

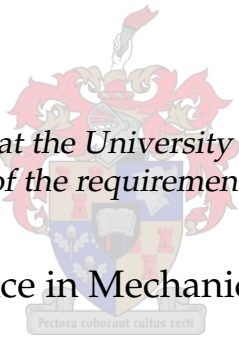


UNIVERSITEIT • STELLENBOSCH • UNIVERSITY
jou kennisvennoot • your knowledge partner

Investigation into Road Rumble in a Light Utility Vehicle

by

Andrew David Wade



*Thesis presented at the University of Stellenbosch in
partial fulfilment of the requirements for the degree of*

Master of Science in Mechanical Engineering

Department of Mechanical Engineering
Stellenbosch University
Private Bag X1, 7602 Matieland, South Africa

Study leader: Prof. J.L. van Niekerk

April 2005

Declaration

I, the undersigned, hereby declare that the work contained in this thesis is my own original work and that I have not previously in its entirety or in part submitted it at any university for a degree.

Signature:

A.D. Wade

Date:



Abstract

Investigation into Road Rumble in a Light Utility Vehicle

A.D. Wade

*Department of Mechanical Engineering
Stellenbosch University
Private Bag X1, 7602 Matieland, South Africa*

Thesis: MScEng (Mech)

April 2005

Vehicle Noise, Vibration and Harshness (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 best vehicle.

Road rumble is a key concern for vehicle manufacturers. Light Utility Vehicles (LUVs) are especially sensitive to a low frequency booming noise due to the fundamental acoustic mode that exists in the vehicle cabin. An investigation into this booming noise in an LUV 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 Acoustic Modal Analysis (AMA), Experimental Modal Analysis (EMA) and Transfer Path Analysis (TPA). The cabin's fundamental acoustic mode lay at 100 Hz. Finally the source of the vibrations in the vehicle leading to the booming noise in the cabin is identified, 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. Solutions to the problems associated with the fundamental acoustic mode of LUVs are also proposed and discussed.

Uittreksel

Ondersoek van Padgeraas in 'n Ligtevragevoertuig

("Investigation into Road Rumble in a Light Utility Vehicle")

A.D. Wade

*Departement Meganiese Ingenieurswese
Universiteit Stellenbosch
Privaatsak X1, 7602 Matieland, Suid Afrika*

Tesis: MScIng (Meg)

April 2005

Die vibrasie en geraas van voertuie is nou 'n meer belangrike komponent van die voertuig se ontwerpproses as ooit tevore. Padgeraas is een van die hoof kriteria wat onbewustelik potensiële kopers se besluit beïnvloed om die beste voertuig te kies.

Padgeraas is van groot belang vir die voertuigfabrikant. Ligtevragevoertuie ("bakkies") is veral sensitief vir lae frekwensie geraas as gevolg van die fundamentele akoestiese modus wat in die voertuig kajuit bestaan. 'n Ondersoek na hierdie dreuning in 'n bakkie is onderneem. Die geraas word geïdentifiseer en gekwantifiseer, waarna die bron van die geraas in die voertuig se kajuit geïdentifiseer word deur van tegnieke soos Akoestiese Modale Analiese, Eksperimentele Modale Analiese en Transmissiepad Analiese gebruik te maak. Die kajuit se fundamentele akoestiese modus kom by 100 Hz na vore. Ten slotte word die bron van die vibrasies in die voertuig wat tot die dreuning in die kajuit lei, geïdentifiseer, asook die pad van oordrag.

Oplossings vir die betrokke voertuig se geraas word voorgestel, waarvan twee getoets is met beperkte sukses. Oplossings vir die probleme verbonde aan die fundamentele akoestiese modus van bakkies word ook voorgestel en bespreek.

Acknowledgements

A sincere word of thanks to my study leader, Prof. Wikus van Niekerk, for all his help and guidance in completing this project. To my fiancée Sarah: I can't begin to describe how much your encouragement, support and prayer has helped me even start this project.

Many thanks also to all the people involved in the project, especially everyone in the Structures Laboratory of the Mechanical Engineering Department for their help and advice, Pieter for his help with computer programming and typesetting queries and Jaco Erasmus for his help with queries on the vehicle and its design. A word of thanks also to the National Research Foundation (NRF) for their partial funding of the project.



Dedications

This report is dedicated to God, my Father. He is my shepherd so I shall lack nothing.



Contents

Declaration	i
Abstract	ii
Uittreksel	iii
Acknowledgements	iv
Dedications	v
Contents	vi
List of Figures	viii
List of Tables	xi
Acronyms	xii
1 Introduction	1
2 Literature Survey	5
2.1 Booming noise in vehicles	5
2.2 Sound Quality	6
2.3 Tools used to analyse NVH problems in vehicles	8
2.4 Some success stories	18
2.5 Latest developments in NVH and vibration analysis	19
3 Noise Source Identification	21
3.1 Run-up test	21
3.2 Acoustic Modal Analysis	28

3.3	Experimental Modal Analysis	30
3.4	Transfer Path Analysis	32
3.5	Reciprocal test	33
3.6	Conclusion	34
4	Problem Identification	37
4.1	Back Panel tests	37
4.2	Drive shaft assembly tests	40
4.3	Operational Transfer Path Analysis	46
4.4	Conclusion	51
5	Potential Solutions	53
5.1	Vibration attenuation at the source	53
5.2	Transfer path modification	55
5.3	Vehicle cabin modification	56
5.4	The LUV's fundamental acoustic mode	57
5.5	Conclusion	60
6	Conclusions and Recommendations	61
	List of References	64
A	Test figures	68
A.1	Run-up test	68
A.2	Experimental Modal Analysis	71
A.3	Drive shaft assembly tests	72
A.4	Operational Transfer Path Analysis	73
B	The A-weighting scale	76

List of Figures

1.1	Schematic of an LUV	1
1.2	Fundamental frequency in a saloon car	3
1.3	Fundamental frequency in an LUV	3
2.1	Vector diagram of noise contributions of different panels (Hendricx <i>et al.</i> , 1997)	17
3.1	The binaural recording head inside the vehicle cabin	22
3.2	Sound Pressure Level (SPL) plot of the passenger's inner ear microphone	23
3.3	Campbell diagram in linear Pa^2/Hz of the passenger's inner ear microphone	24
3.4	Campbell diagram in A-weighted Pa^2/Hz of the passenger's inner ear microphone	25
3.5	Frequency band contribution in linear decibel (dB) of the passenger's inner ear microphone	26
3.6	Frequency band contribution in A-weighted decibel (dB(A)) of the passenger's inner ear microphone	27
3.7	Frequency Response Function (FRF) for the driver inner and outer ear microphones during the experimental AMA test	29
3.8	Predicted fundamental mode of the vehicle's cabin using analytical AMA	29
3.9	FRF for an accelerometer on the Back Panel	31
3.10	FRFs between vibration at the front and rear right wheels and the driver's inner ear	32
3.11	Campbell diagram of the vibration level of the front right wheel hub during the Run-up test	34

3.12 FRF for the front left and right wheel hubs due to reciprocal excitation	35
4.1 SPL levels at the passenger inner ear during the Sandbag test . . .	38
4.2 SPL levels at the passenger inner ear during the sound deadening pads test	40
4.3 FRFs for the right hand side drive shaft assembly	42
4.4 Schematic diagram of the right hand side drive shaft assembly, showing the first mode shape	43
4.5 SPL plot for the vehicle with and without drive shafts	44
4.6 Campbell diagram for the vehicle with and without drive shafts .	45
4.7 Campbell diagram of the vibration level of the right wishbone's outer swivel connection	47
4.8 Campbell diagram of the vibration level of the right wishbone's inner swivel connection	48
4.9 Campbell diagram of the vibration level of the vehicle chassis near the right wishbone	49
4.10 Campbell diagram of the vibration level of the driver's footwell floor	49
5.1 Right drive shaft bearing housing and bracket	54
A.1 Campbell diagram of the vibration level of the right rear wheel hub during the Run-up test	69
A.2 Campbell diagram of the vibration level of the Back Panel during the Run-up test	69
A.3 Campbell diagram of the vibration level of the windscreen during the Run-up test	70
A.4 The "speaker" mode shape of the Back Panel at 90 Hz	71
A.5 FRF for the Outer Shaft's outer-most accelerometer showing the calculated curve fit	72
A.6 Campbell diagram of the vibration level of the left wishbone's inner swivel connection	73
A.7 Campbell diagram of the vibration level of the left wishbone's outer swivel connection	74
A.8 Campbell diagram of the vibration level of the vehicle chassis near the left wishbone	75

LIST OF FIGURES

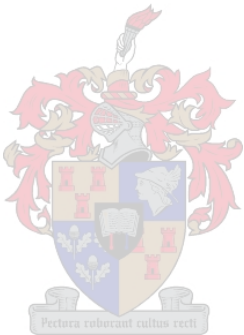
x

A.9 Campbell diagram of the vibration level of the driver’s footwell floor

75

B.1 Plot of the A-weighting scale

76



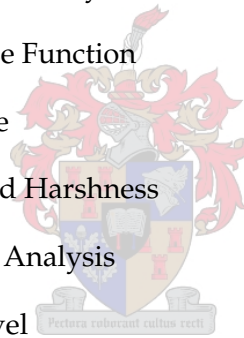
List of Tables

3.1	Comparison of cabin’s experimental and analytical acoustic modes	30
B.1	The A-weighting table	77



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
NVH	Noise, Vibration and Harshness
PCA	Panel Contribution Analysis
SPL	Sound Pressure Level
SQ	Sound Quality
TPA	Transfer Path Analysis



Chapter 1

Introduction

During the development of a Light Utility Vehicle (LUV) (Figure 1.1) a low frequency road rumble was identified by the motor manufacturing company as a Noise, Vibration and Harshness (NVH) area that needed 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.

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 (air-borne noise) and through the vehi-

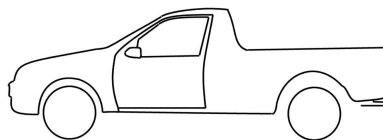


Figure 1.1: Schematic of an LUV

cle'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 Sound Quality (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 phase of the design process.

The automotive industry is becoming increasingly competitive. Motor manufacturers bring out many versions of a standard vehicle and there are consequently many manufacturers vying for their segment of the market. In South Africa there are three LUV manufacturers. Any edge that one manufacturer can obtain over another can therefore result in significant differences in sales figures. Consequently, when a manufacturer notices a noise problem in their LUV model, the problem must be solved as quickly as possible.

LUVs have a fundamental noise problem. When noise is generated in a cavity, the acoustic mode with the lowest frequency is generated 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 1.2) 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 corner of the cabin — just behind the driver's head (as seen in Figure 1.3). The mode's amplitude is therefore at its maximum near the

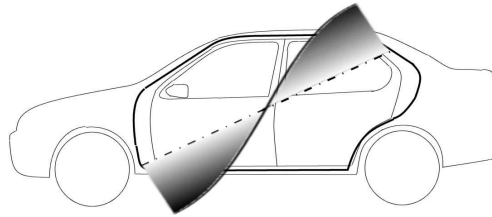


Figure 1.2: Fundamental frequency in a saloon car

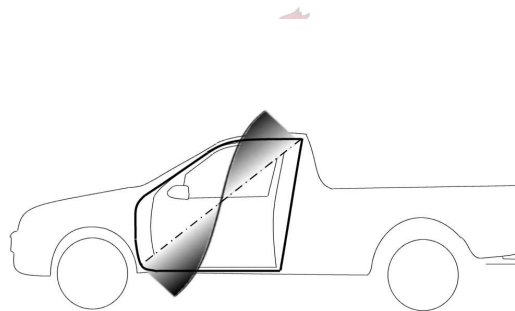


Figure 1.3: Fundamental frequency in an LUV

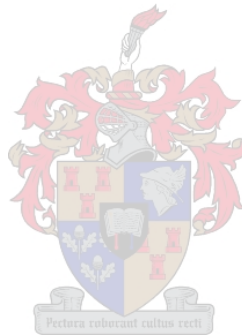
driver's ear. This is the fundamental problem that an LUV has with booming noise in the cabin and to remove it, the components in the vehicle that excite the fundamental mode must be identified and modified.

This report documents an investigation into an LUV's booming noise problem. A literature survey giving an overview of the field of vehicle NVH and specific methods used in the field is presented in Chapter 2. A selection of case studies is also included as well as some of the latest developments in NVH testing.

The investigation into the vehicle's booming noise is then documented in Chapter 3, starting with an analysis of the actual noise that pinpoints the characteristics of the noise that are of concern. Once the booming noise had been fully understood, the vehicle was then analysed to determine the source of the booming noise inside the cabin.

In Chapter 4 the source of the vibrations in the vehicle that lead to the booming noise in the cabin was sought and located. A cheap post-production "fix" was attempted on the vehicle with limited success.

Potential solutions to the booming noise in the vehicle as well as for the acoustic mode in LUVs in general are presented and discussed in Chapter 5.



Chapter 2

Literature Survey

Vehicle vibration and noise play a large role in determining a driver's attitude towards a vehicle (van der Auweraer *et al.*, 1997; Anon, 2003). Dehandschutter & Sas (1998) therefore state that Noise, Vibration and Harshness (NVH), which leads to the reduction in noise levels inside the vehicle cabin, is a crucial aspect in vehicle design.

The field of NVH is extremely large and there exist many different techniques and methods to analyse and solve noise related problems in vehicles. A survey was undertaken firstly to gain a general overview of the field of NVH and then to focus in on a few of the NVH techniques currently employed. These techniques included Sound Quality diagnosis, Correlation Analysis, Experimental Modal Analysis (EMA), Transfer Path Analysis (TPA), Acoustic Modal Analysis (AMA) and Statistical Energy Analysis (SEA).

2.1 Booming noise in vehicles

Vibrations in a vehicle are caused by myriad sources. Some of the more obvious sources are the interaction between the road surface and the vehicle's suspension system, engine vibration, tyre rolling vibrations and aerodynamic forces. All of these vibrations generate noise. The noise heard in the vehicle cabin can be generated in a number of ways. It is commonly accepted that noise with a frequency of over 1000 Hz is primarily transmitted via air-borne paths while noise with a frequency of under 500 Hz is primarily structure-borne (Lilley and Weber, 2003). The noise commonly referred to as "booming noise" is a low frequency noise — certainly below 500 Hz

— and is therefore almost entirely structure-borne. It has also widely been seen that this booming noise is especially present when a resonant frequency of a panel in the cabin coincides with one of the cabin's acoustic resonant frequencies (van der Auweraer *et al.*, 1997; Lee *et al.*, 2000; Matsuyama and Maruyama, 1998). A definition for “booming noise” in vehicles can therefore be constructed as follows:

“Booming noise in a vehicle is the low frequency noise heard when a structure-borne vibration enters the vehicle cabin and creates noise at the same frequency as one of the cabin's acoustic modes, exciting that mode and amplifying the noise.”

2.2 Sound Quality

When looking to solve a noise problem in a vehicle, the noise must first be assessed to determine which components of the noise are unpleasant. Van der Auweraer *et al.* (1997) suggests that this should be performed by replaying the sound either to professionals (if diagnosis is the objective) or to a general panel of listeners (if a description of quality is the objective). These results are then used as a point of departure for the rest of the analysis.

Once the noise has been rated subjectively, an objective Sound Quality (SQ) diagnosis is performed to understand why a sound possesses a certain quality. This involves analysis by means of signal processing (using techniques such as spectrum analysis) to determine the signal's characteristics, monitoring the effect of a modification of the noise's signal and synthesising sounds that contain certain characteristics from the noise's signal for parametric analysis. Van der Auweraer *et al.* (1997) shows (using filters) that a significant drop in noise level over a narrow frequency range is almost indiscernible, while a smaller drop in noise level over a wider frequency range is clearly audible. It is therefore more desirable to slightly reduce the noise level in a vehicle over a large frequency range than to dramatically reduce the noise level over a very small frequency range.

When evaluating a vehicle's SQ performance, so-called SQ metrics are used which rate the quality of a certain aspect of the sound. The most commonly used metrics are loudness, sharpness and fluctuation strength and roughness (Fastl, 1997).

The human ear does not hear all sounds with the same sound pressure content as equally loud; it depends heavily on the frequency content of the

sound. For this reason, the “loudness” of a measured sound cannot be determined using a simple Sound Pressure Level (SPL) measurement. Instead, the frequency content must be adjusted to account for the variation in the ear’s sensitivity. This has led to the development of a number of weighting scales, most notably the A-weighting scale, which is described in more detail in Appendix B. The sound’s loudness can then be more accurately represented once the linearly measured sound has been weighted.

Due to the non-linear response of the human ear to sound pressures of different frequencies, the “sharpness” of the sound also plays a major role in evaluating the quality of a measured sound. Higher frequencies are usually experienced as more irritating and annoying than lower frequencies, leading to the measure of the sound’s sharpness. It is often more desirable for a sound to have a low sharpness value, leading to a technique known as “masking”. Using this technique, noise is added to the original noise in an effort to make the sound more pleasing. A lower frequency noise can be added to the original sound to lower its sharpness. This would only be attempted if the more desirable option of removing the higher frequency noise is not possible.

An interesting phenomenon that arises out of “masking” is that the removal of one sound could bring another, more irritating sound to the fore that was being “masked” by the former. This phenomenon is very difficult to foresee when improving a sound’s quality and would only be discovered once the dominating sound has been removed.

The fluctuation strength or roughness of a sound refers to the amount a sound’s intensity or frequency fluctuates. The definition of fluctuation strength and roughness are the same, however fluctuation strength is defined below 20 Hz and roughness above 20 Hz. The reasoning behind this metric is that a sound that fluctuates around its main frequency or intensity will be more noticeable and consequently more irritating to the human ear. A common example of fluctuation strength is the “beating” frequency heard when two similar frequencies superimpose to create a beating sound. This is often heard when a guitar string is slightly out of tune and the (supposedly) same note is produced by a true string and the false string. Apart from the dissonance created, a throbbing sound with a beat frequency of about 1 Hz is also heard. This beat frequency can become quite noticeable and irritating.

2.3 Tools used to analyse NVH problems in vehicles

Once the noise's signal is understood, the problem must be physically understood by identifying the vehicle components that generate, transmit and intensify the vibrations that cause the identified problematic noise in the cabin. Correlation Analysis considers the relationship between the noise and particular vehicle components. This involves comparing the critical components of the signal to components in the vehicle. These can be vibrations (bodywork), wind noise, noise from the tyres rolling on the road surface, engine noise or a number of other factors. This can range from a simple visual or intuitive comparison to an in-depth coherence study of the signals. This technique is often applied subconsciously when a particular sound is "recognised" to come from a certain source.

The following techniques, often employed in NVH studies, have very mathematical foundations and as a result give fairly objective results.

2.3.1 Experimental Modal Analysis

Experimental Modal Analysis (EMA) is one of the most widely used techniques in NVH today, probably due to its wide applicability, simplicity and easily interpretable results. It has been used in many studies, from determining the natural modes of a thin disk (Wassermann and Springer, 2001), to identifying the fundamental modes of an SUV (Hwang *et al.*, 1997), to the analysis of a violin octet (Bissinger, 2003). While the more basic and traditional method of accelerometers will be used for measurements, more advanced methods are being introduced into the field of NVH. One of these, holographic modal or holo-modal analysis, where the component is analysed using laser sensors, is described in more detail in Section 2.5.1.

2.3.2 The principle of reciprocity

While not a technique by itself, the principle of reciprocity is an extremely useful tool in analysing a vehicle's NVH properties. It has mainly been used in TPA (Matsuyama and Maruyama, 1998; van der Linden and Fun, 1993) and Panel Contribution Analysis (PCA) (Hendricx *et al.*, 1997). The principle of reciprocity reduces to "if a certain vibration causes a certain resultant vibration, then exciting the resultant vibration must cause the initial vibration" i.e. the system behaves in a reciprocal manner. This sounds logical but is only

possible if the system is linear. “*The basic requirement for reciprocity is linearity*” (van der Linden and Fun, 1993). Fortunately, as shown by van der Linden & Fun (1993), the deviation of peak values between direct and reciprocally measured transfer functions in vehicles is less than 3 dB between 30 Hz and 500 Hz. This has also been verified by Matsuyama & Maruyama (1998). It is therefore acceptable to use reciprocal measurements when calculating transfer functions in vehicles.

When measuring noise in a vehicle’s cabin by exciting points on the vehicle, it is often extremely difficult and impractical to attach the excitation equipment to the required components, due to a lack of space around the components. This is where reciprocal measurements become invaluable. Instead of vibrating say the left rear suspension and measuring the noise inside the cabin, noise can be generated inside the cabin and the resulting vibrations measured on the suspension. Instead of attaching a shaker to the suspension (or using a modal hammer) all that is required is an accelerometer. Another advantage is that more than one measurement can be made at once; notionally all components could be measured in one fell swoop if the equipment is available. Reducing the number of experiments, apart from saving time and effort, improves the consistency of the data as the transfer functions are calculated relative to only a small number of different excitations as opposed to every transfer function being calculated from an independent excitation (van der Linden and Fun, 1993).

The sound source in the cabin must be of a certain quality for reciprocal measurements to be effective. According to van der Linden & Fun (1993) there are three requirements that need to be fulfilled when choosing a sound source. Firstly the sound source should “approximate a point source with omnidirectional radiation”. The source should also produce high levels of acoustic output at low frequencies and its effective volume velocity (the velocity at which the sound source pumps the air) must be accurately measured. Other authors have not seen the need for the first requirement and have obtained good results using a simple loudspeaker as a sound source (Matsuyama and Maruyama, 1998; Maruyama *et al.*, 1999). The third requirement can be met by using either a microphone placed next to the sound source, or a small, light accelerometer attached to the sound source (Hendricx *et al.*, 1997). The latter is usually more practical and accurate.

2.3.3 Transfer Path Analysis

Transfer Path Analysis (TPA) is a fairly well-established technique (Wyckaert and van der Auweraer, 1995; Plunt, 1998). It is used by the NVH engineer to determine which components of the vehicle are most responsible for transmitting vibrations from the source to the cabin.

Structure-borne noise generation involves three elements: the source, the transfer path and the receiver (Dehandschutter and Sas, 1998). While the source can very rarely be changed (a tyre must be in contact with the ground and suspension must oscillate on a rough road) the transfer path — if known — can be modified so that less vibration, and consequently noise¹ is transferred to the cabin.

TPA is a partial pressure technique. The partial contribution as a function of frequency or engine revolutions per minute (rpm) of each transmission path to the cabin's total sound pressure is measured to identify the most dominant path (van der Auweraer *et al.*, 1997). The total cabin sound pressure is calculated as follows (Lee *et al.*, 2000)

$$P = \sum P_i = \sum F_i \cdot (P/F_i) \quad (2.3.1)$$

where P = total cabin sound pressure;
 P_i = cabin sound pressure transferred through i 'th path;
 F_i = applied force at i 'th path;
 P/F_i = mechano-acoustic transfer function for i 'th path.

The applied force can be calculated either directly or indirectly (van der Linden and Fun, 1993).

The direct method is an estimation based on the stiffness of the vehicles mountings. These could be the bushings, hydraulic mounts, a coupling, a joint or a bearing. The force is then calculated using the following equation

$$\overrightarrow{f_{body}} = \frac{1}{\omega^2} \cdot [K] \left(\ddot{\vec{x}}_{ex} - \ddot{\vec{x}}_{body} \right) \quad (2.3.2)$$

where $\overrightarrow{f_{body}}$ = estimated force vector on the body side of the mounting (N);

¹“noise” here meaning the specifically identified problem noise

- ω = frequency (rad/s);
 $[K]$ = mounting stiffness matrix (N/m);
 $\ddot{\vec{x}}_{ex}$ = operational acceleration vector at the excited side of the mounting (m/s²);
 $\ddot{\vec{x}}_{body}$ = operational acceleration vector at the body side of the mounting (m/s²).

The disadvantage of this method is obviously the reliance on the stiffness matrix $[K]$. This is not always readily available. In addition to this, “sufficient deformation of the mounts” is required for reliable data (van der Linden and Fun, 1993).

The indirect method estimates the force based on the acceleration of the supporting structure i.e. the chassis, body or cabin as follows

$$\vec{f}_{body} = [H]^{-1} \cdot \ddot{\vec{x}}_{body} \quad (2.3.3)$$

where $[H]$ is the accelerance matrix (ms⁻²/N) which is the transfer function matrix between forces and accelerations at all attachments in all directions. The accelerance matrix is usually measured with hammer impact measurements.

If a road test is performed, multiple incoherent sets of data are measured, making the task of separating each set into coherent sets more complex. The data measured during the test cannot be simply added as a complex sum to yield the total contributions; the data must be separated into several sets of coherent contributions, on which regular TPA is then performed. Once the partial pressure contributions of each transfer path have been calculated, they can be summed using the ordinary rms equation

$$P_{rms} = \sqrt{P_1^2 + P_2^2 + P_3^2 + \dots + P_n^2} \quad (2.3.4)$$

where P_i is the partial pressure for transfer path i of n transfer paths. This is because phase relationships only exist between signals coherent with one phenomenon and not between incoherent phenomena (Hendricx and Vandenbroeck, 1993). In simple terms this means that since there is no phase relationship between the sound pressure inside the cabin caused by, say, the right front wheel shaking and that caused by the drive train vibrating, the

resultant sound pressures can be summed using the ordinary rms equation (Equation 2.3.4).

During a laboratory test, this is not a concern as each potential source can be individually excited, resulting in a coherent set of measurements for each experiment. Conventional rms sums can then be used to compare the partial pressures due to excitation at each source.

2.3.4 Acoustic Modal Analysis

Acoustic Modal Analysis (AMA) concerns the investigation of the cabin's acoustic properties. This investigation can either be performed analytically or acoustically. Both methods are often used for correlation.

Experimental AMA

Experimental AMA has been performed for many years now. There are a variety of methods, although the two most commonly used methods are the “paired microphone” and “single microphone” methods.

The paired microphone method (described in detail by Knittel & Oswald (1987)) uses a finite-difference method to approximate the acoustic particle acceleration. The governing equation is Newton's Second Law (for example in the x-direction)

$$\rho \frac{\partial v(x, t)}{\partial t} = - \frac{\partial p(x, t)}{\partial x} \quad (2.3.5)$$

and is solved for the particle acceleration using two microphones and a finite difference approximation i.e. substituting $(p_1 - p_2)$ for ∂p and the microphone spacing Δr for ∂x . Assuming the microphone spacing and the density remain constant², Equation 2.3.5 can be simplified as

$$\rho \frac{\partial v(x, t)}{\partial t} = k(p_2 - p_1) \quad (2.3.6)$$

where k is a constant $((\rho \cdot \Delta r)^{-1})$.

This method is then implemented by inserting paired microphones — a distance Δr apart — in the acoustic cavity being excited by a noise source and measuring the acoustic particle accelerations in each direction. Particle

²The incompressible flow approximation is true for flows with a velocity of less than Ma 0.3 (about $100 \text{ m} \cdot \text{s}^{-1}$) (White, 1999) which is well within this application's limits.

acceleration Frequency Response Functions (FRFs) are then obtained using either a third microphone mounted near the noise source or an accelerometer mounted on the noise source.

The main advantage of this method is that standard Fast Fourier Transform (FFT) equipment can be used to perform the measurements (this method was developed in the late 1980's) and that highly accurate measurements are possible.

The main disadvantages of this method are the reliance on an accurately determined finite difference for the pressure ($p_2 - p_1$) and the large number of phase-matched microphones required. Multiple experiments could be performed, moving a single set of phase-matched microphones to each measurement location for each experiment, but this would result in furiously many experiments and consequently a greater chance of measurement errors.

Another experimental technique that has proved quite accurate (Hwang *et al.*, 1997) and much simpler to implement requires only one microphone per measurement. The acoustic cavity is simply divided into a three-dimensional grid, excited using white noise with the required bandwidth and the acoustic pressure measured using a single microphone in each direction at each node. The source's acceleration is measured using an accelerometer placed on the source. The FRFs are then measured by averaging a number of measurements. This is the most common method for performing an experimental AMA.

The advantage of this method is that only one microphone is required for each measurement so numerous nodes can be measured simultaneously.

Analytical AMA

Analytical AMA has been used by many NVH engineers. As a result there are many techniques and as many software packages that use this technique. Most Finite Element Analysis (FEA) software packages have some acoustic analysis capabilities ranging from a rather simplified "rigid boundary" approach to highly complex "fluid boundaries", acoustic barriers and fluid couplings (MSC, 2001). The level of complexity of the analysis depends on the software available, the accuracy required, the time available and the level of expertise of the user. If the only AMA being performed on the vehicle is

analytical, a high level of accuracy is preferred as there will be no experimental data available to correlate the analytical results with.

Primary acoustic mode shapes of a rectangular box, and their corresponding frequencies can be approximated using the equation

$$f(n_x, n_y, n_z) = \frac{c}{2} \sqrt{\left(\frac{n_x}{l_x}\right)^2 + \left(\frac{n_y}{l_y}\right)^2 + \left(\frac{n_z}{l_z}\right)^2} \quad (2.3.7)$$

where c is the speed of sound in air, $n_{x,y,z}$ are the mode orders in each direction (0, 1, 2, ...) and $l_{x,y,z}$ define the cabin dimensions (Hwang *et al.*, 1997).

Acoustic modes have been determined analytically and experimentally by Hwang *et al.* (1997) and then compared to the cabin noise. Correlations were successfully found at some of the frequencies.

Structural and (analytical) acoustic modal analyses were also performed by van der Auweraer *et al.* (1997). They showed coincident structural and acoustic modes between the rear suspension and cabin of the vehicle. This, together with a contribution analysis for the rear suspension to the overall noise in the cabin, gave a clear indication that the rear suspension was largely responsible for the interior noise at the targeted frequency.

2.3.5 Operational Modal Analysis

Operational Modal Analysis is used to obtain data more realistically than laboratory experiments can yield. Basically, the vehicle is instrumented with many accelerometers and microphones and driven on a suitable track. A number of signals are used as reference signals (typically the microphones at the driver's and passengers' ears) and the phases of all the measured data are matched to these reference signals. By decomposing these signals into signals coherent with each reference signal (in much the same way as is done in TPA), structural and acoustic modes can be calculated from the resulting transfer functions (Deweert and Dierckx, 2000; Wyckaert and van der Auweraer, 1995; Lee *et al.*, 2000).

2.3.6 Panel Contribution Analysis

Panel Contribution Analysis (PCA) is essentially a variant of TPA. In TPA the paths along which the vibrations travel are analysed to identify the path

contributing most to the cabin noise at the targeted frequencies. In PCA the panels in the cabin are analysed to identify the panel contributing most to the cabin noise at the targeted frequencies. A detailed description of this method can be found in Hendricx *et al.* (1997).

The mathematical background to PCA is very similar to that of TPA. The total interior noise in a vehicle can be described as a linear sum of partial pressure contributions (Hendricx *et al.*, 1997):

$$P = \sum P_i = \sum Q_i \cdot (P_i / Q_i) \quad (2.3.8)$$

where P = total cabin sound pressure;
 P_i = cabin sound pressure contribution of body panel i ;
 Q_i = volume velocity of body panel i ;
 P_i / Q_i = acoustic FRF between the interior microphone and body panel i .

There are consequently two unknowns for each body panel's sound pressure contribution (P_i): the volume velocity of the panel and the acoustic FRF between the interior microphone and the body panel. In practice the volume velocity of the body panel is difficult to measure so the volume acceleration $\dot{Q}_i$ of each panel is measured instead. The acoustic FRF then becomes $P_i / \dot{Q}_i$.

The volume acceleration of the panels in the cabin is measured using accelerometers. Each panel is divided into subpanels and an accelerometer is fixed to the centre of the subpanel to measure the normal accelerations of the subpanel. Only accelerations coherent with cabin interior noise should be considered. These coherent accelerations are calculated by normalising the crosspowers between the target microphone and each of the accelerations as shown in Equation 2.3.9

$$\ddot{x}_\perp|_p = G_{\ddot{x}_p} / \sqrt{G_{pp}} \quad (2.3.9)$$

where $\ddot{x}_\perp|_p$ = normal acceleration of subpanel coherent with target microphone pressure p ;
 $\ddot{x}_\perp$ = measured normal acceleration of subpanel;
 p = measured sound pressure at target microphone;
 $G_{\ddot{x}_p}$ = crosspower between $\ddot{x}_\perp$ and target microphone;
 G_{pp} = autopower of measured pressure.

The main assumption of the method is that the measurements by the accelerometers are representative of each subpanel in its entirety; this leads to measurement errors. Hendricx *et al.* (1997) states that the allowable size of each subpanel depends on the frequency band of interest. Two requirements are that each subpanel should be

1. smaller than half of the structure's mechanical wavelength;
2. smaller than half of the acoustical wavelength.

The length of a 300 Hz sound wave is more than 1 m so the latter requirement is easily met. The first requirement is more complex as a full vehicle modal analysis is required to calculate the structure's mechanical wavelength. Both Hendricx *et al.* (1997) and Lee *et al.* (2000) assumed the wavelength to be sufficiently long with no ill effects.

The volume acceleration of each panel $\dot{Q}_i$ is then calculated by multiplying each panel's measured normal acceleration $\ddot{x}_i$ with its surface area S_i

$$\dot{Q}_i = \ddot{x}_i \cdot S_i \quad (2.3.10)$$

The acoustic FRF is most easily measured using reciprocal measurements (Hendricx *et al.*, 1997). The cabin is therefore excited using a sound source at, say, the driver's ear and the pressure responses are measured very close to the different panels. The volume acceleration of the sound source is measured in the same way as for the panels, and serves as reference for the acoustic FRF calculation. The response pressures of the panels are measured by microphones that are placed in exactly the same position as the accelerometers were for the volume velocity measurement.

Once the acoustical contributions of all the panels have been determined, they can be added together and compared to the total measured interior noise. This will validate the panel contribution calculations as the correspondence should be quite good. A very important point to note is that even if the contribution of one specific panel dominates the contributions at a certain frequency, it is *not necessarily* the main acoustic radiator. The total interior noise is a *vector sum* of the acoustical contributions of the surrounding panels at that particular frequency (Hendricx *et al.*, 1997; Lee *et al.*, 2000). This is shown in Figure 2.1. This figure shows that while one panel can easily create a lot of noise, it might be balanced by another panel (the firewall balances

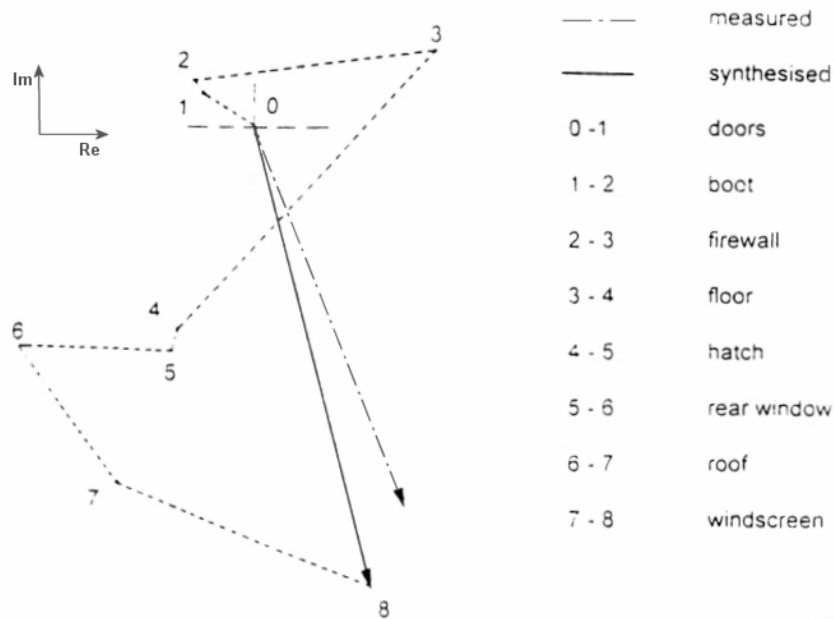


Figure 2.1: Vector diagram of noise contributions of different panels (Hendricx *et al.*, 1997)

the rear window), or the sum of a number of panels. In this case, one panel acts as a noise source and the other as a noise absorber. Likewise panels such as the roof and windscreen in Figure 2.1 can work together to amplify each other's noise. Hendricx *et al.* (1997) advise detailed analyses to better understand the vehicle's dynamics before jumping to conclusions. Lee *et al.* (2000) combined the results of PCA with TPA, EMA and AMA before deciding on the greatest contributing panels.

2.3.7 Statistical Energy Analysis

Statistical Energy Analysis (SEA) is a method used to analyse systems with many modes where it does not make sense to analyse each mode separately. SEA groups all the modes into subsystems and statistically determines their interaction. This is particularly useful when the tested object — in this case the vehicle body — has many closely-spaced modes.

SEA is documented to work well for frequencies above 200 Hz (Parrett *et al.*, 1997) but not for any frequencies lower than that (Hepburger *et al.*,

2002). DeJong (1985) states that a vehicle passenger compartment would not yield good results using SEA as the frequencies of the first modes are too low.

2.4 Some success stories

Generally, a paper is not published unless there is some success. This section is aimed more at highlighting some of the more typical combinations of methods that have been employed to solve real vehicle NVH problems.

Van der Auweraer *et al.* (1997) dealt with a vehicle having a booming noise at around 100 Hz. A TPA was performed, which identified the rear axle mounts as noise propagating paths. A multi-reference operational analysis was then performed using a large amount of microphones in the cabin and accelerometers on the suspension. An acoustic mode at 75 Hz and a large rear suspension displacement at 80 Hz were identified. An EMA on the rear suspension and an AMA in the cabin showed both a structural and an acoustic mode at 80 Hz. Since the cabin structure could not be redesigned, a modified suspension design (stiffening of a beam) was recommended to reduce the cabin noise.

Van der Linden & Varet (1996) applied a type of PCA to a booming noise problem in a diesel van. Second order engine noise dominated at the passenger's ear at around 1400 rpm and 2600 rpm under full load. FRFs were calculated between the cabin panels and the passenger's inner ear. An AMA was also performed on the cavity. The wind shield proved to be the dominant source of the noise at 1500 rpm. A modification lowered its first resonant mode, meaning that it caused more noise at 1100 rpm. This was acceptable as it is less common for a vehicle to be under full load at 1100 rpm than at 1500 rpm. The noise at 2400 rpm posed more of a problem. The cabin's lateral acoustic mode was excited at the noise's frequency and the cabin floor could not be easily modified so instead two smaller panels were "tuned" so that their first bending modes moved to 90 Hz. They were near the pressure maxima and could therefore act as efficient panel absorbers (as described at the end of Section 2.3.6). A reduction of about 6 dB was achieved around 90 Hz as a result of this modification.

Matsuyama & Maruyama (1998) analysed a 4 cylinder minivan to reduce the booming noise inside the cabin due to second order engine noise. The cabin structure was excited using both direct structural excitation and recip-

rocal excitation using sound sources. Both methods showed that the vehicle's B pillar deformed much more than any other components and was accordingly stiffened. This resulted in a 10 dB improvement in the 90 Hz booming noise at the driver's ear. An AMA test also showed the cabin to have an acoustic mode at 90 Hz which would have aggravated the problem.

2.5 Latest developments in NVH and vibration analysis

2.5.1 Holographic modal analysis

One of the most significant recent advances in EMA is holographic modal (or holo-modal) analysis (Anon, 2003). In conventional EMA, a panel is instrumented with a number of accelerometers which are connected to some sort of data capturing device. This method has a number of drawbacks, including mass loading of the panel (especially significant for thin, lightweight panels) and poor resolution; there is only information for a point where there is an accelerometer. With holo-modal analysis, the test panel is illuminated by a laser beam. The reflected light is then measured and a model is generated of the vibrating panel. This means that there is no mass loading of the panel and that the resolution can be as high as required, with computing power and memory being the only restrictions. It therefore makes modal analysis of especially complex structures much simpler and more accurate. The main disadvantage of this method is obviously its price; many companies would not be able to afford the holo-modal testing equipment.

This method was developed by BMW, LMS International, Dr Steinbichler and the Free University of Brussels.

2.5.2 A Wave Based Method for analytical AMA

A new approach to analytical AMA has been developed by Hepburger *et al.* (2002) at the Acoustic Competence Centre G.m.b.H. . An analysis tool was developed using the Wave Based Method after seeing the inaccuracies in the Finite Element Method for frequencies above about 100 Hz and in Statistical Energy Analysis below about 400 Hz. The Finite Element Method becomes inaccurate as a result of the increased number of elements required to describe the short wavelength behaviour at higher frequencies.

Tests showed the Wave Based Model to be more accurate than the Finite Element Method and to use far less computing time in calculating a test cavity's acoustic properties in the 0 Hz to 500 Hz range.



Chapter 3

Noise Source Identification

As mentioned in Chapter 1, 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.

When investigating an apparent noise problem in a vehicle, the first thing that must be done is to identify exactly what the problem noise’s characteristics are. For this project that meant identifying the so-called booming noise and then quantifying this noise in engineering terms. The noise’s intensity and especially its frequency content were of particular import. To these ends, a series of tests were performed on the vehicle.

3.1 Run-up test

As mentioned in Section 2.1, a low frequency rumble is generally transmitted via the vehicle’s bodywork. The Noise, Vibration and Harshness (NVH) engineer does not know though whether the noise is generated by the vehicle’s interaction with the road, by the engine’s vibrations or by a combination of the two, although he does hope that it is not the latter. To determine the cause of the vibrations in the vehicle that lead to the rumble, a run-up test was performed.

The test was carried out on a straight, well-bitumenised road of near-constant slope. The vehicle accelerated from the top of the slope from rest to over 80 km/h and was instrumented with accelerometers on the hubs of



Figure 3.1: The binaural recording head inside the vehicle cabin

three of the four wheels (to measure the vibration input to the vehicle) as well as on the panels surrounding the cabin. Two microphones were placed inside the vehicle: one at the driver's left (inner) ear and one on the right hand side of the passenger's seat. The test was repeated with a binaural recording head placed on the passenger's seat (Figure 3.1). The specific equipment used for the test is detailed in Appendix A.1.

The engine was left to idle at less than 1000 rpm and it was assumed that its vibratory contributions were constant and negligible. This therefore meant that if the low frequency "booming" noise were to be identified during this test, the rumble would have been caused by the road-induced vibration of the vehicle and not by the engine's vibrations. The noise could then be classified as "road rumble". Any transient changes in the noise level would also only be attributable to the road-induced vibration as the engine vibrations remained constant during the test.

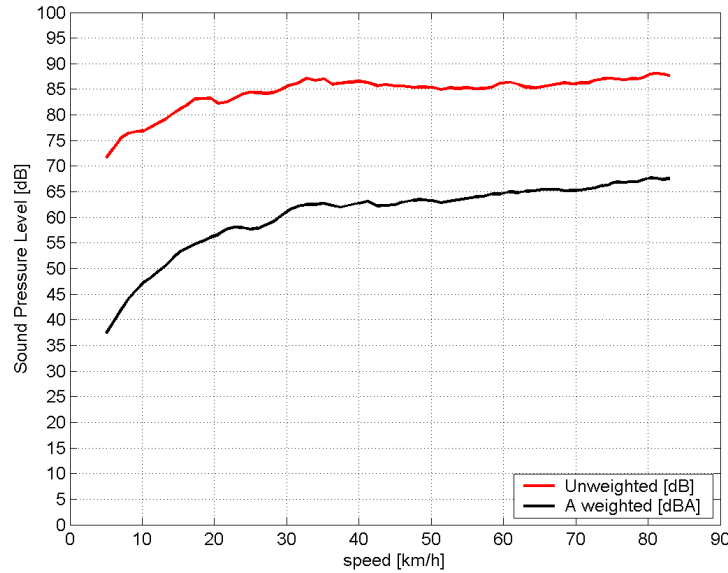


Figure 3.2: SPL plot of the passenger's inner ear microphone

The Sound Pressure Level (SPL) for both microphones was determined, first using the linear decibel (dB) scale and then using the A-weighted decibel (dB(A))¹ scale. Figure 3.2 shows the dB and dB(A) SPLs for the passenger's inner ear. The dB(A) levels are significantly lower than the dB levels due to the signal's high low frequency content that the A-weighting process suppresses. A marked increase of 5 dB can be seen in the A-weighted signal from 25 km/h to 30 km/h. It must be remembered that on the decibel scale, an increase of 3 dB is considered to be "just perceptible", 5 dB is clearly perceptible and 10 dB is perceived to be twice as loud as the original level. The linear scale on the other hand showed two distinct "bumps" in the signal: from 15 km/h to 20 km/h and from 30 km/h to 35 km/h.

Since the noise signal contained so much low frequency data, it was not clear whether the A-weighted signal was a legitimate measure of what the driver would have heard. It was therefore decided to investigate further

¹The A-weighted dB scale is a theoretical measure of the perceived loudness of a noise. It is a weighting function that amplifies certain frequencies and attenuates others. It is especially severe on noise below 100 Hz. It has been adopted in many national and international codes and standards (van Niekerk, 2002). A description and plot of the weighting table is included in Appendix B.

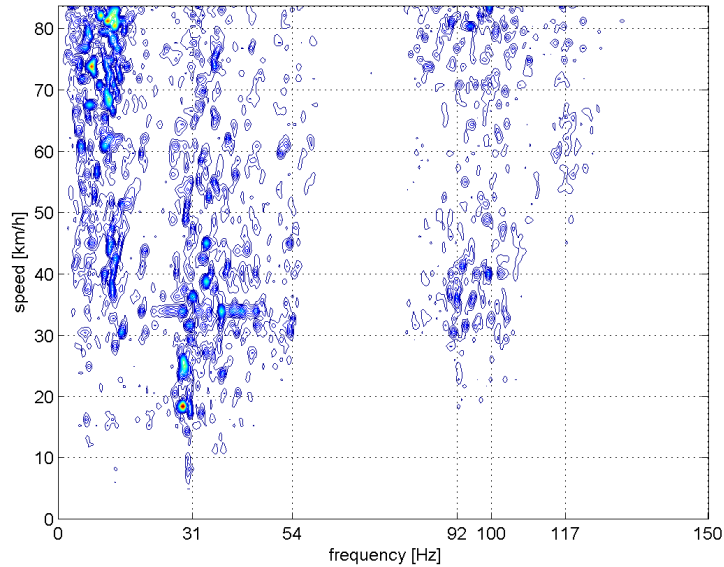


Figure 3.3: Campbell diagram in linear Pa^2/Hz of the passenger's inner ear microphone

whether the linear signal or the A-weighted signal was the most legitimate method to represent the noise that the human ear would have “heard”.

The noise data was converted from the time to the frequency domain to analyse the frequency content of the noise. Normal Frequency Response Function (FRF) plots were of little use to analyse a run-up test of this nature as there was no time reference to compare the frequencies to. For this reason Campbell diagrams were plotted. These plots use Power Spectral Densities (PSDs) to show the intensity of the different frequencies of the noise as a function of frequency and time. The time axis was then replaced by the vehicle's speed as this made the data interpretation easier. Figure 3.3 shows a Campbell diagram of the passenger's inner ear microphone. The data was not A-weighted. The engine's second order vibration can be clearly seen at 31 Hz as well as noticeable noise between 80 Hz and 110 Hz in the 30 km/h to 40 km/h region. These large low frequency components were very likely the reason for the high linear dB levels in Figure 3.2.

The data were once again A-weighted to obtain “perceived” noise levels. The A-weighted Campbell diagram in Figure 3.4 shows the effect of

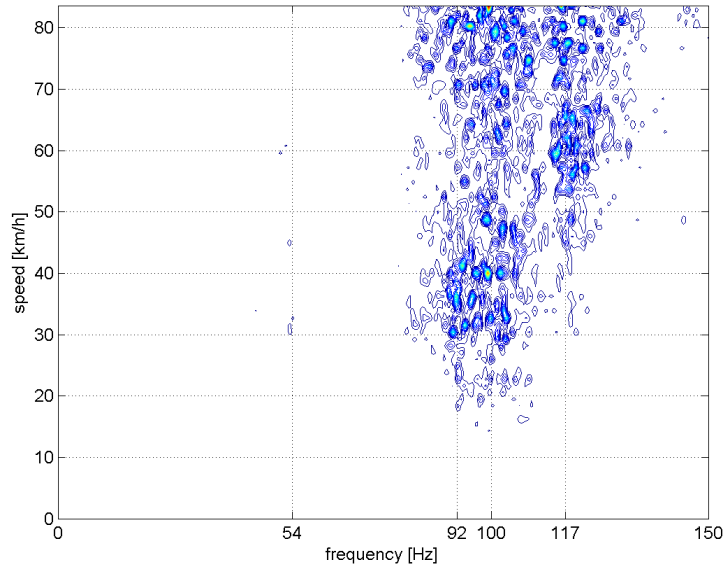


Figure 3.4: Campbell diagram in A-weighted Pa^2/Hz of the passenger's inner ear microphone

A-weighting on low frequency noise. All noise below 80 Hz is effectively removed, and the 80 Hz to 110 Hz region is more dominant than in the linear dB plot. Another area of noise around 100 Hz that is more obvious after A-weighting is between 20 km/h and 25 km/h.

Campbell diagrams give the data analyser a quick view of how a signal's frequency content changes with time but are not very useful for in-depth data analysis. To determine exactly what the contribution of each frequency band was to the overall interior noise, a frequency-based "Contribution analysis" (van der Linden and Varet, 1996; van der Auweraer *et al.*, 1997) was performed on the noise data. The noise signal was divided into frequency bands, namely 10 Hz to 50 Hz, 50 Hz to 150 Hz, 150 Hz to 300 Hz and 300 Hz to 500 Hz and the contribution of each frequency band (in dB and dB(A)) was calculated. The results are plotted in Figures 3.5 and 3.6.

In Figure 3.5, the 10 Hz to 50 Hz and 50 Hz to 150 Hz frequency bands clearly dominate the signal, and are consequently the cause of the "bumps" in the total SPL between 15 km/h and 21 km/h and between 31 km/h and 37 km/h. There is also a marked rise in the contribution of the 50 Hz to

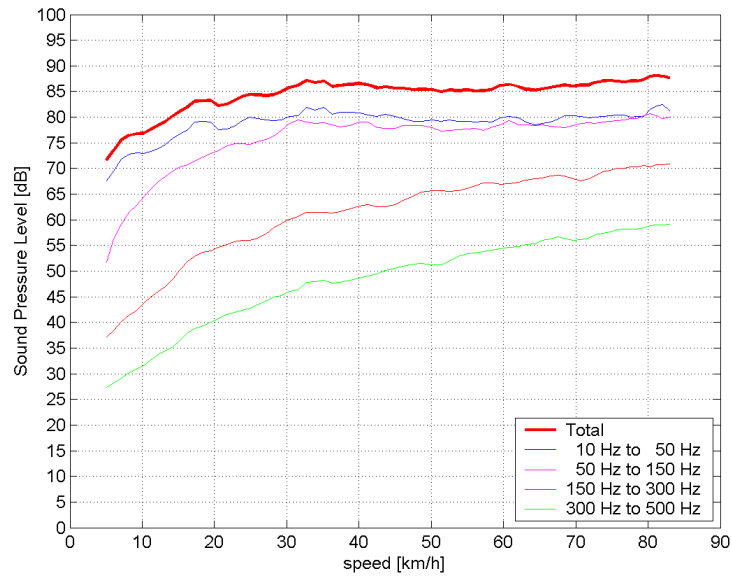


Figure 3.5: Frequency band contribution in dB of the passenger's inner ear microphone

150 Hz frequency band between 25 km/h and 31 km/h.

Once the signal is A-weighted, the frequency band contributions change significantly, as seen in Figure 3.6. Due to the strong penalising of low frequencies by the A-weighting method, the 10 Hz to 50 Hz frequency band now contributes much less to the total SPL. Up to about 45 km/h, the 50 Hz to 150 Hz frequency band contributes for all but 5 dB of the total recorded noise. The sharp rise between 25 km/h and 31 km/h in this second frequency band heavily influences the overall sound level. The higher frequency bands simply increase almost linearly with increasing speed.

The obvious question arising from these two figures was which one was to be trusted? The A-weighting method is used widely in industry so could not be ignored, but the directly measured sound levels showed a very big contribution from very low frequencies and could therefore not be totally ignored either. A meeting was held with employees of the vehicle's manufacturing company that participated in the development of the vehicle, where this concern was raised and the two SPL plots presented.

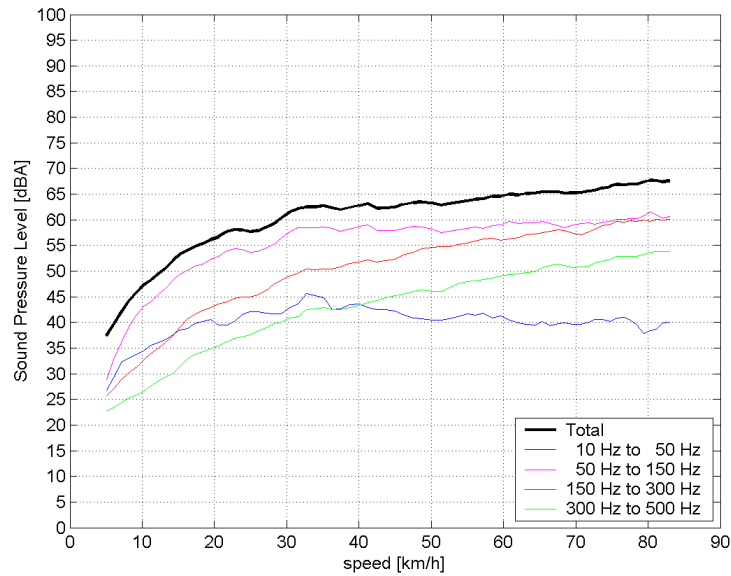


Figure 3.6: Frequency band contribution in dB(A) of the passenger's inner ear microphone

Two speed ranges and two contributing frequency bands were identified from Figures 3.5 and 3.6: 15 km/h to 21 km/h due to the 10 Hz to 50 Hz frequency band and 21 km/h to 35 km/h due to the 50 Hz to 150 Hz frequency band. An audio test was performed where the measured sound data was played to the employees, as well as the vehicle's test driver, followed by a filtered version of the sound data where only the 10 to 50 Hz and then the 50 Hz to 150 Hz frequencies were played. All parties unanimously agreed that the 50 Hz to 150 Hz frequency range was the problematic frequency and was perceived to contribute most to the overall noise level.

This showed that despite the high levels of low frequency noise, the A-weighting scale was legitimate as the frequency content of the noise perceived by a listening panel was more similar to the A-weighted version of the noise than the actual recorded version.

3.2 Acoustic Modal Analysis

The vehicle cabin was then analysed using Acoustic Modal Analysis (AMA) to determine its resonant frequencies. Experimental AMA was first performed, followed by analytical AMA.

3.2.1 Experimental Acoustic Modal Analysis

The vehicle cabin was instrumented with a binaural recording head on the driver's seat and a microphone at the passenger's inner ear. None of the microphones were directional. The cabin was excited using white noise with a maximum frequency of 500 Hz played by a 10 in dual-cone speaker located in the driver's footwell. 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. As the first acoustic mode was expected around 100 Hz, the coherence was excellent. Hendrickx *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 3.7) showed an acoustic resonance at about 100 Hz which was in the same range as the identified booming noise. The coherence levels were very good above 60 Hz.

3.2.2 Analytical Acoustic Modal Analysis

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 mesh size was refined until no noticeable difference in results between one mesh size and a finer one could be seen.

Concerns were raised regarding the legitimacy of the rigid boundary, non-absorbent material assumptions. The model was therefore analysed for three scenarios: firstly the entire cabin with all interior objects, secondly with the steering wheel removed, and lastly with the steering wheel and the seats removed. The results for the entire cabin and for the cabin with the steering

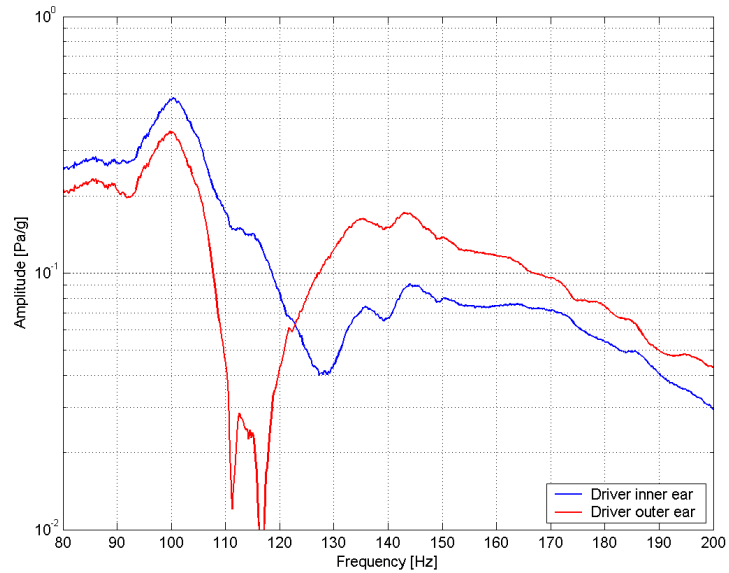


Figure 3.7: FRF for the driver inner and outer ear microphones during the experimental AMA test

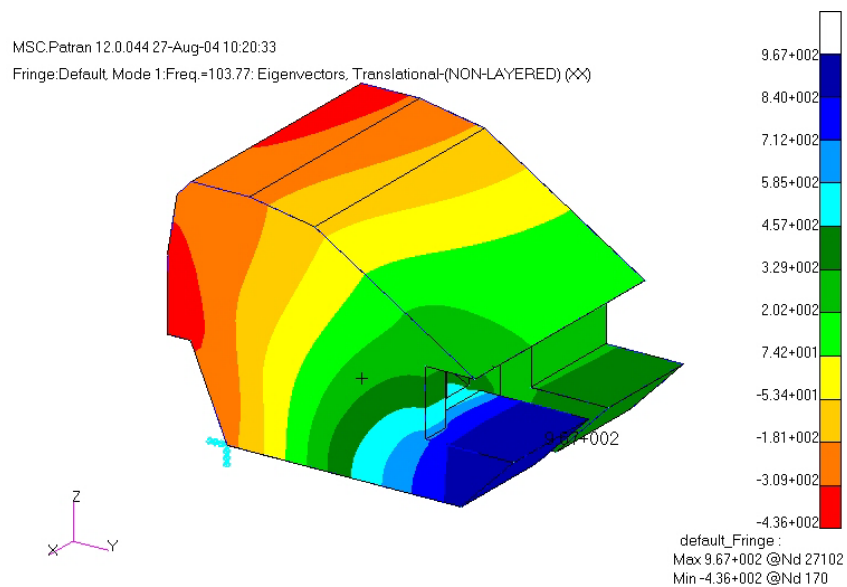


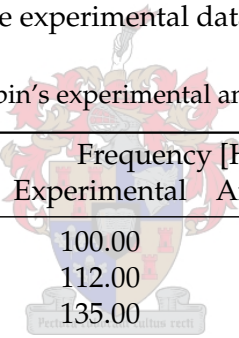
Figure 3.8: Predicted fundamental mode of the vehicle's cabin using analytical AMA

wheel removed were very similar as was expected, however the frequencies changed dramatically once the seats were removed. The natural modes predicted by the software for the entire (unaltered) cabin matched the experimental data's results best.

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 3.8. 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.

The analytical modes were very similar to the experimental results. Considering the simplifications made in the construction of the analytical model, the fundamental mode's frequency—as well as those of further modes—corresponded very well to the experimental data, as shown in Table 3.1.

Table 3.1: Comparison of cabin's experimental and analytical acoustic modes



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

3.3 Experimental Modal Analysis

The noise analysis in Section 3.1 showed the “booming” noise to have a frequency range from about 80 Hz to 110 Hz. The cabin's acoustic mode found during the AMA had accounted for the noise around 100 Hz but did not account satisfactorily for the noise below that frequency. It was postulated that one of the cabin's panels could be resonating near 90 Hz, accounting for much of the residual recorded noise. Experimental Modal Analysis (EMA) was performed to determine whether or not this was the case.

EMA was only carried out on one of the vehicle cabin's panels as fortunately the first panel tested yielded the required results. This panel (named

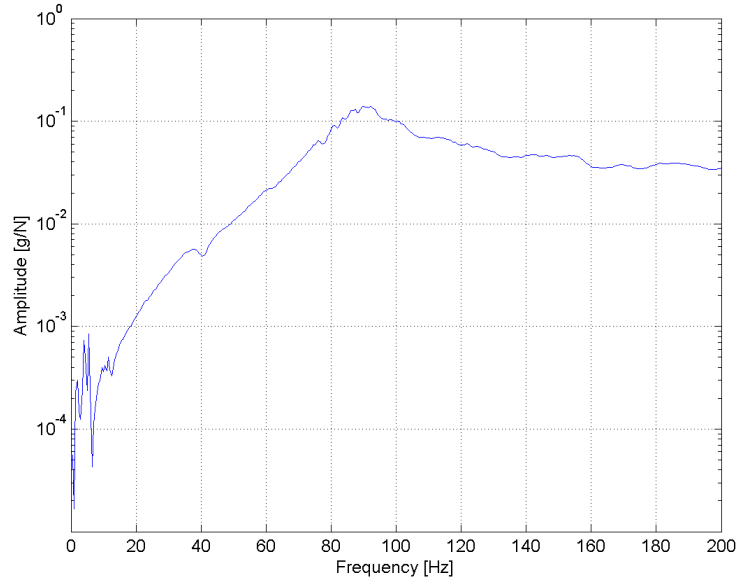


Figure 3.9: FRF for an accelerometer on the Back Panel

the “Back Panel”) was located at the end of the load box and directly behind the seats. It was chosen as it was the largest cabin panel and therefore the most likely to have modes at or below 100 Hz.

The panel was instrumented with seven accelerometers, six in a grid of 2 rows and the seventh at the point where the panel was excited by a modal hammer. FRFs were calculated between the force input by the hammer and the resulting accelerations measured by the accelerometers.

The FRF between the accelerometer on the bottom row in the middle of the panel and the force input by the hammer is shown in Figure 3.9. The dominating mode at about 90 Hz is obvious. Another noticeable property of the mode is its damped nature; the amplitude from 80 Hz to 100 Hz does not change significantly, with a slight increase at 90 Hz. This means that any acoustic amplification would occur over a wide (20 Hz) frequency range, fitting in well with the results of the Run-up test described in Section 3.1. The slight bump in the FRF at 100 Hz is due to the panel coupling with the cabin’s 100 Hz fundamental acoustic mode.

The experimental mode was then visualised using a “structural dynamics” software package (SD Tools v5.1). The mode was seen to be a “speaker”

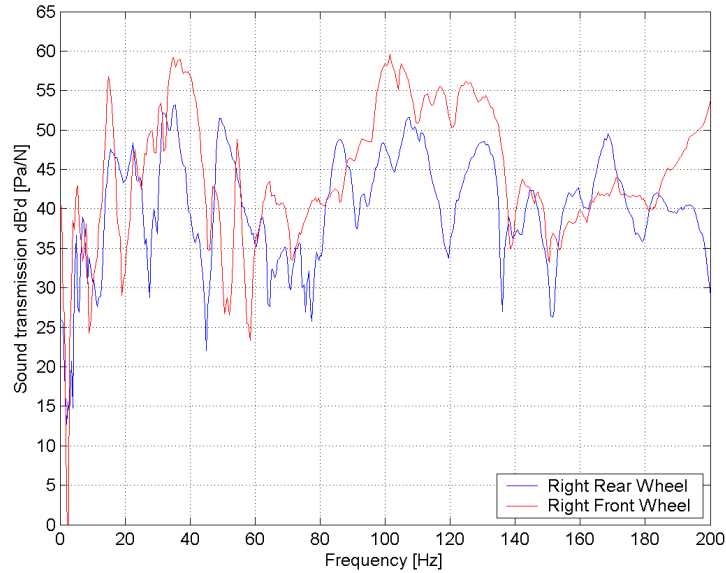
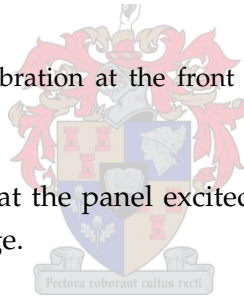


Figure 3.10: FRFs between vibration at the front and rear right wheels and the driver's inner ear

mode², convincing proof that the panel excited the acoustics well over the mode's wide frequency range.



3.4 Transfer Path Analysis

During the Run-up test (Section 3.1) the vehicle was excited by two phenomena. These were firstly the vibrational input at the wheels due to the tyre / road surface interaction and secondly the rotation of the four wheels, the front drive shafts and other components. These two phenomena needed to be separated and the contribution measured of each one to the interior noise. A Transfer Path Analysis (TPA) test was carried out on the vehicle to determine how effectively the vibrational input at the wheels coupled with the interior cabin noise.

The cabin interior was instrumented with a binaural recording head on the driver's seat and microphones at the passenger's inner ear and footwell.

²A "speaker" mode is a 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.

A shaker was used to excite each wheel in turn, after which FRFs were calculated between the force input at each wheel and the resulting sound pressure measured by the microphones.

The FRFs showed that the front and rear wheels – as well as the right and left side wheels – contributed to the interior noise at different frequencies. Figure 3.10 shows the FRFs between vibration input at the right front and right rear wheels, and the interior noise at the driver's inner ear. The difference between the two FRFs around the "booming" frequencies of 85 Hz to 105 Hz is clearly seen. The right front wheel creates much more noise from 90 Hz to 130 Hz than the right rear wheel, while the rear wheel shows a definite resonance at 85 Hz. It was speculated that the vibration at the rear wheels excited the fundamental mode of the back panel tested in Section 3.3 whereas the front wheels, being closer and more directly connected to the cabin, excited the fundamental acoustic mode of the cabin around 100 Hz.

3.5 Reciprocal test

Data from the accelerometers on the wheels recorded during the Run-up test showed that all the wheels vibrated unusually in the same speed and frequency ranges as the "booming" noise. The vibrations on the front right hub matched the interior booming noise's frequency signature better than the rear wheels. This is shown in Figure 3.11 where the sound pressure levels for the passenger's inner ear are plotted in light green and the vibrations levels on the front wheel hub are superimposed in colour. Comparing Figure 3.11 to Figure 3.3 (on page 24) the same "thumbprint" is seen between 80 Hz and 110 Hz and 25 km/h and 40 km/h. For this to have happened, something connected to the wheel could have been vibrating which in turn excited the noise in the cabin, the noise in the cabin could have been exciting something connected to the wheel, or a mixture of the two scenarios was possible. The rear wheels showed increased vibration in the same region, but only from about 100 Hz to 110 Hz.

To determine whether the noise in the cabin was exciting something connected to the wheels or whether it was the other way round, a test was performed using the principle of reciprocity. A speaker was placed at the driver's inner ear and accelerometers were attached to the four wheel hubs. The calculated FRFs showed that, while there was a good response at the

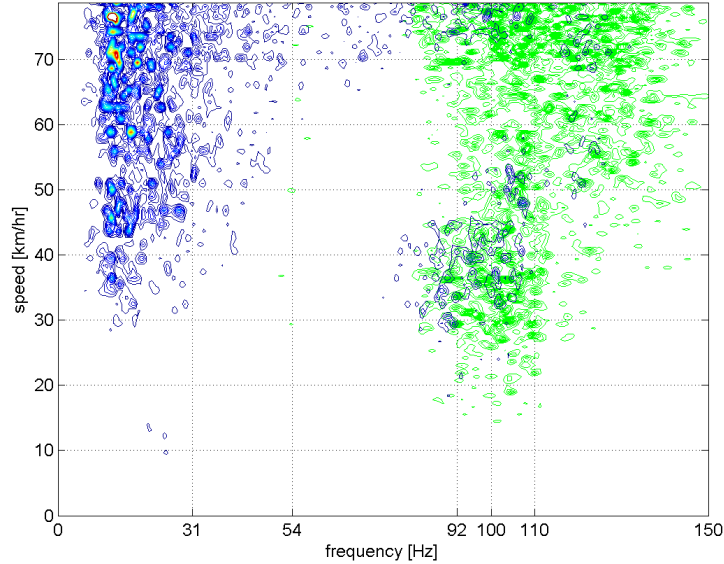


Figure 3.11: Campbell diagram of the vibration level of the front right wheel hub during the Run-up test (in full colour), overlaying the sound pressure at the passenger inner ear (in green)

wheels for noise at 100 Hz, there was very little measured by the accelerometers for any lower frequencies. The high response around 100 Hz can be clearly seen in Figure 3.12, the FRF between the noise input in the cabin and the resultant acceleration at the front left and right wheel hubs. The inability of the noise inside the cabin to excite the wheel hubs at frequencies lower than 100 Hz meant that any vibration measured on the wheel hubs below 100 Hz could not have been due to noise inside the cabin. It was possible though for noise in the cabin to excite the rear wheels in the manner that was recorded since there were only vibrations recorded between 100 Hz and 110 Hz. This was the region where noise inside the cabin excited the accelerometers attached to all four wheels.

3.6 Conclusion

The noise in the vehicle that had been identified by the vehicle manufacturer as the “problem noise” was identified as a low frequency road rumble in

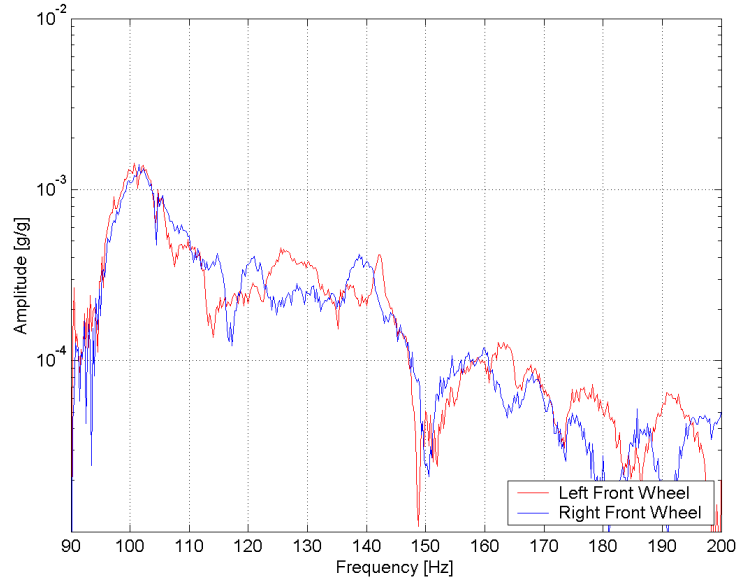


Figure 3.12: FRF for the front left and right wheel hubs due to reciprocal excitation

the frequency range of 80 Hz to 110 Hz between 25 km/h and 40 km/h. This “booming” noise was caused by bodywork vibrations and not by engine vibrations.

Two phenomena were found to be most responsible for converting vibrations in the bodywork into the booming noise in the cabin. These were the cabin’s primary acoustic mode at 100 Hz and a “speaker” mode of a large panel in the cabin behind the driver and passenger’s seats (the so-called “Back Panel”) at around 90 Hz.

Further testing showed firstly that vibrational excitation at the wheels was not exclusively responsible for the vibrations in the bodywork that lead to the booming noise. It also showed that the vibrations measured on the front right wheel hub showed the same “thumbprint” of vibration from 80 Hz to 105 Hz and from 25 km/h to 40 km/h, and that these vibrations on the hub could not have been generated in a reciprocal manner by the noise in the cabin. In other words the noise in the cabin could not have excited the wheel hub by exciting the vehicle’s body work and subsequently the wheel hub at any frequencies below 100 Hz.

An in-depth investigation would be needed into the sources of the vibra-

tion in the bodywork that caused the boom, as well as into the contribution of the Back Panel's noise to the cabin's booming noise. This work is detailed in Chapter 4, "Problem Identification".



Chapter 4

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.

4.1 Back Panel tests

The main speaker was identified during the Vehicle Analysis (Section 3.3) as the panel directly behind the driver’s seat at the front of the load box (the “Back Panel”). This 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.

4.1.1 Sandbag test

The vehicle was tested under the same conditions and on the same road as for the Run-up test in Section 3.1 except this time with sandbags in the vehicle’s

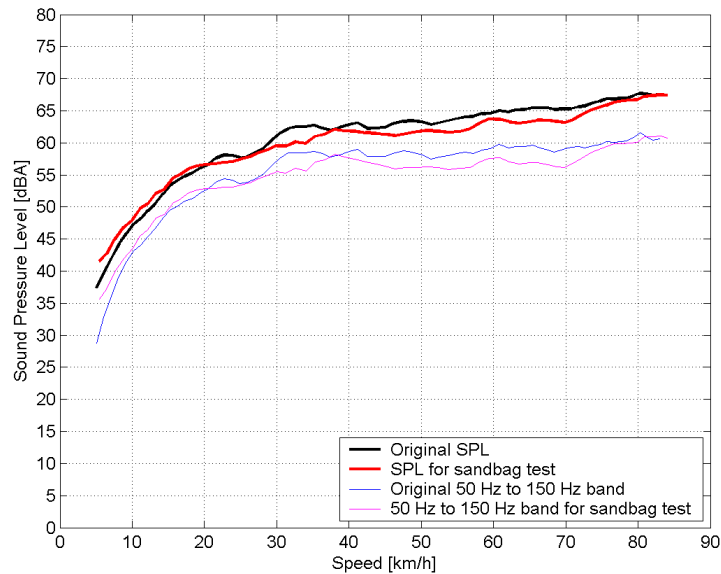


Figure 4.1: SPL levels at the passenger inner ear during the Sandbag test

load box pushed up against the Back Panel. The sound pressure inside the cabin was recorded using a binaural recording head on the passenger's seat.

Figure 4.1 shows a definite difference in the Sound Pressure Level (SPL) at the passenger's ear compared to the unmodified vehicle. The increase at 20 km/h and especially the dramatic, non-linear increase between 26 km/h and 32 km/h was reduced while the overall SPL increase was more linear with speed. The overall SPL remained much the same with only a 2 dB improvement recorded above 40 km/h. A normal human ear would not notice this difference. The test driver remarked on an improvement in the booming noise at low speeds. This is seen in Figure 4.1 where the 50 Hz to 150 Hz frequency band, which has much the same shape as the total SPL line, is more linear below 40 km/h and remains about 3 dB below the original level until 70 km/h. This difference is considered to be "just perceptible" (van Niekerk, 2002) to the human ear. It must be noted though that the sandbags only partially modified the dynamics of the panel. A total redesign of the panel could result in a greater reduction than was possible in this test.

Changing the design of this panel could therefore improve the SPL inside the vehicle, especially in the booming frequency band of 50 Hz to 150 Hz but

would not eliminate the problem. The primary acoustic mode of the cabin at 100 Hz would still contribute enough to the noise levels for the boom to be significant. Fixing the so-called “speaker” was therefore not sufficient to remove the booming problem from the vehicle, although it could reduce the booming noise in the cabin by a noticeable amount.

4.1.2 Sound deadening pads test

For this test, the entire inside surface of the Back Panel was covered with commercial sound deadening pads. The pads were 550 mm long by 190 mm wide by 1,5 mm thick and weighed 250 g each. In total, 6 pads were affixed to the Back Panel, adding 1,5 kg to the weight of the panel. The pads were made of a dense, pliable material: adhesive applied to one side and a matt finish to the other.

The vehicle was tested in precisely the same conditions as during the Run-up test. Since the test was performed after the drive shafts were removed and replaced for the tests described hereafter in Section 4.2, a run was first performed without the sound deadening pads to get a new reference value for the unmodified vehicle. The test with the pads affixed was then performed directly afterwards.

The pads “deadened” the sound primarily by adding mass to the structure (about 1,5 kg of material was added to the panel) which lowers the natural frequency of the structure, since the natural frequency of the structure

$$\omega_n \propto \sqrt{\frac{k}{m}} \quad (4.1.1)$$

where k and m are the stiffness and mass of the structure respectively. Newton’s famous equation (simplified for constant mass systems)

$$F = m \cdot a \quad (4.1.2)$$

dictates that if the force input into the structure remains the same and the mass of the structure is increased, the level of acceleration must decrease. The natural frequency of the structure should therefore decrease, along with the levels of acceleration. Less acceleration means less air is excited by the panel, which means less noise heard by the occupants in the cabin. It remained to be seen whether the accelerations could be attenuated enough to make a

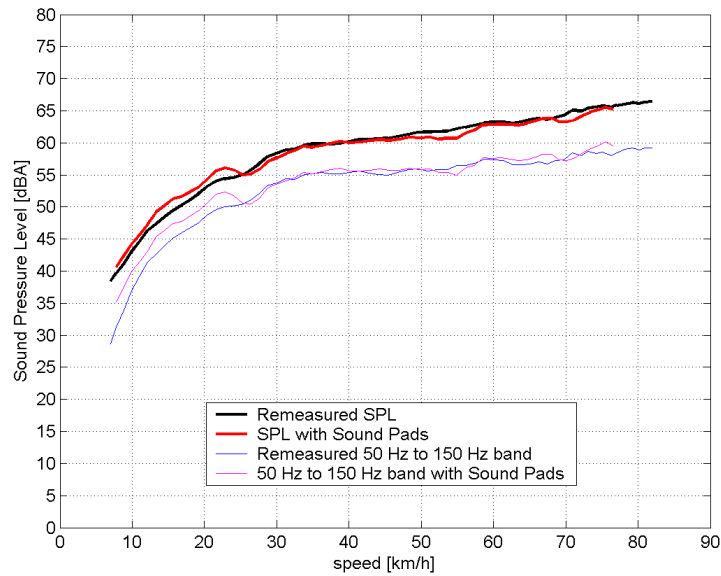


Figure 4.2: SPL levels at the passenger inner ear during the sound deadening pads test

noticeable difference to the noise level in the cabin.

After the test, little difference in the sound levels was noticed both by the test driver and when looking at the test data in Figure 4.2. The noise levels remain much the same over the entire speed range. The added mass therefore could not make enough difference to the dynamics of the Back Panel to reduce the interior noise by any noticeable amount.

This test showed conclusively that such a post-production “fix” would have no effect on the interior cabin noise.

4.2 Drive shaft assembly tests

During the Run-up test (Section 3.1) it was seen that the front wheel vibrated unusually and showed the same vibration “thumbprint” as the vehicle’s booming noise (shown in Figure 3.11 on page 34). 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.

The Transfer Path Analysis (TPA) test (Section 3.4) showed that the front

wheels contributed more directly to the cabin sound pressure from 90 Hz to 130 Hz than the rear wheels. It was also seen that the right front wheel contributed more than the left front wheel. Since the only difference between the left and right front sides of the vehicle's wheel and suspension assembly was the drive shaft assembly, it was decided to investigate the influence the right side drive shaft assembly had on the interior noise.

This influence was tested using two tests: an Experimental Modal Analysis (EMA) of the right drive shaft assembly and an operational test of the vehicle with both left and right side drive shaft assemblies removed.

4.2.1 EMA of the right drive shaft assembly

Before an operational test could be conducted on the drive shafts, a basic EMA of especially the right drive shaft assembly was conducted to give some insight into their vibrational characteristics.

It is important to note that the drive shaft assemblies are not symmetrical. The left side drive shaft connects the wheel bearing directly to the gearbox, with Constant Velocity (CV) joints at either end. The CV joint at the wheel hub is a ball joint, while the CV joint at the gearbox is a tripod joint. Since the engine is mounted transversely in the engine bay, the gearbox does not lie in the centre of the vehicle but instead closer to the left front wheel. There is consequently an "Extension Shaft" that connects the standard length drive shaft on the right hand side to the gearbox. This shaft is connected to the standard right drive shaft via a tripod CV joint. It is supported next to this CV joint in a bearing that is mounted in a bracket and bolted to the engine housing. The standard drive shaft will be referred to as the "Outer Shaft".

The right side drive shaft assembly was instrumented with six accelerometers: three on the Extension Shaft and three on the Outer Shaft. The shafts were excited using a shaker connected to the Outer Shaft next to the wheel hub, directly below one of the accelerometers. A random signal with a bandwidth of 200 Hz was used as input and measured using a force transducer. Frequency Response Functions (FRFs) were calculated between the force input at the shaker and the acceleration responses at the six accelerometers.

The FRFs showed the right hand side drive shafts to have a significant mode at about 83 Hz. Figure 4.3 shows the FRFs of the six accelerometers. The mode at 83 Hz is clearly visible. It is quite a damped mode (as expected since there are two CV joints and a bearing connected to the shafts) which

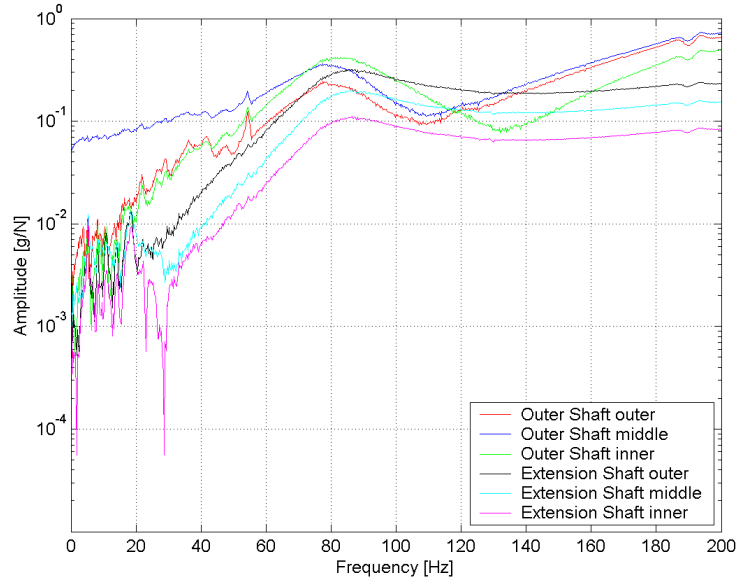


Figure 4.3: FRFs for the right hand side drive shaft assembly

means that there is almost as much vibration in the shaft at the frequencies close to the resonant frequency as there is at that frequency. It would therefore result in a broad excitation of connected components if it were to be excited. Note also the different FRF shapes for the two shafts.

It was important to know the shape of the resonant mode at 83 Hz so the FRFs were imported into a structural dynamics software package. The mode shape was shown to be a rigid body mode, as expected. What was not expected was that the *entire* shaft assembly (i.e. both the Extension and the Outer shafts) moved with all points in phase. The point that experienced the greatest displacement was at the bracket that connected the Extension Shaft to the engine housing i.e. at the midpoint of the shaft assembly, as seen in Figure 4.4. 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.

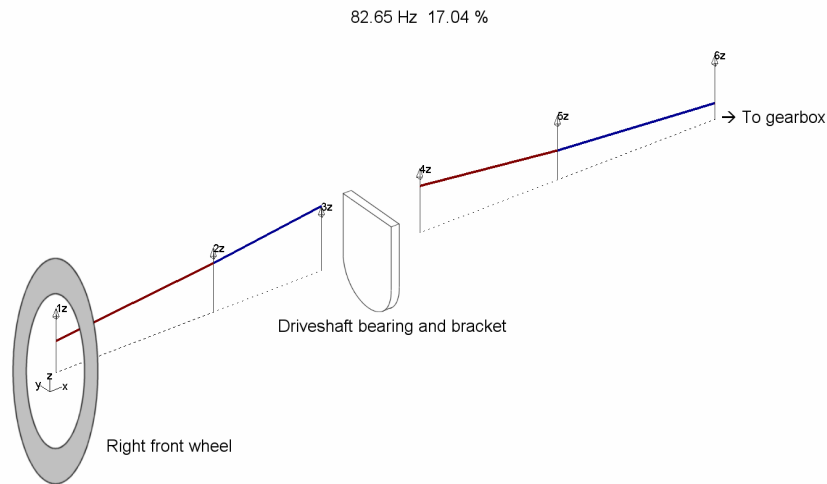


Figure 4.4: Schematic diagram of the right hand side drive shaft assembly, showing the first mode shape

4.2.2 Operational test

The TPA test in Section 3.4 showed no conclusive proof that the vibration inputs at the wheels were solely responsible for the booming noise in the cabin. The only other sources of excitation during the Run-up test were the rotating elements in the vehicle. These were the wheels, the front drive shaft assemblies, the front differential and the final drive gears and shaft. There was no feasible way to test these components under operating conditions in the laboratory so instead the vehicle was tested under the same conditions as the Run-up test, with its drive shafts and CV joints removed.

Accelerometers were placed on the front left and right wheel hubs as well as on the Back Panel (the panel identified as the “speaker” in Section 3.3). The binaural recording head was once again put on the passenger’s seat to record the interior SPL. All testing conditions, including the stretch of road used, were the same as for the Run-up test.

A marked decrease in road rumble was noticed both by the test driver and afterwards during the data processing. Figure 4.5 shows the SPL levels

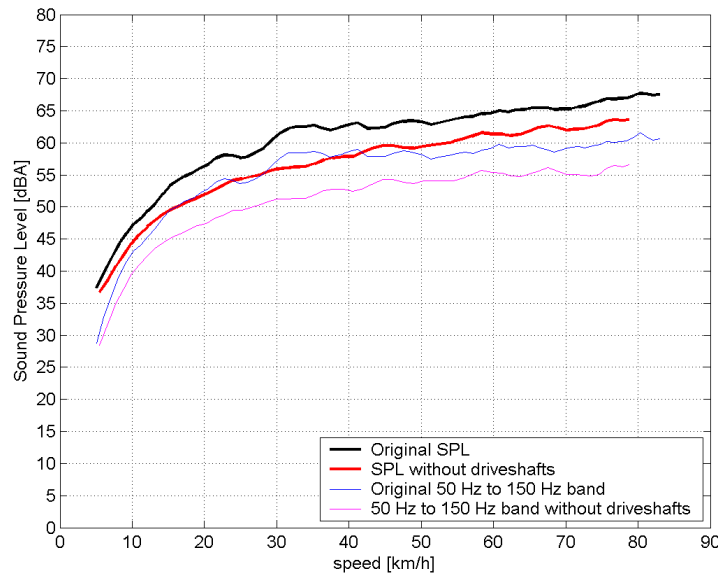


Figure 4.5: SPL plot for the vehicle with and without drive shafts

as recorded by the binaural head's inner ear microphone for the vehicle with and without drive shafts. It also shows the SPL of the 50 Hz to 150 Hz frequency band for the same conditions. The new SPL line is much smoother, increases much more linearly with speed (especially above 15 km/h) and is 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 is a major improvement.

The Campbell diagram in Figure 4.6 shows the noise intensity for the test across the 0 Hz to 150 Hz frequency range. The original levels from the Run-up test are plotted in green with the levels for the vehicle without drive shafts superimposed in full colour. Both of the noise levels are A-weighted. It is clear that the 92 Hz noise was all but removed and the 100 Hz noise was significantly reduced. The “thumbprint” of noise around 20 km/h also disappeared.

The data showed conclusively that the drive shafts contributed greatly to the booming noise in the vehicle at all speeds, but noticeably from 20 km/h to 40 km/h. The most noticeable difference in sound pressure levels was in the 80 Hz to 100 Hz range.

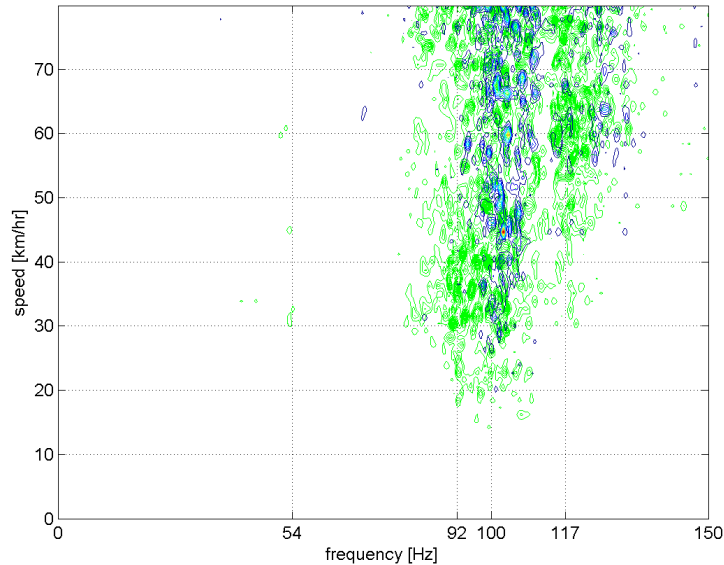


Figure 4.6: Campbell diagram for the vehicle without drive shafts (in full colour), overlaying the sound pressure for the vehicle with drive shafts (in green)

4.2.3 Vibration transmission test

The drive shaft assembly's EMA test in Section 4.2.1 showed that the CV joint and bearing experienced high levels of vibration around 83 Hz. What was not known was the level of vibration experienced by the engine due to this excitation. A vibration transmission test was therefore carried out to determine the magnitude and nature of the displacement of the engine when the right drive shaft assembly was excited at 83 Hz.

The drive shaft assembly was instrumented identically as it was for its EMA test. Accelerometers were also placed on the engine in four positions and on the bracket fastening the Extension Shaft to the engine. The assembly was again excited using a random signal with a bandwidth of 200 Hz.

The engine was shown to vibrate very little (more than 10 times less than the drive shafts at 83 Hz) although the engine's FRFs did show a resonance at 83 Hz. The engine also vibrated exactly 180° out of phase with the shafts, meaning that when the shafts moved upwards, the engine moved downwards and *vice versa*. An investigation showed the bearing on the Extension

Shaft to be mounted inside a compliant rubber housing inside the bracket. This explained the dramatically reduced amount of vibration in the engine compared to that in the shafts. It also explained why the engine could move out of phase with the shafts at 83 Hz.

The results of this test gave no conclusive indication as to the transfer path used by the vibrations to get into the vehicle's bodywork. There was no reason to conclude that the right drive shaft assembly was not at least partially responsible for the vibration excitation in the vehicle that lead to the booming noise in the cabin between 25 km/h and 40 km/h.

4.3 Operational Transfer Path Analysis

One reason put forward as to the lack of conclusive results in Section 4.2.3 was that the drive shafts were being excited by a shaker, and not due to rotational excitation. It was seen in Section 3.4 that vibrational excitation at the wheel hubs was not sufficient to excite the booming noise in the cabin as much as it was during an operational test. Another operational test was therefore conducted to measure the amount of vibration experienced by the front wheel hubs, the engine, the front suspension and the bracket fastening the right drive shaft assembly to the engine. The necessary equipment (laser sensors) was not available to measure the vibration levels on the drive shafts while rotating.

This test was in essence a simplified TPA test. The contributions of the vibrations through specific identified paths from the drive shafts to the vehicle's body were compared. The identified paths were:

1. from the drive shafts, through the wheel bearing, through the wheel hub, through the suspension, to the vehicle body;
2. from the drive shafts, through the drive shaft bearing and bracket, through the engine, through the engine mounts, to the vehicle body.

Since the engine would again idle during the operational test, measurements were first made of the vibrations at the accelerometers' locations for the engine idling while the vehicle was at rest. Any additional vibrations would then be as a result of the vehicle's motion while travelling down the road.

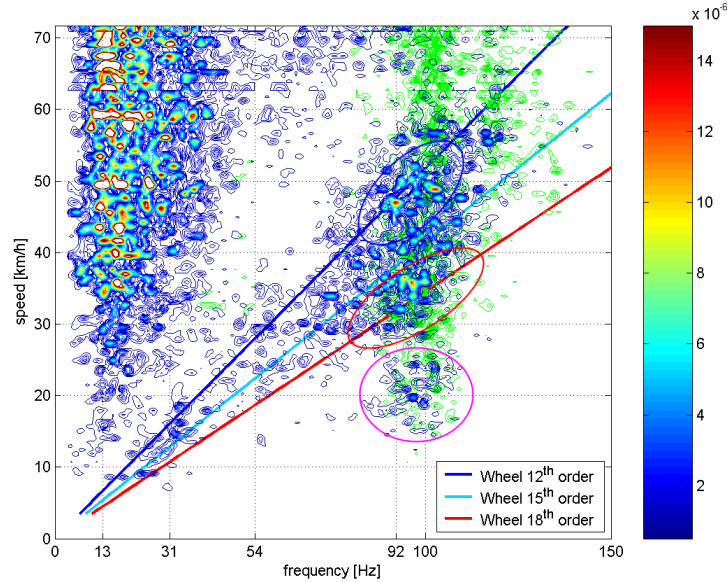


Figure 4.7: Campbell diagram of the vibration level of the right wishbone's outer swivel connection (in full colour), overlaying the sound pressure at the passenger inner ear (in green)

The idling engine's second order of 31 Hz dominated the measurements by the accelerometers on the engine; the engine's idling speed is around 950 rpm. A smaller resonance was seen at 12 Hz. This frequency was already identified by the vehicle manufacturing company and as a result the engine mounts had been optimised for this frequency. No other noticeable resonances as a result of the engine's vibrations while idling were recorded.

The data from the test confirmed that the left and right hubs experienced vibrations between 20 km/h and 40 km/h and between 80 Hz and 105 Hz that matched the booming noise's frequency pattern. In addition it was shown that the left and right suspension wishbones also experienced this vibration; the right side's vibration was at all times greater than the left side's vibration. Figures 4.7 and 4.8 show the Campbell diagrams for the vibrations in the right suspension wishbone at its outer and inner swivel joints respectively. The excited region in question can be clearly seen. The vibrations are once again superimposed over the A-weighted noise signal measured at the passenger's inner ear. The areas in ellipses are discussed later in this section.

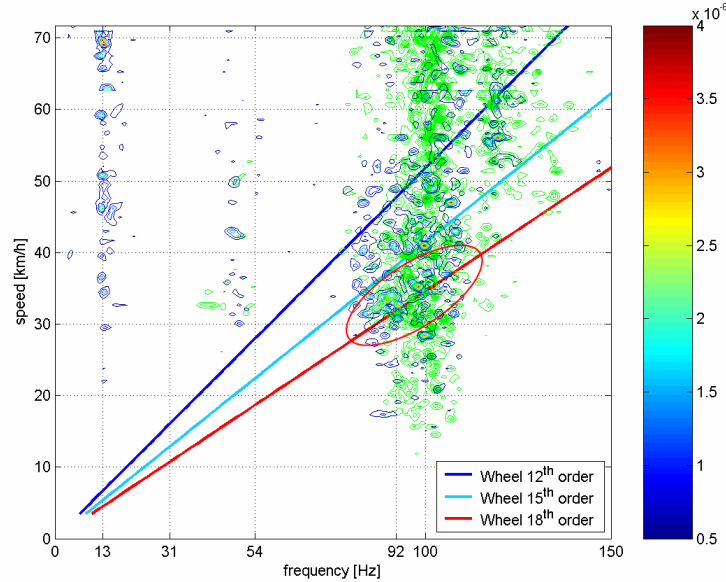


Figure 4.8: Campbell diagram of the vibration level of the right wishbone's inner swivel connection (in full colour), overlaying the sound pressure at the passenger inner ear (in green)

An accelerometer placed next to the point where the right wishbone was bolted to the vehicle chassis, showed the transmission of the vibrations from the suspension to the chassis. Figure 4.9 shows the vibrations measured at this point and the repetition of the vibration “thumbprint” between 20 km/h and 40 km/h and between 80 Hz and 105 Hz. The levels of vibration are less in the chassis than in the suspension, but only slightly. The vibrational amplitudes in the chassis were in the order of $1 \times 10^{-3} \text{ m/s}^2$ whereas amplitudes at the wishbone's inner joint were in the order of $1 \times 10^{-2} \text{ m/s}^2$. Note that the units in a Campbell plot for accelerations are those used for the Power Spectral Densities that form the contours ($(\text{m/s}^2)^2/\text{Hz}$) and not in m/s^2 . The reason for this good vibrational transfer can be largely attributed to the stiff bushings used by the motor manufacturing company to give the vehicle a more “sporty” feel.

The next task was to find whether there was a panel in the vehicle cabin responsible for exciting the 100 Hz acoustic mode. The Back Panel, already identified in Section 3.3 as the main cabin speaker for noise around 92 Hz,

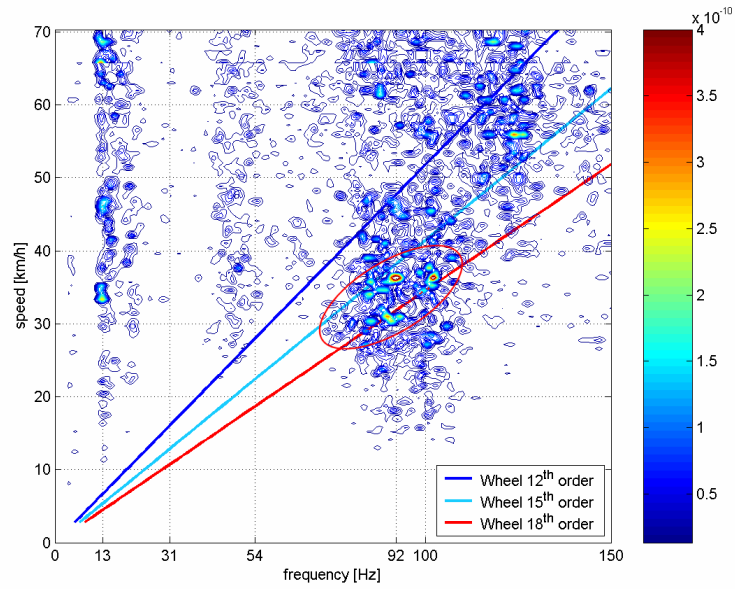


Figure 4.9: Campbell diagram of the vibration of the vehicle chassis near the right wishbone

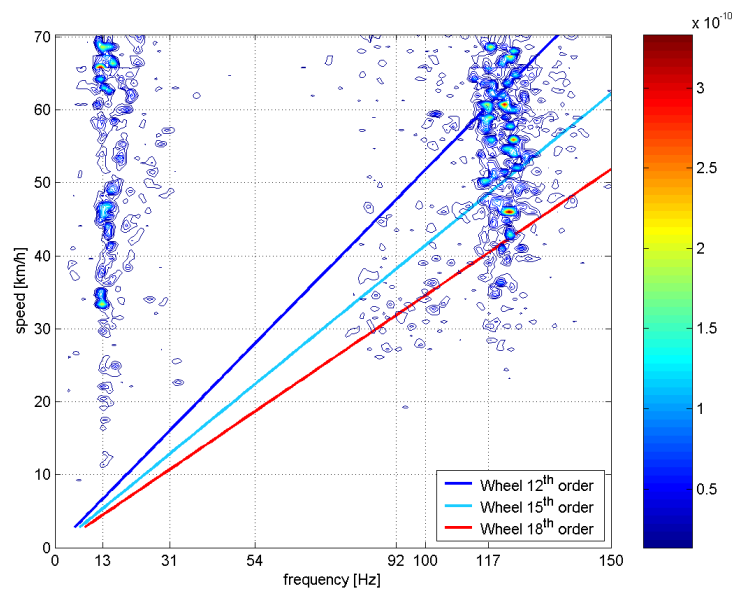


Figure 4.10: Campbell diagram of the vibration level of the driver's footwell floor

was the most likely culprit. With a natural, damped “speaker” mode at 92 Hz that still vibrated almost as much at 100 Hz, it was very likely that it was excited by the vibrations in the chassis floor and resonated around 92 Hz, generating a large part of the booming noise due to its own speaker-like properties and by exciting the acoustic mode in the cabin. Concerns were raised that modifications to the Back Panel—even effectively damping out all movement in the panel in Subsection 4.1.1—did not make a marked difference to the interior SPL. It was therefore decided to search for another resonating panel.

Since the right wishbone was bolted to the chassis close to the cabin floor, accelerometers were placed on the cabin’s left and right floor panels. Very little correlation was found between the vibrations around 100 Hz in the chassis floor and the cabin floor (Figure 4.10). This may be due to the fact that the velocity of the air in a standing wave is a minimum at its ends; the floor panel is situated close to the end of the wave. High levels of vibration were instead seen around 120 Hz especially from 40 km/h. During the EMA of the right drive shaft assembly the right tyre was found to have a resonance at 117 Hz. This resonance may have been a source of these vibrations. The windscreen was also seen during the Run-up test to have the “thumbprint” of vibration around 100 Hz but was not seen as a modifiable vehicle component. It seemed therefore that rather than one identifiable panel vibrating and generating the booming noise, many panels in the cabin vibrated with varying intensities and, together with the cabin’s acoustic mode, caused the interior booming noise.

It was confirmed therefore that the vibrations in the drive shafts around 80 Hz to 100 Hz caused the vibrations in the vehicle’s bodywork that led to the booming noise. What was not known was the mechanism that excited the drive shafts at these frequencies.

4.3.1 Constant Velocity joint investigation

The only transiently changing characteristic of the drive shafts was the rotational speed. Since the Outer Shafts had CV joints at both ends, it was necessary to investigate the possible role that the rotation of the CV joints could have on the vibrations in the drive shaft assemblies. Biermann (2004) explains that drive shaft assemblies with CV joints will usually show sensitivity to the rotational orders that match the types of CV joints used. This

vehicle's drive shaft assemblies, as described in Subsection 4.2.1 (page 41), have ball CV joints on the gearbox sides of the Outer Shafts and tripod CV joints on the wheel sides of the Outer Shafts. Due to the so-called "plunging" forces inside the CV joints, the joints are sensitive to orders that are multiples of the number of balls in the joint. Tripod joints are therefore sensitive to every third order (the third, sixth, ninth, and so on) and ball joints are sensitive to every sixth order (the sixth, twelfth, eighteenth and so on).

When overlaying the vibration measurements on the wishbone with the wheel's twelfth and eighteenth orders (which are the ball joint's second and third orders and the tripod joint's fourth and sixth orders) it is quite clear what the source of the vibration around 100 Hz is. Both Figure 4.7 and Figure 4.8 show these twelfth and eighteenth orders, as well as the areas that are of specific concern. In Figure 4.7 it can be seen that the ball joint's second order, together with the tripod joint's fourth order, excite the drive shaft assembly at 45 km/h to 52 km/h (blue ellipse). However, these vibrations don't transmit to the inner joint very well, as seen in Figure 4.8. The vibrations that do transmit well are those due to the third order of the ball joint and sixth order of the tripod joint exciting the drive shaft assembly at 30 km/h to 40 km/h, shown in the red ellipses in both figures. A general area of excitation is seen in Figure 4.7 in the sector described by the two wheel order lines, suggesting that the drive shafts are generally excited at the two wheel orders by the two CV joints' resonances at all speeds but only cause a resonance around their first mode shown in Section 4.2.1 to be from 80 Hz to 100 Hz. The tripod joint's fifth order (relating to the wheel's fifteenth order) is also shown in the figures and correlates with higher levels of vibration at 100 Hz, although it is not as pronounced as when the two CV joints' orders coincide.

4.4 Conclusion

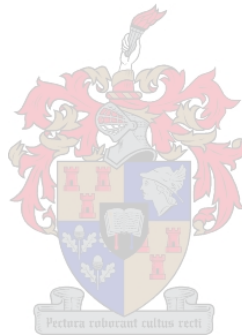
The identified source of the vibrations that lead to the booming noise in the cabin was identified as the drive shaft assemblies, with the right side assembly contributing more than the left. These vibrations were caused by the wheel's rotational speed coinciding with the rotational orders of the ball and tripod CV joints. The vibrations were then shown to enter the cabin through the suspension wishbone, which was bolted onto the vehicle's chassis. While not exclusively tested, the Back Panel was deemed to be the panel most re-

sponsible for creating the booming noise in the cabin from 80 Hz to 105 Hz. The acoustic mode in the cabin at 100 Hz amplified all the noise generated by the Back Panel around that frequency.

It was also seen that a post-production “fix” of affixing “sound deadening pads” to the Back Panel would not improve the overall SPL by an amount noticeable by the human ear. A redesign of the Back Panel would only result, at best, in a 3 dB reduction in the booming noise.

The cabin noise—in particular the booming noise—could only be noticeably reduced by reducing the vibrations at the source (the drive shafts), by modifying the main vibrational transfer path through the suspension wish-bones, or by modifying the cabin to shift its acoustic mode of 100 Hz.

These possible solutions are discussed in more detail in Chapter 5, “Proposed Solutions”.



Chapter 5

Potential Solutions

This chapter discusses possible measures to reduce the booming noise in the tested vehicle, as well as in Light Utility Vehicles (LUVs) in general. Since none of these concepts were tested, they remain potential solutions.

The chapter deals first with potential solutions for the tested vehicle, and moves on to potential solutions to the booming caused by the fundamental acoustic mode in LUVs.

5.1 Vibration attenuation at the source

The most effective solution to a Noise, Vibration and Harshness (NVH) problem is to modify the source of the vibration. This is not always achievable but the possibility must be investigated and only abandoned if found to be impractical or even impossible.

The source of the vibrations in the vehicle was found in Section 4.2 to be the natural frequency of the drive shaft assemblies—especially the right side assembly—when excited by the rotational orders of the ball and tripod Constant Velocity (CV) joints. To reduce the vibrations at the source, either the drive shaft assemblies or the CV joints would need to be modified.

5.1.1 Right drive shaft assembly modification

Since the right drive shaft assembly was seen to contribute more to the internal cabin noise, the initial focus would be on modifying its structure. A modification of the assembly would be concentrated on shifting the mode at 83 Hz away from the resonant frequency of the Back Panel and the fun-



Figure 5.1: Right drive shaft bearing housing and bracket

damental acoustic mode of the cabin. Shifting the frequency up would just move it closer to the Back Panel and cabin's modes, so the assembly's mode should rather be shifted down. It must be remembered that the mode shown in Figure 4.3 on page 42 is very damped so to reduce the vibrations in the drive shaft assembly, the mode would have to be moved down by at least 10 Hz but preferably by 20 Hz. If the frequency were to be raised, it would have to be by at least 20 Hz. If it is raised by less than that, the mode would then just coincide even better with the acoustic mode of the cabin and the speaker mode of the Back Panel.

The main reason for the high levels of vibration at the 83 Hz mode was found to be the compliant rubber in which the bearing at the end of the Extension Shaft was housed (see Figure 5.1 and Figure 4.4 on page 43). By stiffening this rubber mounting, the vibration levels in the shaft would decrease, resulting in less vibration making its way into the vehicle's bodywork via the suspension. Since a great deal of the motion of this mode occurs at the bearing, it is a very effective place to change the stiffness of the right drive shaft assembly.

It was seen in Subsection 4.2.3 that very little of the vibration in the right drive shaft assembly made its way into the engine. This meant that a re-design of the bracket that fastened the Extension Shaft to the engine would not reduce the booming noise in the vehicle, as the engine was not part of a transfer path of the vibrations.

5.1.2 General drive shaft assembly modification

To ensure symmetry in the vehicle, any other modifications on the right drive shaft assembly would have to be repeated on the left drive shaft assembly as well. This would include modifying the Outer Shafts (the standard drive shafts) or the CV joints on both sides.

It is unlikely that the CV joints could be modified as they are standard parts and moreover the ball and tripod joints will always be sensitive to the twelfth and eighteenth wheel orders, the orders that cause the vibrations in the CV joints that excite the 83 Hz mode. Modifying the joints by using different ball configurations or similar techniques may just lead to vibration problems at other speeds.

Modifying the Outer Shafts would probably make very little or no difference to the vibrations in the drive shafts. The mode at 83 Hz does not depend on the Outer Shafts' properties as much as it does on the compliance of the rubber bearing housing. Figure 4.4 shows that the Outer Shaft does not bend at 83 Hz. The weight of the shaft could be increased to lower the frequency of the mode, since more mass means a lower natural frequency (see Equation 4.1.1 on page 39) but it is not likely that this would make enough of a difference to the mode to result in less vibrations entering the vehicle body-work via the wheel hub and suspension.

5.2 Transfer path modification

The main vibrational transfer paths were identified in Section 4.3. They ran from the drive shaft assemblies, through the wheel bearings, wheel hubs and wishbones to the vehicle chassis. The right side drive shaft assembly vibrated more from 80 Hz to 100 Hz than the left side assembly and was therefore identified as the principle transfer path. Once again, modifications would have to be made to both the left and right sides of the path to ensure a symmetrical suspension setup. Recalling Figure 3.11 on page 34, the right wheel

hub showed the same vibrational thumbprint as the thumbprint of the noise at the passenger's (and driver's) inner ear. If the vibrations at the wheel hub were to be isolated from the vehicle's bodywork, the booming noise in the cabin would be dramatically reduced.

The most readily modifiable components in the transfer path are the bushings used in the wishbone suspension setup. The motor manufacturing company admitted to using bushings that were harder than usual to assist in giving the vehicle "sporty" handling. Using softer bushings would decrease the amount of vibration transferred from the wheel hub to the wishbone and from the wishbone to the vehicle chassis. This should then reduce the booming noise in the cabin significantly.

Any other modifications of the transfer path would involve redesigns of vehicle components, a measure that is prohibitively expensive. It would also be extremely difficult to redesign for example the wishbones so that all frequencies between 80 Hz and 110 Hz are not transmitted from the outer to the inner swivel joints while still fulfilling its suspension role.

5.3 Vehicle cabin modification

It was seen in Section 4.3 that, other than the Back Panel, there was no panel that could be identified as a major contributor to the booming noise in the cabin. The Back Panel's damped mode at 92 Hz certainly contributed to the booming noise, however there was only a small reduction in the booming noise (only 3 dBA) when the panel was packed with sandbags. If the Back Panel were to be redesigned, a larger reduction could be possible as the sandbags only partially changed the dynamics of the panel. There was no noted difference in the cabin's Sound Pressure Level (SPL) when the panel was covered in commercial sound deadening pads. Since the other large panels in the cabin (the left and right floor panels) showed no excessive vibration around 100 Hz, it was assumed that many of the panels contributed together to excite the cabin's acoustic mode at 100 Hz. This would mean that the only way to directly reduce the booming noise in the cabin by modifying the vehicle cabin, would be to redesign the Back Panel. The reduction in interior noise could be even higher than the 3 dBA achieved during the sandbag test.

5.4 The LUV's fundamental acoustic mode

As discussed in Section 2.1, the fundamental acoustic mode in the cabin of a Light Utility Vehicle (LUV) can contribute significantly to acoustic booming. It is therefore a general problem that needs to be addressed along with a specific vehicle's NVH problem areas.

A number of concepts to improve or modify an LUV's fundamental mode are described below along with their advantages, disadvantages and possible cost implications to the vehicle manufacturing company.

The human ear, as described in Appendix B, does not hear all frequencies equally well; it is especially insensitive to low frequencies. For this reason it does not make much sense to change the acoustic mode to a higher frequency, even if the sound intensity is slightly less, as the human ear will hear the higher frequency better than the lower frequency. It is more advantageous to lower the acoustic mode's frequency where the human ear is less sensitive to noise.

5.4.1 Cabin modification

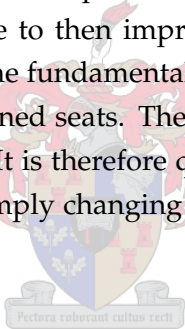
The most obvious way to lower the fundamental acoustic mode is to enlarge the acoustic cavity, i.e. to enlarge the vehicle's cabin. The sound's frequency is directly related to its wavelength so any extra length in the cabin would lower the resonant frequency. Unfortunately, this would mean changing the vehicle's entire design, something that the vehicle's manufacturing company would most probably never do. The issue of cabin size can only be addressed very early in the design phase of the vehicle, not as a design modification once production has begun. The cost of changing the cabin's size makes this solution impractical.

Another means of lowering the frequency is to lengthen the wavelength of the fundamental mode by changing the acoustic boundaries of the cabin. This technique has been applied in the field of speakers, especially low frequency speakers ("sub-woofers"). The length of the sound wave is increased by "folding" the speaker cone into a spiral, thereby yielding a long wavelength in a compact speaker box (Cerwin-Vega, 2004). The concept for the vehicle would then be to design a cavity inside the cabin that increases the wavelength of the fundamental mode, thereby lowering its frequency. This concept is feasible if space is allowed for it at either end of the cabin. The

top rear corner of the cabin would be the best position for such a cavity, so notionally a cavity with a “folded sound path” could be created behind the cabin. It also remains to be seen if the concept can be applied to lengthen the fundamental mode’s wavelength in an LUV.

5.4.2 Sound absorption

Another concept is to absorb the sound waves more effectively in the cabin. The best absorption occurs where the air speed of the mode’s sound wave is the greatest, which is at the centre of the wave: at the node. The best potential sound absorbers—the seats—are already installed; they must just be designed for maximum sound absorption. This includes using cloth seat covers (instead of polyester, plastic or leather) and using open cell rather than closed cell foam for the seat padding. It is well known that vehicles with leather seats have more NVH problems than the same vehicles with cloth seats. It is quite feasible to then improve the vehicle’s overall NVH performance, and not just at the fundamental acoustic mode’s frequency, by using better acoustically designed seats. The tested LUV had polyester seat covers with closed cell foam. It is therefore quite possible that the booming noise would be reduced by simply changing the seat covers and the type of foam used in the seats.



5.4.3 Active control

The Helmholtz resonator has been used for many years to cancel out a noise at a certain frequency. However there are disadvantages to using Helmholtz resonators in vehicles. First of all, the targeted frequency is inversely proportional to the square root of the resonator’s cavity volume, resulting in extremely large cavities for low frequencies (Kashani and Orzechowski, 2001). Secondly, the resonator must dissipate enough energy for the mode to be noticeably damped. This requires a certain amount of friction in the flow at the neck of the resonator, something that is not always possible. Finally, the resonator can only be tuned to a single frequency, which makes its effectiveness questionable considering the amount of space it requires to operate effectively.

This problem lead to the concept of active control of the interior noise. The two active noise control methods currently in use are Active Noise Con-

trol (using a microphone and speaker setup) and Active Structural Acoustic Control (ASAC) (using a control actuator attached to the vehicle's bodywork). Dehandschutter & Sas (1998) recommend that for frequencies below 300 Hz to 400 Hz ASAC should be the method of active control as most noise at such low frequencies is transmitted through the vehicle's bodywork. Such a system could be installed at the points where the suspension components are bolted to the chassis. This would cancel the vibrations in the cabin that cause the booming noise while not affecting the ride quality of the vehicle by modifying the suspension.

This type of active control however is for control of general road-induced noise and not of a known frequency. Knowing the frequency of the acoustic mode, and the conditions under which the mode is best excited, a much more basic active control system can be installed that very simply targets the acoustic mode's frequency. The cabin's acoustic mode should be easy to target and cancel as it is a simple, stable, low frequency sound wave. The active control system concept does not change the properties of the vehicle; it is very much a "fix" that can be installed in the vehicle if required. This means that it is a cheap modification, the only associated cost being that of the active control system itself.

A slightly more complex active noise control system was developed by Kashani & Orzechowski (2001), based on the dynamics of a Helmholtz resonator. By incorporating the system into the vehicle's sound system, all that was required was a microphone and a cheap operational amplifier. It was found that the position of the speaker did not affect the level of damping of the targeted acoustic mode. Significant damping of the booming frequency was achieved using this system. The system was then enhanced by including a vibroacoustic element in the control system. Accelerometers were affixed to the main roof panels and the measurements entered into the control loop to the control speaker. Some damping of the roof's first bending mode was achieved. This concept could be applied to LUVs in general and specifically to the tested vehicle. Active control could be applied to the acoustic mode at 100 Hz using the vehicle's current sound system, and to the Back Panel's 92 Hz mode using a vibroacoustic system in parallel to the basic system. This would be a more expensive but probably more effective control system than the one described in the previous paragraph. It is also possible to use a panel (such as the roof) as a speaker rather than a speaker connected to the vehicle's

sound system to cancel out the targeted noise.

5.5 Conclusion

There are a number of possible ways to reduce the booming noise in the vehicle cabin. Only the option of modifying the Back Panel was tested, with limited success. The final choice as to which concept should be tested and possibly implemented is strongly dependant on economics. The vehicle manufacturing company has to decide whether the added NVH value of the modification would justify its price.

The most promising modification appears to be the softening of the suspension bushings. This could have absolutely no cost implications as the modification requires a simple replacement of the bushing and not any physical modification to the vehicle's structure. Unfortunately the bushing were specifically chosen to improve the vehicle's handling and softer bushings would probably affect this. The company therefore would need to decide whether sporty handling with booming or less sporty handling with less booming would result in increased vehicle sales and act according to that.

Active control of the acoustics in the cabin would probably be effective but most likely too expensive to even contemplate implementing on a relatively low-cost vehicle.



Chapter 6

Conclusions and Recommendations

This report documented an investigation into a low frequency road rumble or “booming” noise in a Light Utility Vehicle (LUV). A literature survey covering the field of modern vehicle Noise, Vibration and Harshness (NVH) was presented documenting all of the techniques used in the project as well as others that are commonly used but were either not applicable to the project’s requirements (such as low frequency analysis) or that required equipment that was not available.

This was followed by an investigation into the booming noise in the vehicle that specified—in engineering terms—the characteristics of the booming noise (Chapter 3). It was found that:

- the booming noise manifested in the frequency range of about 80 Hz to 105 Hz at speeds of between 25 km/h and 40 km/h, after which it decreased only to increase again from 60 km/h;
- the two principal components of the booming noise lay at around 90 Hz and 100 Hz;
- the fundamental acoustic mode in the cabin lay at 100 Hz with the maximum pressure of the wave situated behind the driver and passenger’s headrests.

The reasons for the noise in the cabin were then identified using a number of NVH tools, such as Experimental Modal Analysis (EMA) and Transfer Path Analysis (TPA). It was found that:

- the Back Panel (a panel situated at the end of the load bed directly behind the driver and passenger's seats) had a natural mode at 92 Hz that was a major source of cabin noise at that frequency;
- the amplification caused by the cabin's acoustic mode remained the greatest contributor to the internal cabin booming noise;
- no other panels contributed significantly to the booming noise.

The source of the vibrations in the vehicle that led to the generation of the booming noise in the cabin was then identified. It was found that:

- the drive shaft assemblies were excited by the coinciding natural rotational orders of the ball and tripod Constant Velocity (CV) joints;
- the most problematic occurrence was in the right drive shaft assembly when the ball joint's third and the tripod joint's sixth rotational orders coincided with a natural resonance in the drive shaft assembly at 83 Hz;
- this coincidence occurred at about 28 km/h.

The damped nature of this resonance resulted in high vibrational excitations being transferred to the wheel hub over a frequency range from 70 Hz to 90 Hz. The high compliance of a rubber mounting housing the bearing in the middle of the drive shaft assembly was largely responsible for the large amounts of vibration of the mode. The vibrational transfer path was identified as running from the drive shaft assembly through the wheel bearing to the wheel hub, and then through the wishbones to the chassis floor. No panel was identified as being exclusively responsible for converting the vibrations into the cabin booming noise. The Back Panel only accounted for some of the noise. It was put forward that the noise from the Back Panel, together with general vibration of the cabin's other panels was enough to be amplified by the acoustic mode and create the booming noise.

Potential solutions to the booming noise were proposed, with one of the solutions (a modification of the Back Panel) tested with a degree of success.

It is recommended that the proposed solutions be further investigated and tested. The most promising solutions are:

- the modification of the bushings on the left and right wishbones, as described in Section 5.2;
- the redesign or modification of the Back Panel;
- the concept of lengthening the fundamental acoustic wavelength in the cabin as described in Section 5.4.1.

The first two solutions are the most promising from the vehicle manufacturing company's point of view whereas the third is the most interesting and innovative from an academic perspective.

The work performed during this project contributed in strengthening the NVH expertise at Stellenbosch University. The main contributions were:

- software that calculates the A-weighted Sound Pressure Level (SPL) of a signal as a function of road speed;
- software that calculates 3 dimensional Power Spectral Densities (PSDs) of a signal as functions of frequency and road speed and plots the Waterfall or Campbell plot thereof;
- experience in using the previously unused software package SD Tools to extract experimental mode shapes and general modal information;
- Acoustic Modal Analysis (AMA) expertise and experience, both experimental and analytical (with the use of MSC.Nastran);
- TPA expertise and experience;
- EMA expertise and experience.

List of References

- Anon (2003). There are no good vibrations. *Materials World*, vol. 11, no. 8, pp. 33–34. ISSN 0967-8638. (Cited on pages 2, 5, and 19.)
- Biermann, J.W. (2004 September). Impact of side shafts on the interior noise of cars. *Proceedings of the 2004 International Conference on Modal Analysis Noise and Vibration Engineering*. (Cited on page 50.)
- Bissinger, G. (2003). Modal analysis of a violin octet. *Journal of the Acoustical Society of America*, vol. 113, no. 4, pp. 2105–2113. ISSN 0001-4966. (Cited on page 8.)
- Cerwin-Vega (2004). Vega!bass folded horn subwoofers.
Available at: <http://www.cerwin-vega.com/products/professional/vegabass/subs.html> (Cited on page 57.)
- Dehandschutter, W. and Sas, P. (1998). Active control of structure-borne road noise using vibration actuators. *Journal of Vibration and Acoustics*, vol. 120, pp. 517–523. ISSN 1048-9002. (Cited on pages 5, 10, and 59.)
- DeJong, R.G. (1985). A study of vehicle interior noise using statistical energy analysis (SEA). *SAE 850960*. (Cited on page 18.)
- Deweert, J. and Dierckx, B. (2000). Modal parameters from data measured on a car in operating conditions. *Proceedings of the International Modal Analysis Conference (IMAC)*, vol. 1, pp. 58–65. ISSN 1046-6770. (Cited on page 14.)
- Fastl, H. (1997). The psychoacoustics of sound-quality evaluation. *Acta Acoustica*, vol. 83, no. 5, pp. 754–764. (Cited on page 6.)
- Hendricx, W., Choi, Y.B., Ha, S.W. and Lee, H.K. (1997). Experimental body panel contribution analysis for road induced interior noise of a passenger vehicle. *SAE 971913*. (Cited on pages viii, 8, 9, 15, 16, 17, and 28.)
- Hendricx, W. and Vandenbroeck, D. (1993). Suspension analysis in view of road noise optimisation. *Proceedings of the 1993 Noise and Vibration Conference, SAE P-264*, pp. 647–652. (Cited on page 11.)

- Hepburger, A., Pribsch, H.-H., Desmet, W., van Hal, B., Pluymers, B. and Sas, P. (2002). Application of the wave based method for the steady-state acoustic response prediction of a car cavity in the mid-frequency range. *Proceedings of ISMA 2002*, vol. 2, pp. 877–884. (Cited on pages 17 and 19.)
- Hwang, Y., Lee, J. and Kim, H. (1997). Analysis of severe booming noise on a SUV. *Proceedings of the International Modal Analysis Conference (IMAC)*, vol. 1, pp. 438–443. (Cited on pages 8, 13, and 14.)
- Irwin, J.D. and Graf, E.R. (1979). *Industrial Noise and Vibration Control*, chap. 2, p. 49. 1st edn. Prentice-Hall Inc. (Cited on page 76.)
- Kashani, R. and Orzechowski, J. (2001). Active boom noise damping of dodge durango. *SAE 2001-01-1614*. (Cited on pages 58 and 59.)
- Knittel, J.D. and Oswald, L.J. (1987). An experimental technique for acoustic modal analysis in three-dimensional cavities. *SAE 870974*. (Cited on page 12.)
- Lee, D.-H., Hwang, W.-S. and Kim, M.-E. (2000). Booming noise analysis in a passenger car using a hybrid-integrated approach. *Society of Automotive Engineers Inc. (SAE) Trans. Journal of Passenger Cars: Mechanical Systems Journal*, vol. 109, pp. 1069–1075. (Cited on pages 6, 10, 14, 16, and 17.)
- Lilley, K.M. and Weber, P.E. (2003). Vehicle acoustics solutions. *S V Sound and Vibration*, vol. 37, no. 7, pp. 16–19. ISSN 0038-1810. (Cited on pages 2 and 5.)
- Maruyama, S., Hasegawa, A. and Y., H. (1999). Interior noise analysis based on acoustic excitation tests at low-frequency range. *SAE 1999-01-1806*. (Cited on page 9.)
- Matsuyama, S. and Maruyama, S. (1998). Booming noise analysis based on acoustic excitation test. *Society of Automotive Engineers Inc. (SAE) Special Publications*, vol. 1363, pp. 99–103. (Cited on pages 6, 8, 9, and 18.)
- MSC (2001). *MSC.Nastran Manual*. MSC, U.S.A. (Cited on page 13.)
- Parrett, A.V., Hicks, J.K., Burton, T.E. and Hermans, L. (1997). Statistical energy analysis of airborne and structure-borne automobile interior noise. *SAE 971970*. (Cited on page 17.)
- Plunt, J. (1998). Strategy for transfer path analysis (TPA) applied to virbo-acoustic systems at medium and high frequencies. *Proceedings of ISMA23*. (Cited on page 10.)

- van der Auweraer, H., Wyckaert, K. and Hendricx, W. (1997). From sound quality to engineering of solutions for NVH problems: Case studies. *Acta Acustica (Stuttgart)*, vol. 83, no. 5, pp. 796–804. ISSN 0001-7884. (Cited on pages 2, 5, 6, 10, 14, 18, and 25.)
- van der Linden, P.J.G. and Fun, J.K. (1993). Using mechanical-acoustic reciprocity for diagnosis of structure borne sound in vehicle. *SAE 931340*. (Cited on pages 8, 9, 10, and 11.)
- van der Linden, P.J.G. and Varet, P. (1996). Experimental determination of low frequency noise contributions of interior vehicle body panels in normal operation. *SAE 960194*. (Cited on pages 18 and 25.)
- van Niekerk, J.L. (2002 December). *NVH Course*. Institute for Thermodynamics and Mechanics, University of Stellenbosch. (Cited on pages 23 and 38.)
- Wassermann, J. and Springer, H. (2001). Acoustic noise reduction of rotating structures by the aid of experimental modal analysis. *Proceedings of the International Modal Analysis Conference (IMAC)*, vol. 2, pp. 1005–1009. ISSN 1046-6770. (Cited on page 8.)
- White, F.M. (1999). *Fluid Mechanics*, pp. 220–221. 4th edn. WCB/McGraw-Hill, Singapore. (Cited on page 12.)
- Wyckaert, K. and van der Auweraer, H. (1995). Operational analysis, transfer paths analysis, modal analysis: Tools to understand road noise in cars. *Proceedings SAE Noise and Vibration Conference*, pp. 139–143. (Cited on pages 10 and 14.)

Appendices

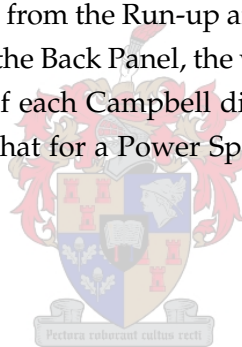


Appendix A

Test figures

A.1 Run-up test

Selected Campbell diagrams from the Run-up are included for reference purposes. The diagrams are for the Back Panel, the windscreen and the right rear wheel hub. Note the scale of each Campbell diagram has been adjusted for clarity and that the scale is that for a Power Spectral Density (PSD), namely $(\text{m/s}^2)^2/\text{Hz}$.



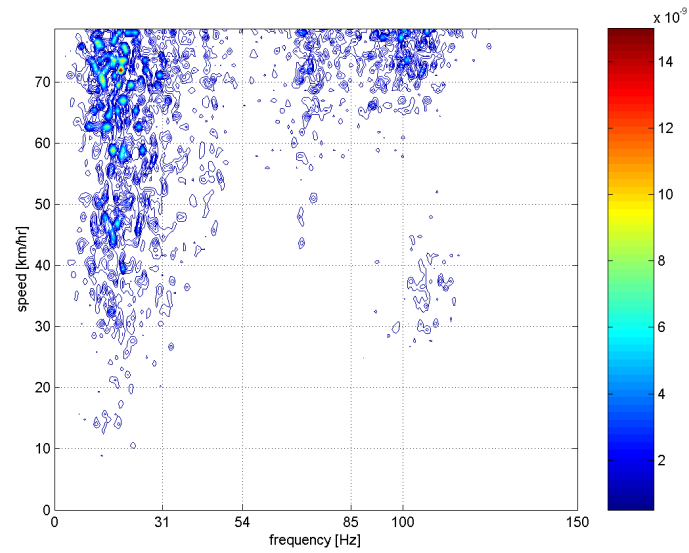


Figure A.1: Campbell diagram of the vibration level of the right rear wheel hub during the Run-up test

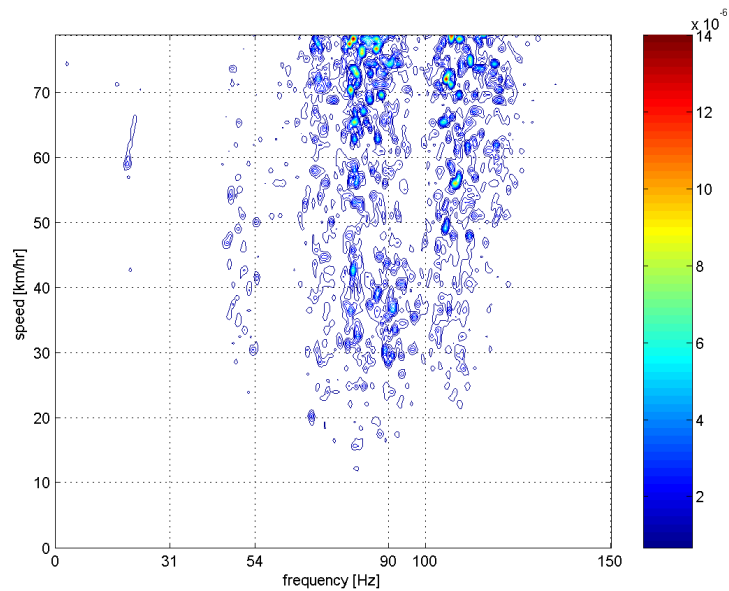


Figure A.2: Campbell diagram of the vibration level of the Back Panel during the Run-up test

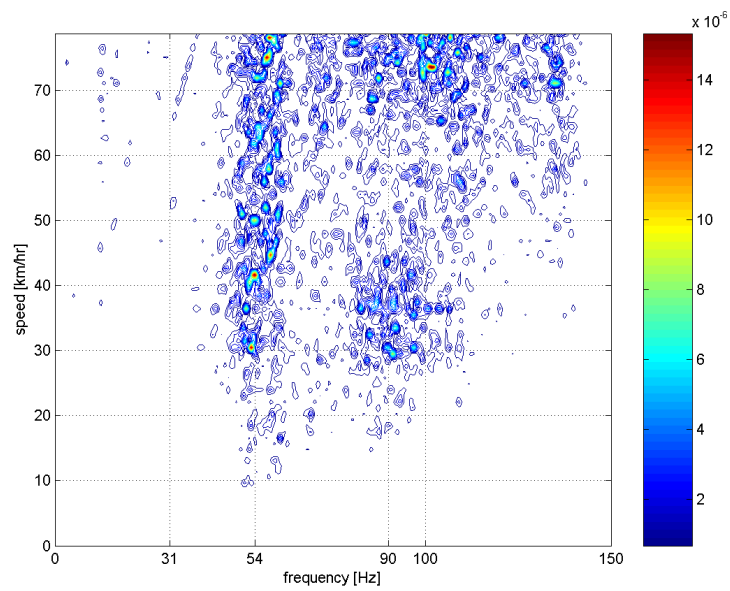


Figure A.3: Campbell diagram of the vibration level of the windscreen during the Run-up test

A.2 Experimental Modal Analysis

A.2.1 EMA of the Back Panel

The Back Panel was instrumented with seven accelerometers forming a grid on the panel. The panel was then excited using a modal hammer. The concept of a “speaker” mode can be clearly seen in Figure A.4, generated using SD Tools v5.1.

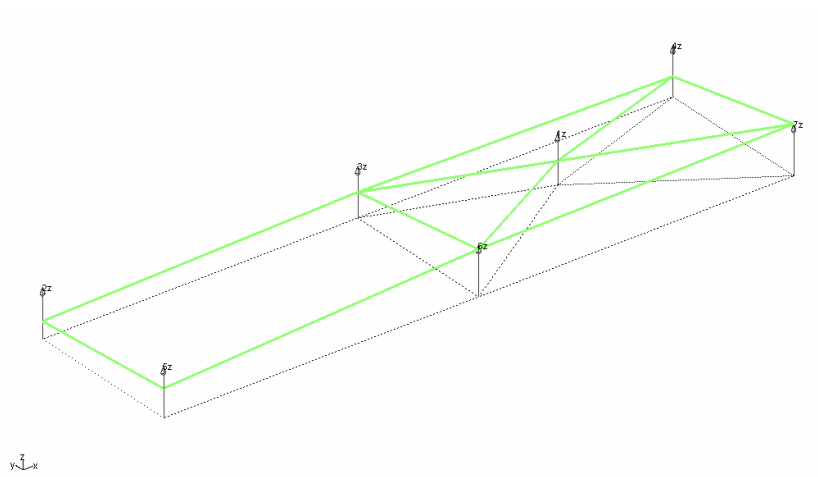


Figure A.4: The “speaker” mode shape of the Back Panel at 90 Hz

A.3 Drive shaft assembly tests

A.3.1 Experimental Modal Analysis

The right drive shaft assembly was instrumented with six accelerometers: three on each shaft (at the ends and middle of each shaft). Frequency Response Functions (FRFs) were calculated, after which SD Tools was used to calculate curve fits for each FRF. The curve fit along with the calculated FRF for the accelerometer closest to the wheel hub (i.e. on the outer most point of the Outer Shaft) is shown in Figure A.5. SD Tools v5.1 uses a modified circle fitting method to determine the system's poles, after which an optimisation script is run to optimise the frequency and damping values of the poles. For more information visit <http://www.sdtools.com/sdt/index.html>.

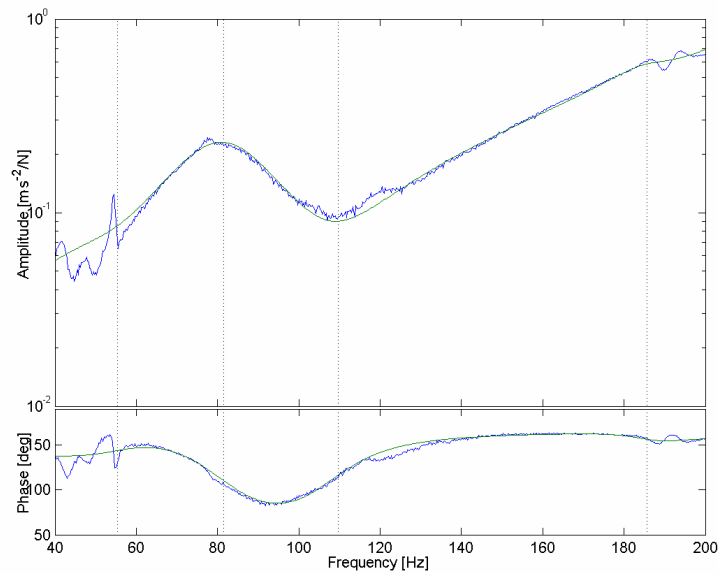


Figure A.5: FRF for the Outer Shaft's outer-most accelerometer showing the calculated curve fit

A.4 Operational Transfer Path Analysis

The operational Transfer Path Analysis (TPA) test was actually conducted in two stages. In the first stage the engine and suspension members were instrumented. Once it was clear that the vibrations did not enter the vehicle via the engine, another test was conducted with accelerometers again on the suspension members and wheel hubs, but also on the chassis floor and on the cabin's left and right floor panels. All testing was performed symmetrically. In other words all accelerometer locations on the right hand side of the vehicle were mirrored as closely as possible on the left hand side.

The measurements for the left side measurements are included here for comparison to the measurements of the right hand side on pages 47 to 49. Note once again that the units are $(\text{m/s}^2)^2/\text{Hz}$ and not m/s^2 and that scales have been adjusted for clarity. The same areas of vibration are seen in both these plots and the plots of the right side measurements, however the amplitudes are much smaller than those on the right hand side.

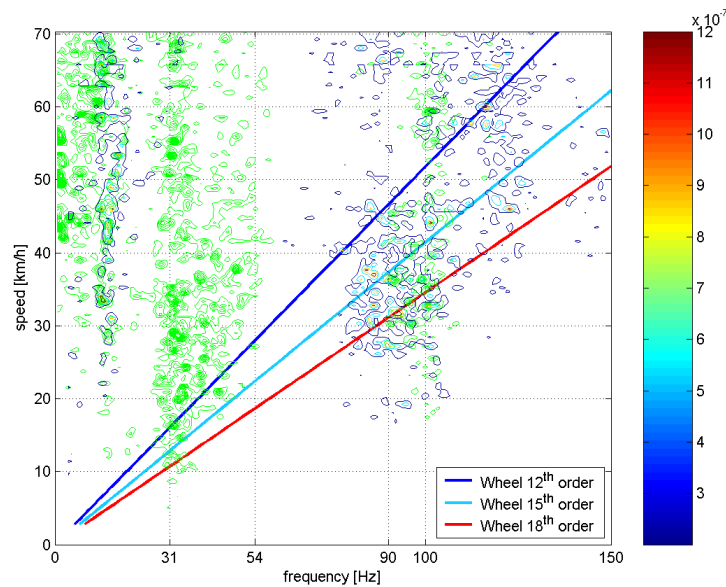


Figure A.6: Campbell diagram of the vibration level of the left wishbone's inner swivel connection

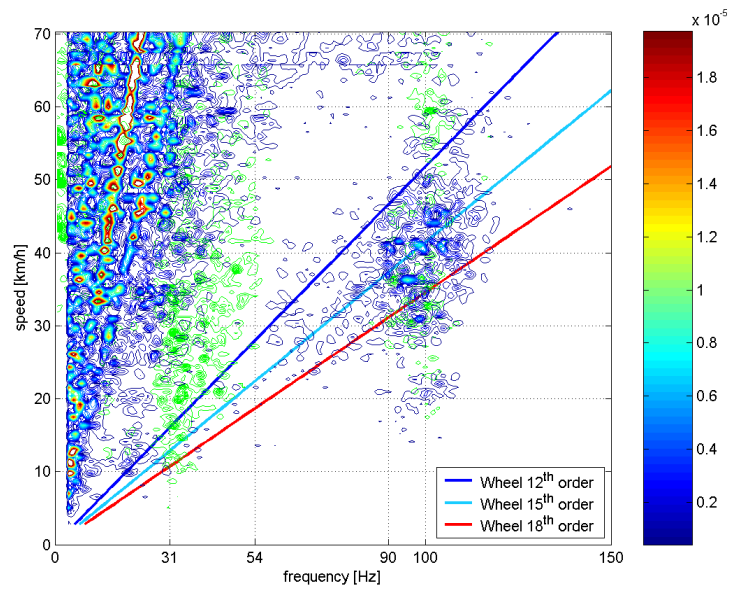


Figure A.7: Campbell diagram of the vibration level of the left wishbone's outer swivel connection

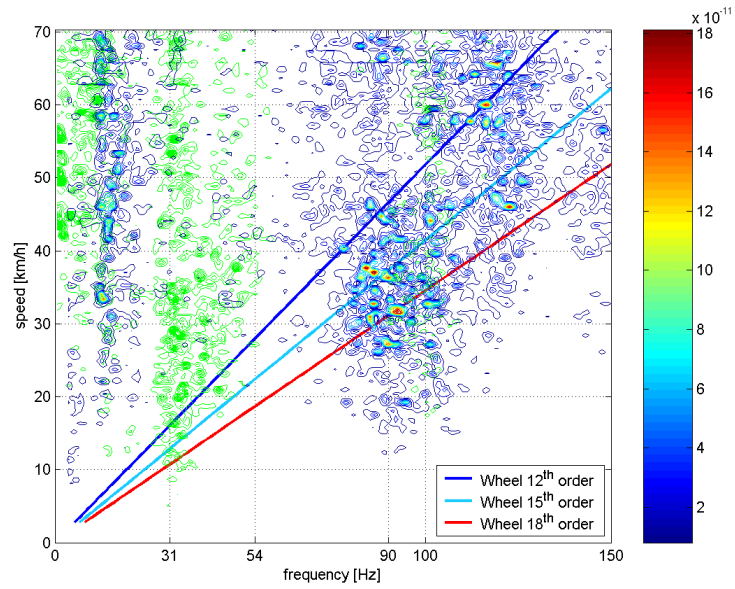


Figure A.8: Campbell diagram of the vibration level of the vehicle chassis near the left wishbone

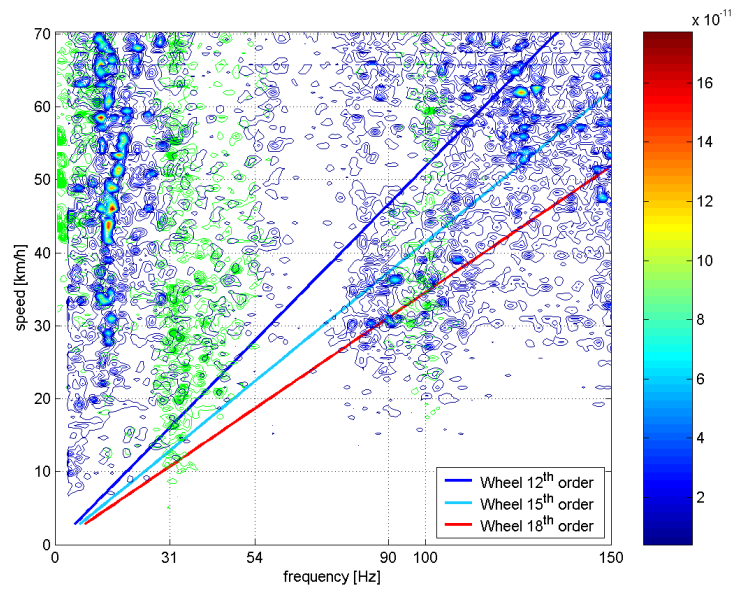


Figure A.9: Campbell diagram of the vibration level of the driver's footwell floor

Appendix B

The A-weighting scale

The human ear does not hear all frequencies equally loudly. Low frequencies are not heard very well, while frequencies between 1 kHz and 5 kHz are actually perceived to be louder than they are. The A-weighting scale weighs the frequency content of noise signals in such a way as to account for the sensitivity of an average human ear to those frequencies. The values in Table B.1 are taken from Irwin & Graf (1979).

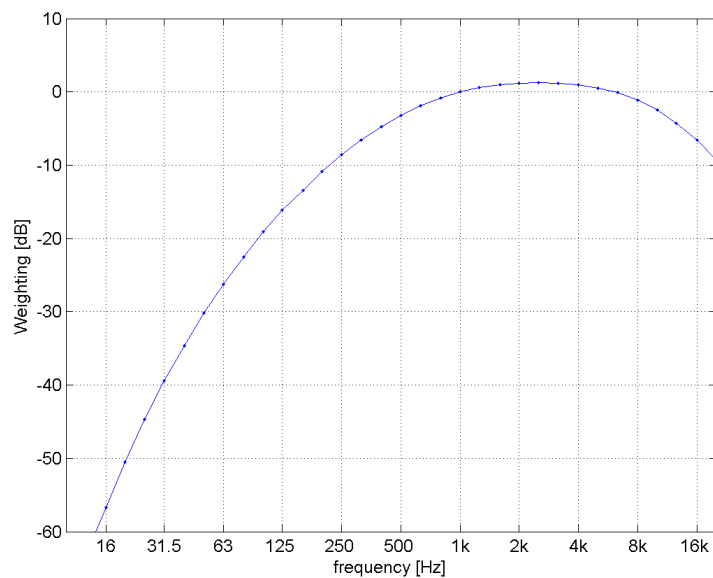


Figure B.1: Plot of the A-weighting scale

Table B.1: The A-weighting table

Frequency [Hz]	Weighting [dB]
10	−70.4
12.5	−63.4
16	−56.7
20	−50.5
25	−44.7
31.5	−39.4
40	−34.6
50	−30.2
63	−26.2
80	−22.5
100	−19.1
125	−16.1
160	−13.4
200	−10.9
250	−8.6
315	−6.6
400	−4.8
500	−3.2
630	−1.9
800	−0.8
1000	0.0
1250	0.6
1600	1.0
2000	1.2
2500	1.3
3150	1.2
4000	1.0
5000	0.5
6300	−0.1
8000	−1.1
10000	−2.5
12500	−4.3
16000	−6.6
20000	−9.3