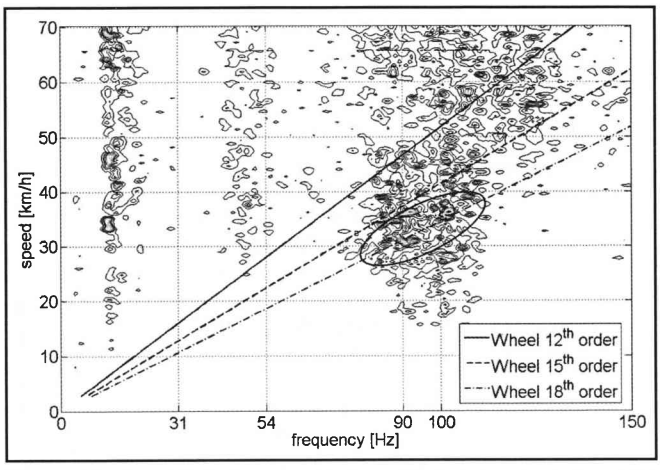

Figure 13: Campbell diagram of the vibration level of the right wishbone's inner swivel connection (Darker colours are higher vibration levels expressed m/s²)

The path of the vibrations and the reason for the amplification of the vibrations around 85 Hz were therefore known. The source of the vibrations in the drive shaft was still not known though. Biermann (2004) noted the effect of the orders of constant velocity (CV) joints on the vibrations of drive shafts. A tripod joint, with three balls, is especially sensitive to multiples of the third order of its drive shaft's rotation, while a ball joint, with six balls, was sensitive to multiples of the sixth order. When plotted against the Campbell diagrams generated for the Operational TPA data, it was seen that especially the wheel's 18th order line passed straight through the highest peaks in the "thumbprint" of vibration (Figure 13). The drive shafts were fitted with a tripod joint at the inner end and a ball joint at the outer end and the 18th wheel order corresponded with the tripod joint's 6th order and the ball joint's 3rd order.

This therefore meant that vibrations were caused in the drive shaft assemblies (both left and right) due to the CV joints' orders. When the wheels rotated at about 4.6 Hz (i.e. 83 Hz ÷ 18) or 29 km/h, the damped natural mode in the right drive shaft assembly was excited, causing higher vibration levels to enter the wheel hub, which travelled through the wishbone assembly into the vehicle chassis. These vibrations excited initially the Back Panel's speaker mode at around 90 Hz and subsequently the cabins fundamental acoustic mode at around 100 Hz.

Proposed and tested solutions

The vibration level—and therefore the noise level—in the cabin could potentially be reduced in a number of ways.

Firstly the noise source could be removed or altered. The Back Panel was seen to act as a significant speaker and a redesign of the panel would result in a reduction in the booming noise, as proved by the simple sandbag test. Simply adding sound deadening pads to the panel yielded no noticeable improvements. The other option of modifying the cabin was not feasible as the costs involved were prohibitive in the extreme. The sound wave could not be damped in the cabin as 5 cm or even 10 cm of insulation would have little or no effect on a 1.5 m sound wave.

A more realistic solution was to modify the drive shafts, especially the right side assembly, to reduce the excitation around 83 Hz. This included adding weight to the shaft to lower the natural frequency of the assembly, modifying the CV joints in the drive shafts to make the shafts less sensitive to the wheels' orders, and modifying the bracket where the right side assembly is bolted to the engine housing to raise the natural frequency of the assembly.

It was not clear what effect added weight would have on the shaft (aside from lowering its natural frequency) so this was tested. A 2 kg steel sleeve was secured to the right drive shaft's Extension Shaft near the mounting bracket. The resulting SPL (Figure 14) showed that while the sound levels were not noticeably reduced, the increase in the booming noise from 25 km/h to 32 km/h was linearised, making the increase around 30 km/h less intrusive and noticeable to the driver. This proved that the right driveshaft was responsible for the vibrations at the wheel hub and in the cabin. It also proved that a modification of the drive shaft assembly would decrease the booming noise in the cabin.

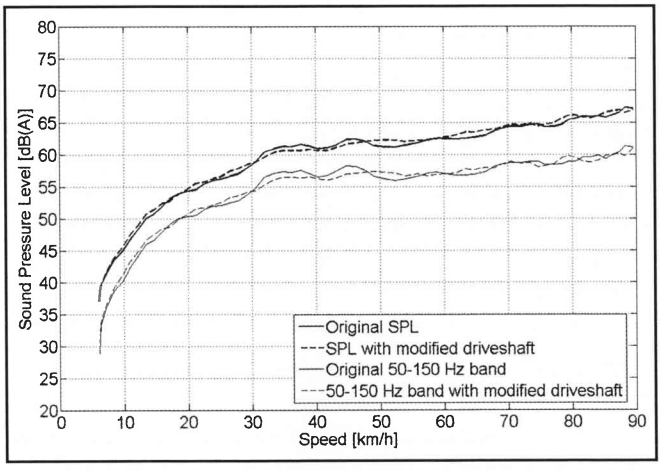

Figure 14: SPL for the vehicle with a modified right drive shaft

It seemed unlikely that modifying the configuration of the CV joints would help as even if the vibrations were avoided at a wheel rotation frequency of 4.6 Hz, it would probably just be moved to another frequency and consequently not solving the problem, only shifting it to another speed. Modification of the compliance of the drive shaft bracket was not deemed possible by the vehicle's manufacturing company.

Another solution that would be relatively easy to implement would be optimising the bushings in the wishbones so reduce the transmission of vibrations between 85 Hz and 100 Hz. This was unfortunately not seen as a realistic solution for the vehicle's manufacturing company as it would soften the vehicle's "sporty" handling characteristics, one of its major marketing features.

The vehicle manufacturing company was therefore presented with two potential solutions: modification of the Back Panel (which could at best only help to alleviate the problem) or modification of the right drive shaft assembly (which was proved to be difficult but effective).

Conclusions

An investigation into road rumble in an LUV has been presented and discussed.

- The road rumble ("booming") noise in the cabin was identified in the region of 85 Hz to 105 Hz at speeds of between 25 km/h and 40 km/h. This coincided with the cabin's fundamental acoustic mode, which was measured and calculated to be 100 Hz.
- EMA showed that a panel behind the seats, at the end of the load box, had a damped "speaker" mode at 90 Hz, which accounted for the booming noise between 85 Hz and 95 Hz. Simple modifications to the panel showed slight improvements in the booming noise (3 dB(A)).
- Vibrations were measured on the front wheel hubs that correlated with the booming noise's frequency and speed ranges. The right front hub showed higher levels of vibration than the left front hub. EMA of the right drive shaft assembly showed a damped mode at 80 Hz. Subsequent investigation showed that the CV joints on each side of the standard drive shaft excited the drive shaft assembly in multiples of their rotational orders. When the orders coincided with the 80 Hz mode, extra vibrations were generated in the drive shaft assembly.
- The path of the vibrations was identified as travelling from the right drive shaft assembly, through the hub and wishbone assembly to the chassis floor. The couplings between the chassis floor and the Back Panel and cabin would be numerous.
- Proposed solutions included modifying the right drive shaft assembly by adding mass to the shafts (which was tested with some success) or modifying the CV joint configuration, optimising the wishbone bushing to absorb 80 Hz to 100 Hz vibrations more effectively and modifying the Back Panel to move or reduce its 90 Hz speaker mode.

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