

**VEHICLE INTERIOR NOISE SOURCE CONTRIBUTION
AND TRANSFER PATH ANALYSIS**

**M.Sc. Thesis by
Fuat Okan TANDOĞAN, B.Sc.**

Department : Mechanical Engineering

Programme: Automotive Engineering

Supervisor : Prof. Dr. Ahmet GÜNEY

MAY 2006

**VEHICLE INTERIOR NOISE SOURCE
CONTRIBUTION AND TRANSFER PATH ANALYSIS**

**M.Sc. Thesis by
Fuat Okan TANDOĞAN, B.Sc.**

(503021008)

Date of submission : 5 May 2006

Date of defence examination: 15 June 2006

Supervisor (Chairman): Prof. Dr. Ahmet GÜNEY

Members of the Examining Committee Prof. Dr. Ali GÖKTAN (İ.T.Ü.)

Prof. Dr. İrfan YAVAŞLIOL (Y.T.Ü.)

MAY 2006

**ARAÇ GÜRÜLTÜ KAYNAKLARININ TOPLAM İÇ
GÜRÜLTÜYE KATKISI VE GÜRÜLTÜ TRANSFER YOLU
ANALİZİ**

**YÜKSEK LİSANS TEZİ
Mak. Müh. Fuat Okan TANDOĞAN
(503021008)**

**Tezin Enstitüye Verildiği Tarih : 5 Mayıs 2006
Tezin Savunulduğu Tarih : 15 Haziran 2006**

**Tez Danışmanı : Prof. Dr. Ahmet GÜNEY
Diğer Jüri Üyeleri Prof. Dr. Ali GÖKTAN (İ.T.Ü.)
Prof. Dr. İrfan YAVAŞLIOL (Y.T.Ü.)**

MAYIS 2006

ACKNOWLEDGEMENT

I would like to express my sincere appreciation to my advisor Professor Ahmet GÜNEY for his patient guidance during all phases of this study.

I would also like to thank all of my friends and colleagues in my company for their help and support.

I wish to say many thanks to my love Derya for her patience and support during the years of my education, and encouragement in the times of desperation.

Finally, I am indebted to my family for their unconditional support and never-ending belief throughout my life.

May 2006

Fuat Okan TANDOĞAN

TABLE OF CONTENTS

ACRONYMS	vi
LIST OF TABLES	vii
LIST OF FIGURES	viii
LIST OF SYMBOLS	x
SUMMARY	xi
ÖZET	xii
1. INTRODUCTION	1
2. THEORY AND BACKGROUND	3
2.1 Fundamentals of Acoustics	3
2.1.1 What is Sound?	3
2.1.2 The Speed of Sound.....	3
2.1.3 Speed and the Frequency	4
2.1.4 Phase and Interaction of Waves.....	5
2.1.5 Sound Pressure and Sound Pressure Level.....	6
2.1.6 Particle Velocity and Particle Velocity Level	7
2.1.7 Sound Intensity and Sound Intensity Level	8
2.1.8 Sound Power and Sound Power Level.....	9
2.1.9 Acoustic Impedance	10
2.1.10 Human Hearing and Psychoacoustics.....	10
2.1.10.1.Ear and Hearing	10
2.1.10.2.Limits of Human Hearing and Perception	12
2.1.10.3.Perceived Loudness	14
2.1.10.4.Weighting Networks.....	16
2.1.10.5.Octave and 1/3 Octave Band Filters.....	19
2.1.10.6.Articulation Index	21
2.2 Fundamentals of Vibration	22
2.2.1 Undamped Free Vibration	23
2.2.2 Undamped Forced Vibration.....	23

2.2.3	Viscous Damping.....	25
2.2.4	Structural Damping.....	26
2.2.5	Frequency Response Function (FRF)	27
2.3	Fundamentals of Vehicle NVH	30
2.3.1	Structure-borne and Air-borne Sound	32
2.3.2	Noise Sources and Transfer Paths in Vehicles	32
2.3.2.1.	Powertrain NVH.....	32
2.3.2.2.	Intake System Radiated Noise	33
2.3.2.3.	Exhaust System Radiated Noise	35
2.3.2.4.	Engine Radiated Noise.....	35
2.3.2.5.	Body Air-borne Noise Transparency.....	38
3.	VEHICLE INTERIOR NOISE SOURCE CONTRIBUTION AND TRANSFER PATH ANALYSIS	40
3.1	Selection of the Test Method	40
3.2	Test Environment.....	41
3.3	Data Acquisition Hardware and Software.....	41
3.4	Measurement Details	42
3.5	Air-borne Noise Contribution Measurement Technique	43
3.5.1	Intake System Air-borne Noise Contribution.....	43
3.5.2	Exhaust System Air-borne Noise Contribution	44
3.5.3	Engine Air-borne Noise Contribution	45
3.6	Structure-borne Noise Contribution Measurement Technique	48
3.7	Measured and Calculated Levels and Root Cause Identification.....	50
3.8	Potential Solution Methods	62
3.8.1	Reducing the Excitation Forces.....	62
3.8.2	Decreasing Noise Transfer Function Amplitudes	66
4.	CONCLUSION & DISCUSSION	69
	REFERENCES.....	70
	AUTOBIOGRAPHY	71

ACRONYMS

CAE	: Computer Aided Engineering
DMF	: Dual-mass Flywheel
EO	: Engine Order
FRF	: Frequency Response Function
LFSS	: Low Frequency Sound Source
MDOF	: Multi Degree of Freedom
NTF	: Noise Transfer Function
NVH	: Noise Vibration and Harshness
RMS	: Root Mean Square
RPM	: Revolution Per Minute
SDOF	: Single Degree of Freedom
SMF	: Split Mass Flywheel
WOT	: Wide Open Throttle

LIST OF TABLES

	<u>Page No</u>
Table 2.1. Relation between absolute sound pressure and sound pressure level.....	12
Table 2.2. Sample calculation of loudness and loudness level.....	16
Table 2.3. A-weighting attenuation values for octave band center frequencies.....	18
Table 2.4. Center frequencies and bandwidths for the preferred octave and 1/3 octave bands.....	21
Table 2.5. Definition of Frequency Response Functions	28
Table 2.6. Engine orders and their definitions/sources.....	38
Table 3.1. Interior Noise Contribution and Transfer Path of Each Source.....	42
Table 3.2. Percentage of Each Contributor.....	59
Table 3.3. Solution Opportunities of the Observed NVH issue with costs and required timing.....	68

LIST OF FIGURES

	<u>Page No</u>
Figure 2.1.	Schematic view of a wave's fundamental parameters.....5
Figure 2.2.	Two waves with Φ phase difference5
Figure 2.3.	Interaction of two in-phase waves6
Figure 2.4.	Interaction of two waves, which are out-of-phase by 180°6
Figure 2.5.	Longitudinal Wave7
Figure 2.6.	Transverse Wave8
Figure 2.7.	Sound Intensity from a Spherical Sound Source9
Figure 2.8.	Structure of Human Ear 11
Figure 2.9.	Vibration of the ear drum by the sound waves coming through the auditory canal. 11
Figure 2.10.	The Range of Human Hearing..... 13
Figure 2.11.	Hearing Range in Animals. 14
Figure 2.12.	Equal Loudness Curves 15
Figure 2.13.	Frequency Response Functions of A, B, C and D weighting filters. 18
Figure 2.14.	Typical Frequency Response Function of a band-pass Filter 20
Figure 2.15.	Basic model of a SDOF system. 23
Figure 2.16.	Force deflection characteristics. 27
Figure 2.17.	Receptance FRF plot for undamped SDOF system. 29
Figure 2.18.	Mobility FRF plot for undamped SDOF system. 29
Figure 2.19.	Inertance FRF plot for undamped SDOF system..... 30
Figure 2.20.	Noise and Vibration Sources in a Vehicle 33
Figure 2.21.	Intake and Exhaust System Components 34
Figure 2.22.	Intake and Exhaust System Noise Transfer Paths 34
Figure 2.23.	Structure-borne and air-borne engine noise radiation. 36
Figure 2.24.	Waterfall Diagram 37
Figure 3.1.	Intake Orifice Radiated Noise Measurement Location 44
Figure 3.2.	Exhaust Orifice Radiated Noise Measurement Location (Top View)..... 45

Figure 3.3.	Exhaust Noise Reduction Measurement Method.....	45
Figure 3.4.	Volume Acceleration Transfer Function Measurement.....	47
Figure 3.5.	Air-borne Noise Sources and Transfer Paths to the Vehicle Interior.....	48
Figure 3.6.	Structure-borne Noise Contribution.....	49
Figure 3.7.	Calculated vs. Measured Sound Pressure Levels.....	50
Figure 3.8.	Identification of a potential NVH problem at 3900 RPM.....	51
Figure 3.9.	Frequency Spectra of the Measured Interior Noise.....	52
Figure 3.10.	Contribution of critical engine orders on overall interior noise level – from the calculated data.....	53
Figure 3.11.	Structure-borne and Air-borne Noise Contribution.....	54
Figure 3.12.	4 th Engine Order Structure-borne contribution of all systems.....	55
Figure 3.13.	Powertrain sub-system level 4 th EO Contributions.....	56
Figure 3.14.	Suspension attachments sub-system level 4 th EO Contributions.....	57
Figure 3.15.	The most dominant sources 4 th Engine Order Contributions.....	58
Figure 3.16.	Effect of all directions on LH L-Arm front 4 th engine order structure-borne contribution.....	59
Figure 3.17.	LH L-Arm Front Noise Transfer Functions in all Directions.....	60
Figure 3.18.	Overall and 4 th Engine Order Force Levels in x, y, z directions from LH L-Arm Front attachment point.....	61
Figure 3.19.	3 rd gear WOT acceleration with SMF.....	65
Figure 3.20.	3 rd gear WOT acceleration with DMF.....	66

LIST OF SYMBOLS

A	: Acceleration
B	: Amplitude of Sine Wave
b	: Viscous Damping Coefficient
c	: Speed of Sound in m/s
f	: Frequency
f_c	: Centre Frequency
f_l	: Lower Limiting Frequency
f_u	: Upper Limiting Frequency
F	: Harmonically Varying Force
I	: Sound Intensity
I₀	: Actual Sound Intensity
I_{ref}	: Reference Sound Intensity
k	: Stiffness
K	: Adiabatic Bulk Modulus
L_I	: Sound Intensity Level
L_N	: Loudness Level
L_p	: Sound Pressure Level
L_w	: Sound Power Level
L_v	: Particle Velocity Level
M	: Mass
n	: Octave Band Interval
N	: Loudness
P	: Sound Pressure
P_e	: Actual Sound Pressure
P_{Ref}	: Reference Sound Pressure
r	: Distance
R	: Gas Constant
s	: Number of Order
S	: Surface Area
T	: Sampling Period
T_K	: Absolute Temperature in Kelvin
V	: Velocity
Z	: Acoustic Impedance
X	: Maximum Displacement
W	: Sound Power
W_I	: Actual Sound Power
W_{Ref}	: Reference Sound Power
λ	: Wave Length
ρ	: Density
γ	: Adiabatic Index
Φ	: Initial Phase
v	: Actual Particle Velocity
v_{Ref}	: Reference Particle Velocity
ω_n	: Natural Frequency
ξ	: Damping Ratio
μ	: Loss Factor

VEHICLE INTERIOR NOISE SOURCE CONTRIBUTION AND TRANSFER PATH ANALYSIS

SUMMARY

In this study, the fundamentals of noise, vibration and important steps in vehicle NVH development process are discussed. Vehicle air-borne and structure-borne noise sources and noise transfer paths are identified. By applying the interior noise source contribution and transfer path analysis method, the contribution of each noise source in the vehicle is calculated by sub-system level measurements. Moreover, the actual vehicle level interior noise has been measured and compared with the calculated levels for correlation. When good correlation is achieved, a critical peak level, which may be a potential NVH problem, has been investigated throughout the measured and calculated data.

The investigations show that good correlation between the measurement-driven model and actual vehicle level measurement can be achieved when each of the noise sources and their transfer paths are identified correctly and the measurements are conducted accurately within the frequency range of interest. It is also concluded that vehicle interior noise contribution and transfer path analysis method is a good approach for vehicle level interior noise calculation, vehicle level target setting and process of target cascading to system and sub-system levels during vehicle NVH development.

ARAÇ GÜRÜLTÜ KAYNAKLARININ TOPLAM İÇ GÜRÜLTÜYE KATKISI VE GÜRÜLTÜ TRANSFER YOLU ANALİZİ

ÖZET

Bu çalışmada, gürültü & titreşimin temelleri, ve araç gürültü ve titreşim geliştirilmesindeki önemli bazı adımların temelleri anlatılmıştır ve çalışmanın ileri safhalarına hazırlık için bazı temel ön bilgiler verilmiştir. Araçtaki hava taşınım ve yapısal taşınım gürültü katkısı olan kaynaklar ve bu kaynakların araç içi gürültü transfer yolları belirlenerek, araç toplam iç gürültü seviyesi, sistem ve alt system seviyesi ölçümlerden yola çıkılarak hesaplanmıştır. Toplam iç gürültünün hesaplanması, sistem ve alt sistemler seviyesinden iletilen her bir katkının toplanması ile elde edilmiştir. İlaveten, aynı araç işletme şartlarında iken, toplam araç iç gürültüsü ölçüm yolu ile de elde edilerek hesaplanan değerler ile korelasyon açısından karşılaştırılmıştır. Ölçülen ve hesaplanan seviyeler arasında iyi bir korelasyonun sağlandığı saptanınca, potansiyel bir gürültü sorunu oluşturabilecek kritik bir tepe noktası değeri hem ölçülen hem de hesaplanan datalar içerisinde incelenmiştir. Bunlara ilaveten, bu spesifik gürültü probleminin temel sebebini anlayabilmek üzere araç gürültü kaynaklarının iç gürültüye etkisi ve gürültü transfer yolu analizi metodu yaklaşımı ile belirlenmiş olan problemi oluşturan etkenler araştırılmıştır.

Çalışmanın sonucunda, şayet araçtaki yapısal taşınım ve hava taşınım katkısı olan gürültü kaynakları doğru bir şekilde belirlenirse ve ilgilenilen frekans aralıklarında güvenilir ölçümler yapılırsa, ölçülen ve hesaplanan araç toplam iç gürültü seviyeleri arasında iyi bir korelasyonun elde edilebileceği sonucuna varılmıştır. İlaveten, araç gürültü kaynaklarının iç gürültüye katkısı ve gürültü transfer yolu analizi metodunun; araç toplam iç gürültüsü hesaplamalarında, araç seviyesi gürültü ve titreşim hedeflerinin belirlenmesinde, ve araç seviyesi hedeflerin sistemler ve alt sistemler seviyelerine indirgenmesinde verimli bir şekilde kullanılacak bir yaklaşım metodolojisi olduğu sonucuna varılmıştır.

1. INTRODUCTION

In the recent years, manufacturers need to meet very strict development targets under restricted budgets for customer satisfaction and loyalty, to be able to remain competitive and profitable in the market in all areas of application. The main reason behind this is that customers become more and more conscious in consuming products based on their requirements. Low cost expectations are now being considered together with high quality and innovative products, which are putting the manufacturers under big pressure during the determination of the development and marketing strategies.

Automotive industry is one of the mostly affected areas, where advanced engineering should jointly be considered with financials and also with good marketing strategies. Marketing strategies are identified based on the requirements from the customers; and upon the targets to be achieved, development studies are carried out to optimize each of the products from quality and cost point of view.

Not only the low cost and high quality demands are valid for passenger cars, it is also becoming essential for commercial vehicles within this challenging market. To be able to meet the customer expectations in all aspects, attribute teams are working within development departments with groups of experienced engineers and specialists. When the customer profile in the automotive industry is considered, a vehicle is expected mainly to show a satisfying performance with less fuel consumption; it should be durable enough and safe; it should ergonomically be used and be serviceable. For a pleasant drive, it should have good driving quality with improved vehicle dynamics performance, improved NVH characteristics, and in addition to all of those mentioned it should have a competitive price. Vehicle noise and vibration performance is one of the main aspects, which can be classified within driving quality, as it has a direct impact on customers. Therefore, automotive manufacturers are spending a lot of money for noise and vibration development from the beginning of first vehicle prototypes till the launch of a vehicle. In this manner, NVH departments are being established within automotive companies and

expertise is getting more and more important with the combination of good theoretical background and enough practical experience.

In this study, it is aimed to describe one of the most important steps in vehicle NVH development and target setting, which can be referred as vehicle interior noise source contribution and transfer path analysis. A commercial vehicle overall interior noise is calculated by summing up the contributions from each noise sources and the achieved vehicle level noise data is then compared with the calculated values. Finally; a critical peak noise level, which may be a potential NVH problem, is investigated in more details with the interior noise contribution and transfer path analysis method and potential solution opportunities are addressed with approximated costs and required development timing.

2. THEORY AND BACKGROUND

In this section of the study, it is aimed to give technical background on acoustic and vibration fundamentals and vehicle NVH development.

2.1 Fundamentals of Acoustics

Acoustics can simply be defined as a branch of science, which deals with sound. Its generation, propagation control and effects on human beings (psychoacoustics) can be classified as the topics of acoustics. Some of the frequently used acoustics terminology will be explained in details in the following sections, in order to provide a technical background.

2.1.1 What is Sound?

Sound is simply a mechanical wave that propagates through a medium. Sound is usually generated by the vibrations of solid objects and it propagates in form of waves. As sound is not an electromagnetic wave like X-ray, a medium is necessary for the propagation of sound waves. The propagation of sound, in other words, means the transfer of energy from one point to another and the speed of sound waves is determined by the mean density of the fluid and its elasticity, which is called bulk modulus. Bulk modulus is related to the acoustic pressure to volumetric strain [3].

Speed of sound wave is about 1450 m.s^{-1} in water and 340 m.s^{-1} in air. This is linked to a fact that the density of water is nearly 800 times more than air; whereas, its bulk modulus, measure of elasticity, is about 15000 times that of air [3], resulting sound wave propagate much faster in water than in air.

2.1.2 The Speed of Sound

An approximate speed of sound in gases can be calculated from the equation below [2]:

$$c^2 = \gamma RT_K \quad (2.1)$$

where;

c is the speed of sound in meters per second.

γ is the adiabatic index (ratio of the specific heat capacity at constant pressure to the specific heat capacity at constant volume)

R is the gas constant

T_K is the absolute temperature in Kelvins

As a more general explanation, the speed of sound can be written as:

$$c = \sqrt{\frac{K}{\rho}} \quad (2.2)$$

where;

K is the adiabatic bulk modulus.

ρ is the density

From equation (2.2), it can be figured out that the speed of sound in a fluid is a function of the fluid density and bulk modulus.

2.1.3 Speed and the Frequency

Considering sound waves, one other very important phenomenon is the frequency of the sound, which is referred as the number of cycles per second or Hertz (Hz.).

Sound waves can be classified into three based on their frequencies. Sound waves having frequencies below 20 Hz. are referred as infra sound. The frequency range of 20 Hz. – 20 kHz. is called the audible range. Above the audible frequency range the ultra sound takes place and covers the application areas like acoustic modelling techniques, medical diagnosis and non-destructive material testing [4].

When sound waves travel from one medium to another, their speed remains the same. Their frequency and wavelength changes as they are inverse proportional to each other.

$$c = \lambda f \quad (2.3)$$

where ;

λ is the wavelength of the sound wave

f is the frequency of the sound wave

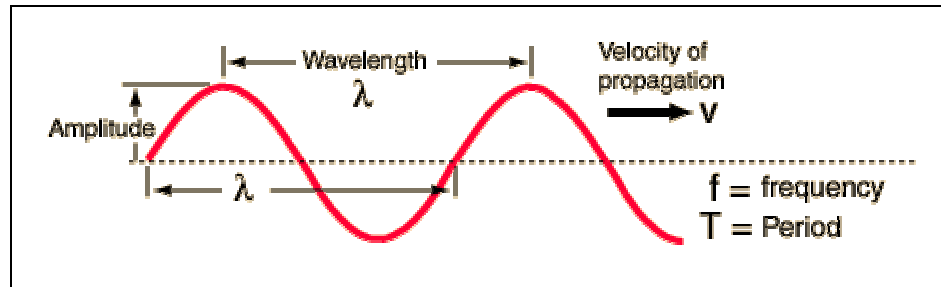


Figure 2.1. Schematic view of a wave's fundamental parameters.

2.1.4 Phase and Interaction of Waves

Sound waves can be modelled as sinusoidal functions, whose amplitude change in time. While frequency describes the oscillations per second, the phase identifies the place of the sound wave in its cycle of amplitude change. A basic sine wave can be considered as $B \cdot \sin(2\pi ft + \Phi)$ where, A represents the amplitude; f , is the frequency and t is the time. The angle $(2\pi ft + \Phi)$ represents the phase of the sine wave and Φ is referred as the initial phase [8].

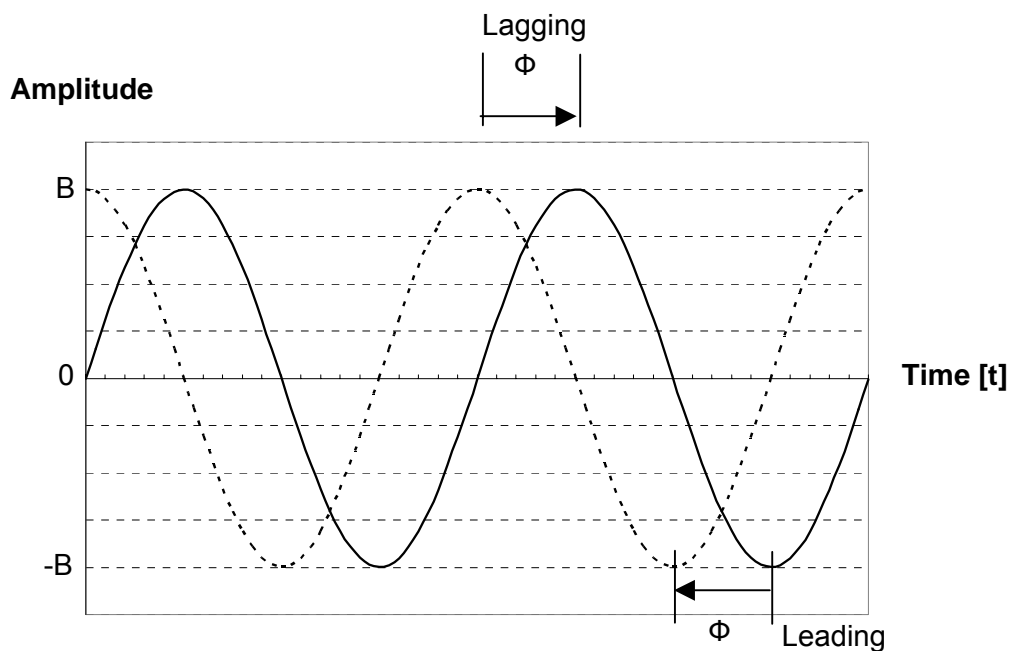


Figure 2.2. Two waves with Φ phase difference

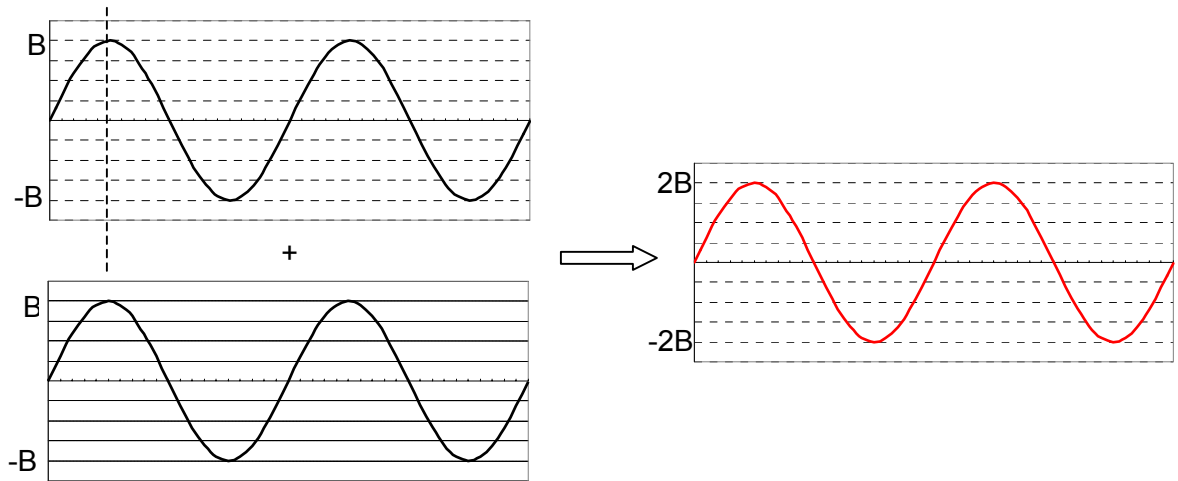


Figure 2.3. Interaction of two in-phase waves

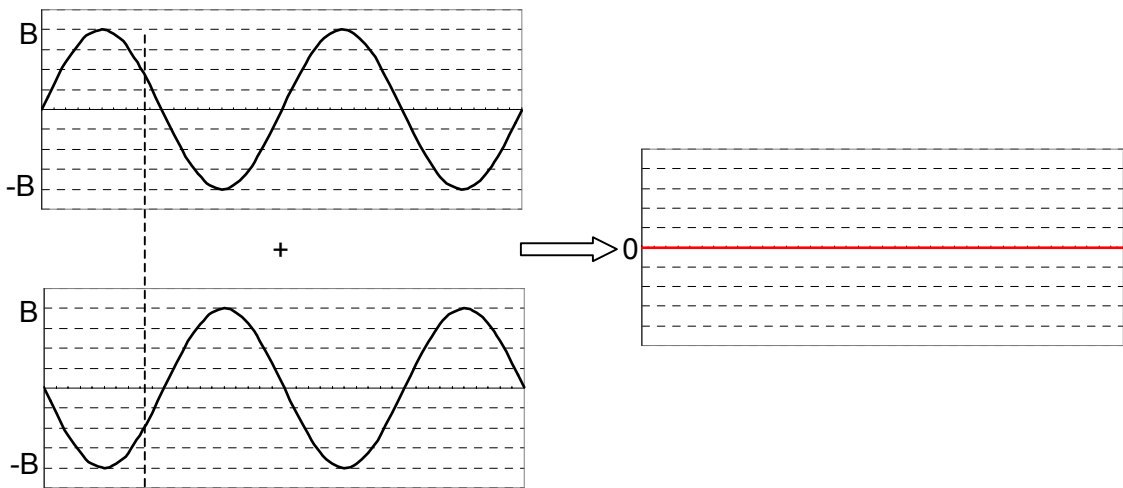


Figure 2.4. Interaction of two waves, which are out-of-phase by 180°

2.1.5 Sound Pressure and Sound Pressure Level

Sound pressure is a measure in Pascals (N/m^2) and is the root mean square (RMS) of the time domain signal [4] caused by a sound wave.

Sound pressure level is a logarithmic measure of the RMS sound pressure of a particular noise relative to a reference pressure [8]. Equation (2.4) expresses the relation between the sound pressure and sound pressure level in decibels.

$$L_p = 20 \log \left(\frac{P_e}{P_{ref}} \right) \quad (2.4)$$

2.1.6 Particle Velocity and Particle Velocity Level

As mentioned in the previous chapters, sound is a wave travelling through a medium. Particle velocity can be defined as the physical speed of the molecules inside a medium as it moves back and forth as it transmits the wave.

Depending on the relation between the direction of particle movement and the travelling wave, the wave motion for a mechanical wave can be classified as longitudinal waves or transverse waves.

If the particle displacement is parallel to the direction of wave propagation, as shown in figure 2.4, this kind of wave motion is called longitudinal wave.

A typical example to the longitudinal wave should be given as a slinky attached horizontally from one end. When the free-end of the slinky is given an excitation in the horizontal direction, the particles are set to a vibration movement in the direction of wave travel, or in other words the direction energy is transported. Sound waves most of the time propagates in a longitudinal motion. A very typical example should be given as functioning loudspeaker. The membrane of the loudspeaker is set to a vibration motion back and forth and excites air molecules in the direction of the particle motion. The propagation of the sound waves from the membrane surface to the listener's ear is considered as a longitudinal wave motion.

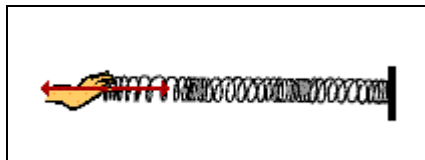


Figure 2.5. Longitudinal Wave

Another very typical motion can be classified as transverse motion, which is observed mostly in electromagnetic waves. In this motion, the direction of particle movement is perpendicular to the direction of wave propagation.

When a rope is excited one dimensionally, the particles oscillate up and down about their individual equilibrium position and do not move along with the wave. Here, the direction of wave propagation is perpendicular to the direction of particle movement (See figure 2.6).

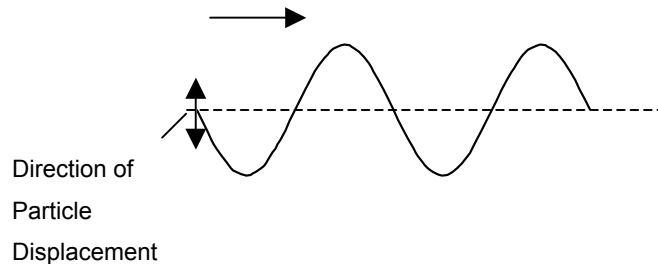


Figure 2.6. Transverse Wave

Particle velocity level is referred as the ratio of the particle velocity to a reference level. Equation 2.5 shows the calculation of the particle velocity level [8].

$$L_v = 20 \log_{10} \left(\frac{v}{v_{\text{ref}}} \right) \text{dB} \quad (2.5)$$

2.1.7 Sound Intensity and Sound Intensity Level

Sound intensity is the measure of sound power in unit area. Sound intensity can be calculated from the sound pressure and the particle (sound) velocity. Equation (2.6) shows the calculation of sound intensity [4].

$$I = W \cdot v \quad (2.6)$$

When an ideal spherical sound source is considered, meaning that the sound waves are distributed uniformly in all directions, the sound intensity as a function of distance r can be written as [8]:

$$I = \frac{W}{A} = \frac{W}{4\pi r^2} \quad (2.7)$$

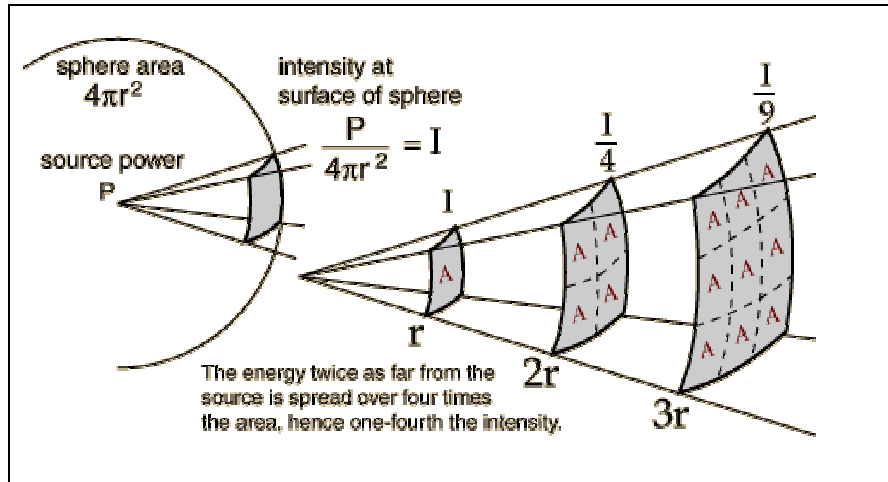


Figure 2.7. Sound Intensity from a Spherical Sound Source

If we introduce the sound intensity in logarithmic scale, logarithm of the ratio of a given sound intensity to a reference intensity, which is the standard reference threshold of hearing intensity, the sound intensity level can be calculated as shown in equation (2.8).

$$L_I = 10 \log \left(\frac{I_0}{I_{\text{ref}}} \text{ dB} \right), \quad I_{\text{ref}} = 10^{-12} \text{ W/m}^2 \quad (2.8)$$

2.1.8 Sound Power and Sound Power Level

Sound power can be defined as the total sound energy radiated from a sound source per unit time. Sound power is measured in units of watts, or sound intensity times surface area. When sound power is considered, the cross-sectional area of the power flow must be taken into account, due to the fact that sound waves are distributed spatially from a source. Therefore, sound power increases as the surface area of the power flow increases as seen in equation (2.9) [4].

$$W = I.S \quad (2.9)$$

Like sound pressure, sound power is generally expressed in decibel scale, relative to a reference sound power level and is called sound power level. Equation (2.10) shows the calculation on the sound power level.

$$L_w = 10 \log_{10} \left(\frac{W_1}{W_{\text{ref}}} \right) \text{ dB} \quad (2.10)$$

2.1.9 Acoustic Impedance

Acoustic impedance, or specific acoustic impedance, is the ratio of sound pressure P to the particle velocity v in a medium. Equation (2.11) shows the calculation of the acoustics impedance.

$$Z = \frac{P}{v} \quad (2.11)$$

Acoustic impedance is a good metric especially in determining the acoustic radiation from sources; acoustic transmission and reflection of the boundary of materials with different acoustic impedance.

2.1.10 Human Hearing and Psychoacoustics

As mentioned in section 2.1.3, the branch of acoustics deals with a wide range of frequencies depending on the area of application. However, when the area of human hearing and human perception of sound is subject to consideration, certain limitations exist, and are investigated under the branch Psychoacoustics.

Engineering acoustics, as previously mentioned, deals with a variety of areas of application, such as infra sound, ultra sound, underwater acoustics and etc. However, in the acoustical design of the environments or equipments, which are directly intended to the use of humans, the important parameter is the way human ear hears. When dealing with subjective perception of acoustics and also acoustic signal processing, it is essential to have knowledge on how ears hear. Sound waves can be considered as continuous analog signals, which include a variety of information such as frequency, phase, amplitude and etc. Nowadays technology enables accurate measurement of sound related physical quantities; however, when human perception of sound is subject to consideration, it is essential to have knowledge on psychoacoustic parameters, which will be explained in details.

2.1.10.1. Ear and Hearing

In order to understand the fundamental parameters in Psychoacoustics, it is essential to figure out the structure of ear and fundamentals of human hearing. In figure 2.8 the human ear structure is shown briefly.

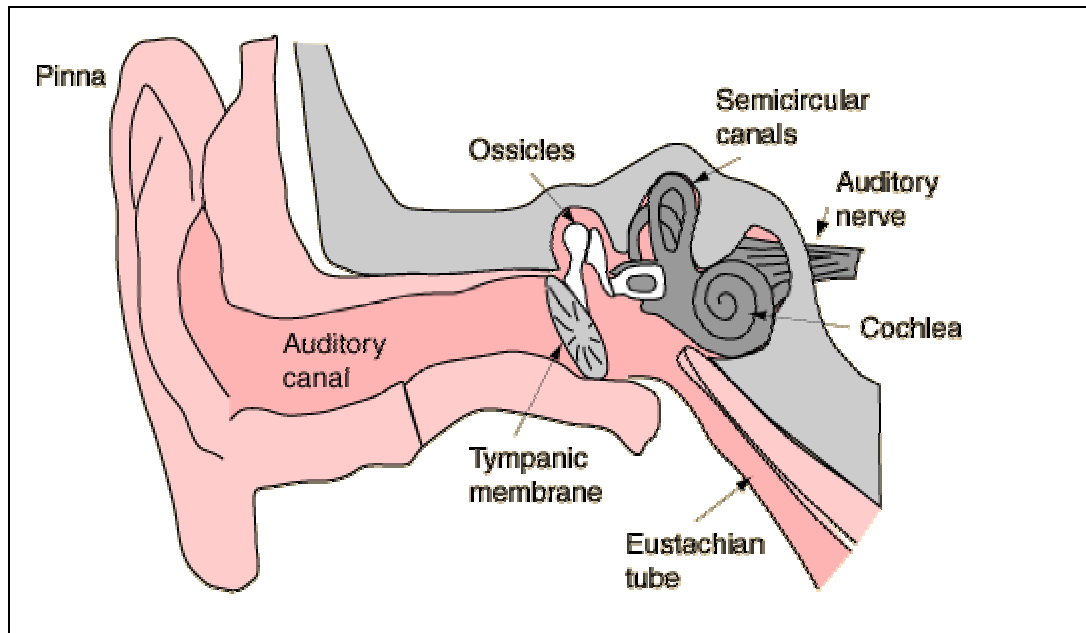


Figure 2.8. Structure of Human Ear

When audible sound waves approach to the ears, the pinna directs the sound waves to the tympanic membrane (ear drum) through the auditory canal, which acts like a tube and has its own physical quantities such as resonance frequency. Therefore, it performs certain amount of amplification to the frequencies roughly between 2 – 5 kHz. The ear drum receives the vibrations coming through the auditory canal and due to the pressure difference between the outer ear and the inner ear, the vibrations are transmitted to the middle ear by the vibrations of the ear drum (see figure 2.9).

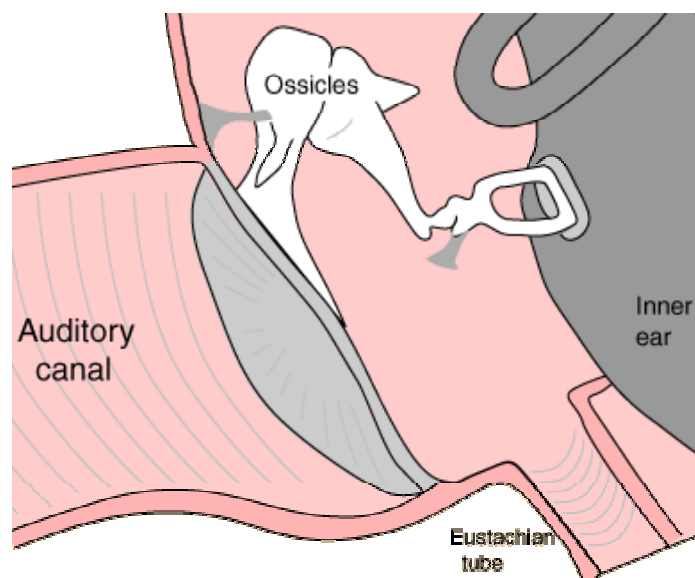


Figure 2.9. Vibration of the ear drum by the sound waves coming through the auditory canal.

Similarly, the sound waves reach to the three small bones inside the middle ear (the hammer, anvil, and stirrup) and travel to the cochlea. Within the complex structure of the inner ear and cochlea, the sound waves are converted into pulses by the auditory nerves and send to the brain's hearing centre.

2.1.10.2. Limits of Human Hearing and Perception

Acoustics deal with a wide range of frequencies in different areas of application; however, there are certain restrictions in human hearing. Therefore, in psychoacoustics, the audible frequencies are under consideration.

The frequency range that is audible by the human ear ideally lies between 20 Hz. - 20 kHz. With increasing age, hearing capability dramatically decreases especially at high frequencies 1kHz per decade [4].

Taking the threshold of hearing and the threshold of pain as upper and lower boundaries, the hearing range can be scaled between 0 dB – 140 dB noise levels [4]. Table 2.1 shows the scale of noise occurring in daily life [4]. This can also be named as the dynamic range of hearing.

Table 2.1. Relation between absolute sound pressure and sound pressure level.

Sound Pressure P [N/m ² , rms]	Sound Pressure Level L _p [dB]	Situation / Description
$2 \cdot 10^{-5}$	0	Hearing Threshold
$2 \cdot 10^{-4}$	20	Forest, slow winds
$2 \cdot 10^{-3}$	40	Library
$2 \cdot 10^{-2}$	60	Office
$2 \cdot 10^{-1}$	80	Busy Street
$2 \cdot 10^0$	100	Pneumatic Hammer, siren
$2 \cdot 10^1$	120	Jet plane during take-off
$2 \cdot 10^1$	140	Threshold of pain, hearing loss

Another fact on human hearing is the ear's sensitivity on different frequencies and perceived equal loudness. This fact introduces the necessity of frequency weighting in acoustic signal processing, which will be explained in details in the following sections.

Ear's perception of a sound is described with a term of loudness. Depending on the sound pressure level and the frequency information contained in sound waves, ear's sensitivity changes as it can be seen in figure 2.10

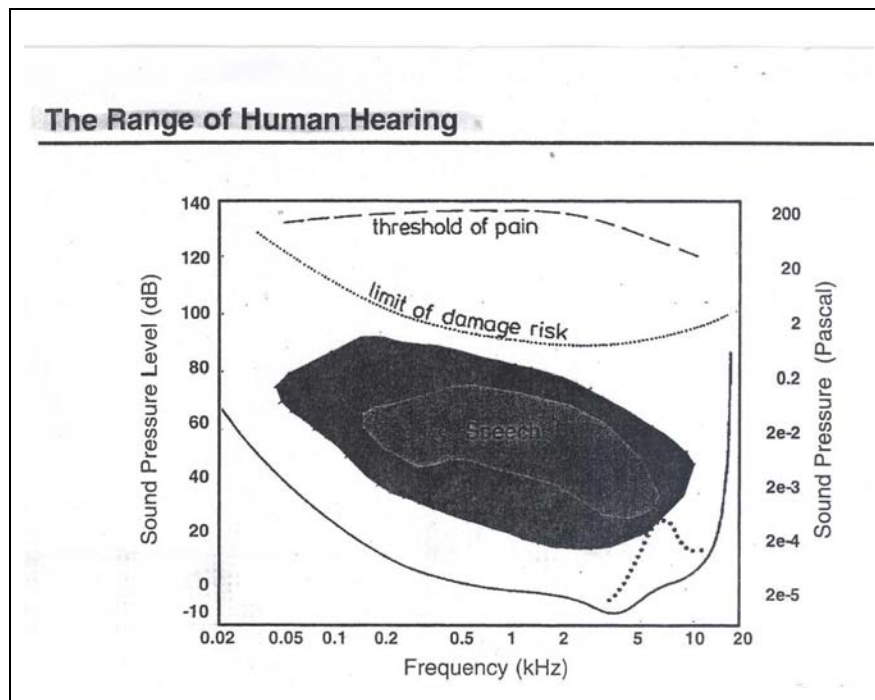


Figure 2.10. The Range of Human Hearing

As it can be seen in figure 2.10, ears do not show the same sensitivity to all frequencies. To be able to hear the low frequencies compared to middle and high frequencies, higher sound pressure levels are required. Acoustic frequency weighting networks are defined in signal processing in order to characterize ear's frequency response as close to reality as possible.

Many other animals hear a much wider range of frequencies compared to human beings. For example dog whistle frequency is higher than human hearing range. Dolphins and whales communicate at very high frequencies beyond human hearing (ultra sound). Figure 2.11 gives some examples on the hearing range of several animals [9].

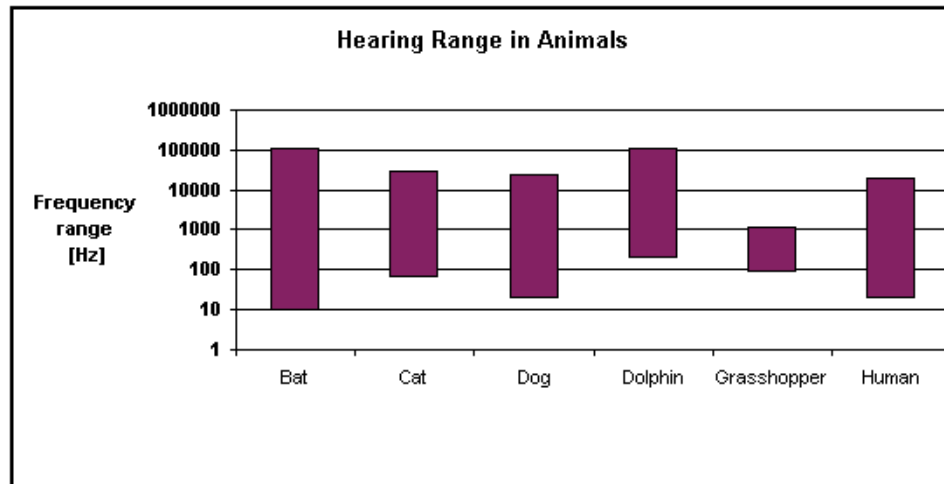


Figure 2.11. Hearing Range in Animals.

2.1.10.3. Perceived Loudness

Loudness is a term to define the psychological perception of a sound, or in other words human perception of the strength of a sound. Loudness is expressed by the units of “Phon” and “Sone”.

Due to the fact that human ear does not treat equally to all frequencies; the equal loudness curves have been developed to identify perceived loudness.

Subjective loudness, N , has a linear relation with loudness level, L_N and there is no frequency dependency as shown in equation 2.12 [2].

$$N = 0.046 \cdot 10^{\frac{L_N}{30}} \quad (2.12)$$

Equal loudness curves have been developed in a way that 1kHz. tone of a certain sound pressure level is taken as reference. Other random sounds are then compared relative to the reference loudness of 1 kHz and adjusted such that it is subjectively perceived as loud as the reference [4]. Equal loudness curves are shown in figure 2.12, where x axis is frequency and y axis is the intensity in decibels.

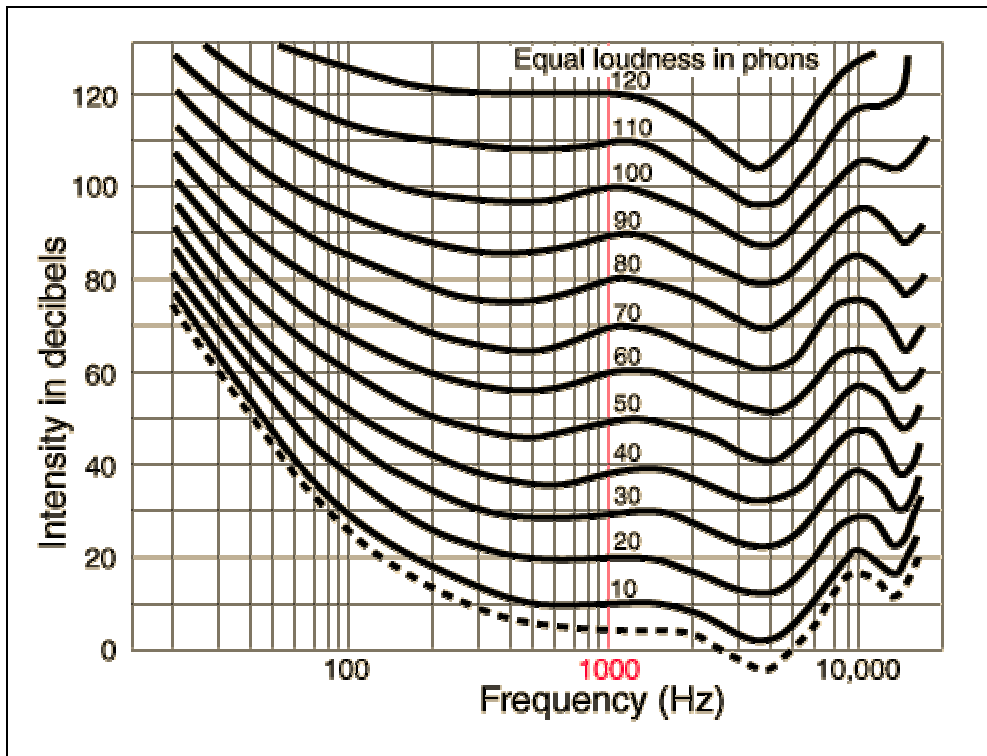


Figure 2.12. Equal Loudness Curves

As seen in figure 2.12, low and high frequencies require greater sound intensity levels compared to mid frequencies. This is a result of human ears being more sensitive to mid frequencies due to the fact that the auditory channel resonance is around 2 – 5 kHz. and amplifies the sound waves at those frequencies. That is, the minimum power required to make a 30 Hz. tone audible is nearly a million times more than that of a 4 kHz tone [2].

Each of these curves is equal loudness contour and represents the loudness level in phons. An equal loudness curve is assigned to all tones, at which the sound intensity level fall on the contour [2]. For example, a tone at 250 Hz. having an intensity level of 60 dB is perceived as loud as a tone at 2 kHz. having 60 dB intensity level, and both tones have a loudness level of 62 phons. There is a rule of thumb for loudness, that is, the perceived loudness doubles when the sound intensity level is increased by a factor of 10 dB. This is not valid for very low and high sound pressure levels where the 10 dB rule no more applies.

There is also a relation between loudness and intensity, which can be described with an empirical formula as in equation (2.13). At 1kHz. the loudness level is said to be equal to the intensity level [2].

Combining equation 2.12 and $L_1 = 10 \cdot \log\left(\frac{I}{10^{-12}}\right)$ equation 2.13 can be obtained [2].

$$N_{1\text{kHz}} = 460 \cdot I^{1/3} \quad (2.13)$$

Over time, another unit has been developed, sone, which enables creation of a linear scale for loudness. An example calculation of loudness level in phons and loudness in sones can be viewed in table 2.2 [2]

Table 2.2. Sample calculation of loudness and loudness level.

Frequency [Hz.]	Intensity Level [dB]	Loudness Level [phon]	Loudness[sone]
125	60	55	3.2
250	60	62	5.4
500	60	63	5.9
1000	60	60	4.7
2000	60	62	5.4
4000	60	69	9.3
Total			33.9

2.1.10.4. Weighting Networks

Sounds occurring in daily life (in the offices, construction areas, public roads or in a vehicle interior, etc.) usually contain a broadband of frequencies. Due to the facts of the human hearing and perception of sound, weighting networks have been introduced in acoustic signal processing in order to conduct correct evaluation of the measured data. This enables engineers to correctly determine the existence of a noise problem, or to achieve realistic noise development in many areas of application.

In acoustic signal processing, several weighting networks have been defined based on the sensitivity of human ear in different frequencies and sound intensity levels. The use of one of the weighting networks labelled as A, B, C and D depends on the place of the noise of interest in ear sensitivity map. The weighting filters, in other words, have certain attenuation levels, which are approximated based on the frequency response of human ear. Choice of a weighting filter strongly depends on the area of application. In vehicle NVH development, A and B weighting are the most commonly used filters; however, in applications where high noise levels are subject to consideration, *such as air craft, jet engine noise*, C and D weighting filters are used.

The most widely used metric of environmental noise is the A-weighted sound level and is expressed as dB(A) [2]. The attenuation character of the A-weighting filters is based on the fact that human ear is not as sensitive to low frequencies as it is to high frequencies. As a consequence, low frequencies and very high frequencies are devaluated compared to the mid frequencies [4]. 1 kHz is the reference value in all filters and does not apply any reduction to the sound of interest. The A-weighting curve follows the 40 phon equal loudness curve. An approximate calculation of the A-weighting as a function of frequency is given in equation 2.14 [9].

$$W_A = 10 \log \left[\frac{1.562339 f^4}{(f^2 + 107.65265^2)(f^2 + 737.86223^2)} \right] \tag{2.14}$$

$$+ 10 \log \left[\frac{2.24288 \cdot 10^{16} f^4}{(f^2 + 20.598997^2)^2 (f^2 + 12194.22^2)^2} \right]$$

The A-weighting function is standardized in EN 60651. In figure 2.13, the frequency response functions (attenuation characteristics) of the A, B, C and D weighting filters can be viewed [8].

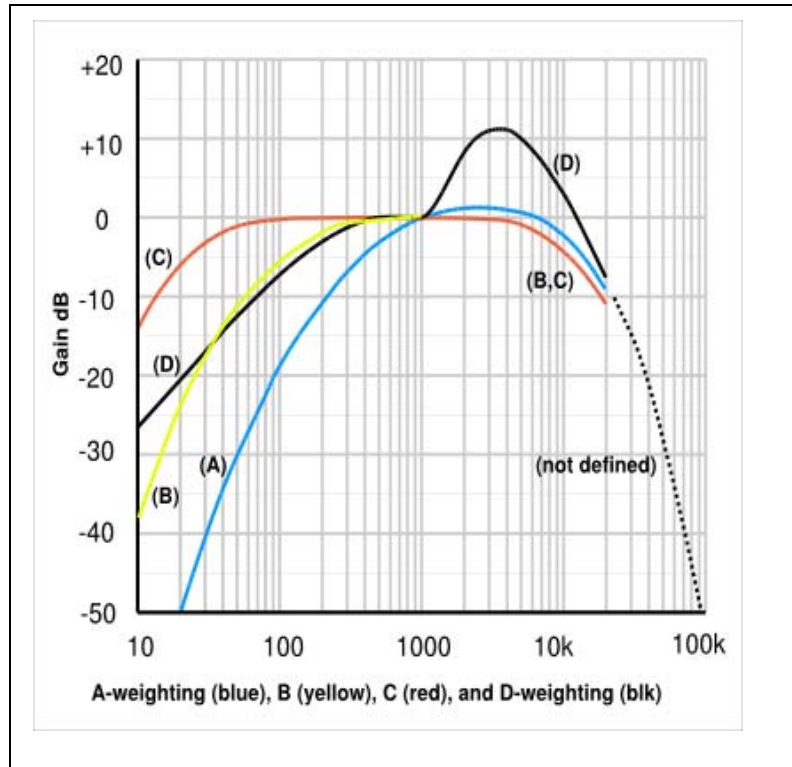


Figure 2.13. Frequency Response Functions of A, B, C and D weighting filters.

In table 2.3, the A-weighting attenuation values for octave band centre frequencies are given [2].

Table 2.3. A-weighting attenuation values for octave band center frequencies

Octave Band Centre Frequency [Hz.]	Attenuation Level [dB]
31.5	-39.4
63	-26.2
125	-16.1
250	-8.6
500	-3.2
1000	0
2000	+1.2
4000	+1
8000	-1.1

2.1.10.5. Octave and 1/3 Octave Band Filters

Sounds we hear in daily life generally involve broad band frequencies. In order to identify the frequency components of a sound, there are certain techniques depending on the area of interest. For example, when designing a resonator for a specific frequency, high spectral resolution would be needed in order to detect the narrow-band resonance peaks of a signal [4]. In such cases, the Fast Fourier Transform (FFT) analysis is a commonly used technique. In this study, the theory of FFT analysis will not be explained in details. As a brief background, FFT is a technique, which assumes that a periodic signal is the summation of sine and cosine functions over a T sampling period. Further information could be found in the references.

In cases, where high spectral resolution is not desired to decompose time domain signals, the method developed is to divide the frequencies into proportional widths. For example, to analyse the frequency components of an interior sound recording of a vehicle idling at a certain RPM, narrow-band frequency analysis may not be required. In acoustic signal processing, very commonly used filters are the octave and 1/3 octave band filters. In $1/n^{\text{th}}$ octave band analysis, each band covers a specific range of frequencies while excluding the other. This is not only beneficial because narrow-band analysis does not always contain useful information in all cases and is time-consuming; but it is also relevant to ears frequency resolution, that ears do not equally perceive the change in every single frequency.

In $1/n^{\text{th}}$ octave band analysis, the defining parameters are the upper, lower and the centre frequencies of each band. The relation between these parameters is shown in equation 2.15, 2.16 and 2.17 [6].

$$f_c = \sqrt{f_l \cdot f_u} \quad (2.15)$$

$$f_u = 2^{\frac{1}{n}} \cdot f_l \quad (2.16)$$

where, n defines the octave band interval of interest.

$$\Delta f = f_u - f_l \quad (2.17)$$

The filter characteristics can be calculated from the limiting frequencies of the bandwidth of interest. In figure 2.14, the frequency response function of a band-pass filter is given [4].

Octave Bandwidth:

$$f_c = \sqrt{2}f_l \tag{2.15a}$$

$$f_u = 2f_l \tag{2.16a}$$

$$\Delta f = f_u - f_l = f_l = f_c / \sqrt{2} \tag{2.17a}$$

Third-octave Bandwidth:

$$f_c = \sqrt[3]{2}f_l = 1.12f_l \tag{2.15b}$$

$$f_u = \sqrt[3]{2}f_l = 1.26f_l \tag{2.16b}$$

$$\Delta f = 0.26f_l \tag{2.17b}$$

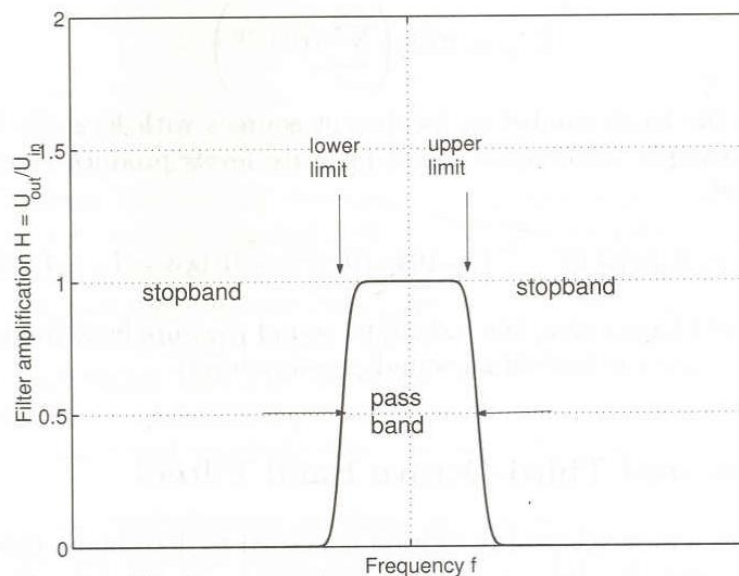


Figure 2.14. Typical Frequency Response Function of a band-pass Filter

In table 2.4, centre frequencies and bandwidths for the preferred octave and 1/3 octave bands are given [2].

Table 2.4. Center frequencies and bandwidths for the preferred octave and 1/3 octave bands.

<i>Center Frequency (Hz)</i>		<i>10log(Bandwidth)</i>	
<i>Octave</i>	<i>1/3-Octave</i>	<i>Octave</i>	<i>1/3-Octave</i>
16	10	10.5	3.6
	12.5		4.6
	16		5.7
	20		6.6
31.5	25	13.4	7.6
	31.5		8.6
	40		9.7
63	50	16.5	10.6
	63		11.6
	80		12.7
125	100	19.5	13.6
	125		14.6
	160		15.7
250	200	22.5	16.7
	250		17.6
	315		18.6
500	400	25.5	19.7
	500		20.6
	630		21.6
1000	800	28.5	22.7
	1000		23.6
	1250		24.6
2000	1600	31.5	25.7
	2000		26.7
	2500		27.6
4000	3150	34.5	28.6
	4000		29.7
	5000		30.6
8000	6300	37.5	31.6
	8000		32.7

2.1.10.6. Articulation Index

Articulation Index is a psychoacoustic parameter, which is used to determine the degree of privacy in an area like a room or a vehicle interior. The speech intelligibility depends on the level and frequency of the background noise. The maximum articulation index (AI) percentage is 100 %, meaning that the speech is completely intelligible; the lower the AI percentage, the more difficult to communicate in an area and 0 % is the minimum AI percentage.

The calculation of the articulation index percentage is based on the human hearing fundamentals. The range of human speech and human hearing limitations determine the basics of the % AI calculation.

The principle of articulation index calculation is as follows [9]:

1. If A-weighted 1/3 octave level lies between upper and lower limits then it will be a linear value between 0 and 1.
2. If value falls above the upper limit then result = 0 for that particular 1/3 octave band.
3. If value falls below the lower limit then result = 1 for that particular 1/3 octave band.
4. Multiply all of the calculated values by the AI weighting and sum all the values to get the AI in percentage terms.

2.2 Fundamentals of Vibration

In the simplest way, vibration can be defined as an oscillating motion. In other words, something is said to vibrate when it is oscillating about an equilibrium position. Vibration of an object is always caused by an excitation force. This force can either be applied externally, or can be originated inside the structure. When vibration motion occurs without externally applied forces, it is referred as free vibration, such as when an elastic system is given a displacement or an internal energy as an impact [7]. When external forces are given to the structure, it is referred as forced vibration. Both free and forced vibrations can be classified as undamped, viscously-damped and hysterically-(or structurally) damped oscillations [1]. Another way of classifying vibration is the number of degrees of freedom of the system. The number of degree of freedom of a system corresponds to the number of independent coordinates that is required to fully describe the motion [7]. It is not always very representative way to model a structure as a Single-Degree-of-Freedom (SDOF) system, but it is of great importance to know and use SDOF system physical properties, because more realistic Multi-Degree-of-Freedom (MDOF) systems can be modelled as superimposed SDOF systems [1].

The basic model of a single-degree-of-freedom system is illustrated in figure 2.15.

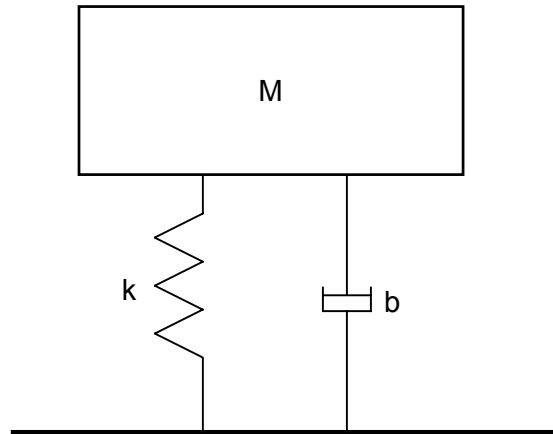


Figure 2.15. Basic model of a SDOF system.

2.2.1 Undamped Free Vibration

As mentioned previously, if there is no external force applied to the system, $f(t) = 0$, then the system will make a free vibration, in which the motion will be due to an initial disturbance. For a single-degree-of-freedom system, the governing equation for the vibration motion will be as shown in equation (2.18)

$$m\ddot{x} + kx = f(t), \text{ where } f(t) = 0 \quad (2.18)$$

Here, the spring is considered as elastic and obeys Hooke's law. Therefore, the spring force will change linearly by displacement and is calculated by kx [7]

2.2.2 Undamped Forced Vibration

When the basic SDOF spring mass system is considered under a certain harmonic force, $F(t)$, the equation of motion will be as in (2.19):

$$-kx + F(t) = m\ddot{x} \quad (2.19)$$

$F(t)$ is referred as the harmonically varying force in radians per second, and F_1 is the maximum value in its sinusoidal motion. Combining the sinusoidal expression of the harmonic force with equation (2.19);

$$m\ddot{x} + kx = F_1 \sin \omega t$$

(2.20)

$$\ddot{x} + \frac{k}{m}x = \frac{F_1}{M} \sin \omega t$$

Solving the differential equation,

$$x = A \cos \omega_n t + B \sin \omega_n t + \frac{F_1}{m(\omega_n^2 - \omega^2)} \sin \omega t \quad (2.20a)$$

In this equation, the last term is the steady-state term and will not be affected by the initial conditions in the system [2]. Therefore, when the first two transient terms are eliminated from the equation,

$$x = \frac{F_1}{m(\omega_n^2 - \omega^2)} \sin \omega t \quad (2.20b)$$

From the basics of the harmonic motion, the maximum displacement will be:

$$X = \frac{F_1}{m(\omega_n^2 - \omega^2)} \quad (2.20c)$$

$$\frac{X}{F_1/k} = \frac{1}{1 - \frac{\omega^2}{\omega_n^2}}$$

If the external force is in an exponential form of $F(t) = F_1 e^{i\omega t}$, then the equation of motion can be written as follows:

$$\ddot{x} + \frac{k}{m}x = \frac{F_1}{m} e^{i\omega t} \quad (2.21)$$

Applying the solution of the differential equation,

$$x = A \cos \omega_n t + B \sin \omega_n t + \frac{F_1}{m(\omega_n^2 - \omega^2)} e^{i\omega t} \quad (2.21a)$$

The steady-state term in the equation (2.21a) referring the maximum displacement term X , the equation can be written as,

$$\frac{X}{F_1/k} = \frac{1}{1 - \frac{\omega^2}{\omega_n^2}} \quad (2.22)$$

Another form of the external force may be as $F(t) = m_0 \omega^2 e \sin \omega t$, which would come from the excitation of a unbalanced rotating system [2]. Following the similar pattern with the previous examples, the equation of motion can be written as:

$$x = \frac{m_0 \omega^2 e}{m(\omega_n^2 - \omega^2)} \sin \omega t \quad (2.23)$$

$$X = \frac{m_0 \omega^2 e}{m(\omega_n^2 - \omega^2)}$$

2.2.3 Viscous Damping

Considering the basic SDOF spring and mass system as illustrated in figure 2.15, if a dashpot is attached to the mass, which is assumed to be negligible, then the system will be subjected to viscous damping. The opposite force coming from damping will be proportional to the velocity of the system and can be written as:

$$F = -bv \quad (2.24)$$

When the damping effect is included, the equation of motion of a basic free and SDOF system can be written as:

$$m\ddot{x} + b\dot{x} + kx = 0 \quad (2.25)$$

The viscous damping force is created by the flow of a fluid through a lot, or around a piston in a cylinder. The very important and fundamental background in viscous damping is that the damping force is not frequency dependent and the level is

proportional to velocity. Moreover, the viscous damping force is opposite to the direction of velocity and therefore, takes a negative sign as in equation 2.24.

Applying the general solution $x(t) = X e^{st}$ (s is not imaginary but complex) [1]

$$(ms^2 + bs + k) = 0 \quad (2.26)$$

$$s_{1,2} = -\frac{b}{2m} \pm \frac{\sqrt{b^2 - 4km}}{2m} \quad (2.27)$$

$$= -\omega_n^2 \zeta \pm i\omega_n \sqrt{1 - \zeta^2}$$

In this equation,

$$\omega_n^2 = \frac{k}{m} \quad ; \quad \zeta = \frac{b}{b_0} = \frac{b}{2\sqrt{km}}$$

2.2.4 Structural Damping

Viscous damping is widely used in structural dynamics modelling and it was beneficial as being a linear equation. However, in real structures, especially in MDOF systems, the commonly used viscous dashpot is not a good representative for the real damping in the system. The required damping model should be the one, whose rate varies with frequency opposite of a viscous damper [1]. As all structures have a rate of internal damping due to the hysteretic properties of the material, and representing the internal damping by a viscous damper leads to serious error [2].

The force deflection characteristics of material hysteresis and viscous damping is shown in figure 2.16

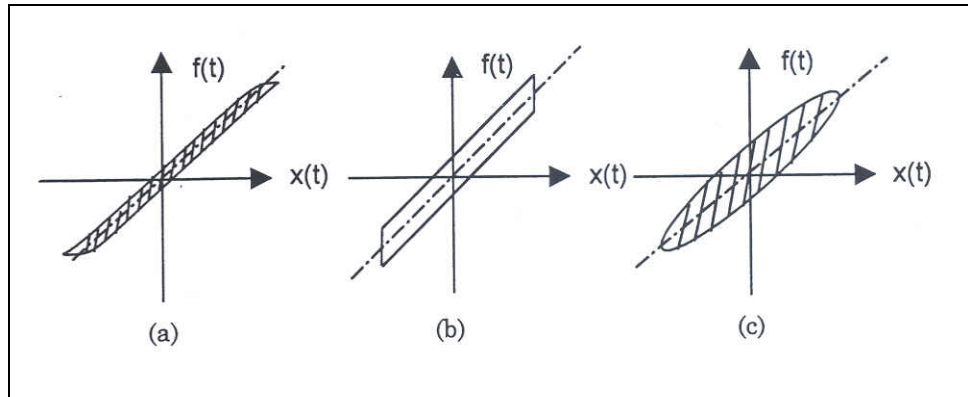


Figure 2.16. Force deflection characteristics.

a) Material Hysteresis b) Dry friction c) Viscous damper

When the equation of motion is written including the alternative structural (or hysteretic) damping model,

$$(-\omega^2 m + k + id)X e^{i\omega t} = F e^{i\omega t} \quad (2.28)$$

$$\frac{X}{F} = \alpha(\omega) = \frac{1}{(k - \omega^2 m) + i(d)}$$

(2.29)

$$\alpha(\omega) = \frac{1/k}{(1 - (\omega/\omega_n)^2 + i\eta)}$$

In equation 2.29, η is the structural damping loss factor and replaces the previously used critical damping ratio, ζ , used to describe a viscous damping model.

2.2.5 Frequency Response Function (FRF)

The Frequency Response Function (FRF) is a very commonly used term, which is used to express the behaviour of a system under a certain input. In other words, FRF is the ratio of the output to the input of the system.

Typical and very commonly used output quantities in vibration are displacement, velocity and acceleration. Therefore, there are several ways to express the transfer function or the frequency response function of a system.

When velocity is used as the output of a system, the frequency response function is referred as mobility and can be expressed as:

$$Y(\omega) = \frac{Ve^{i\omega t}}{Fe^{i\omega t}} = \frac{V}{F} \quad (2.30)$$

Acceleration can also be used as the response parameter, then the frequency response function is referred as accelerance or inertance and we can write the FRF as [1]:

$$A(\omega) = \frac{A}{F} = -\omega^2 \alpha(\omega) \quad (2.31)$$

The frequency response functions can also be written in the inverse forms and represent several physical quantities. In table 2.5, definitions of frequency response functions are given [1].

Table 2.5. Definition of Frequency Response Functions

Response Parameter, R	Standard FRF, R/F	Inverse FRF, F/R
Displacement	Receptance Admittance Dynamic Compliance Dynamic Flexibility	Dynamic Stiffness
Velocity	Mobility	Mechanical Impedance
Acceleration	Accelerance Inertance	Apparent Mass

Typical frequency response function curves of a single-degree-of-freedom system are shown in figures 2.17 – 2.19 [2].

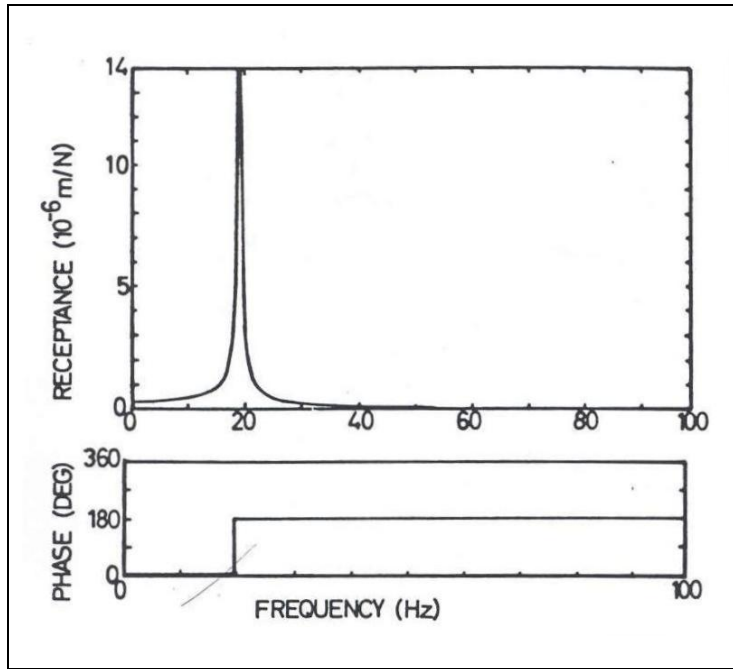


Figure 2.17. Receptance FRF plot for undamped SDOF system.

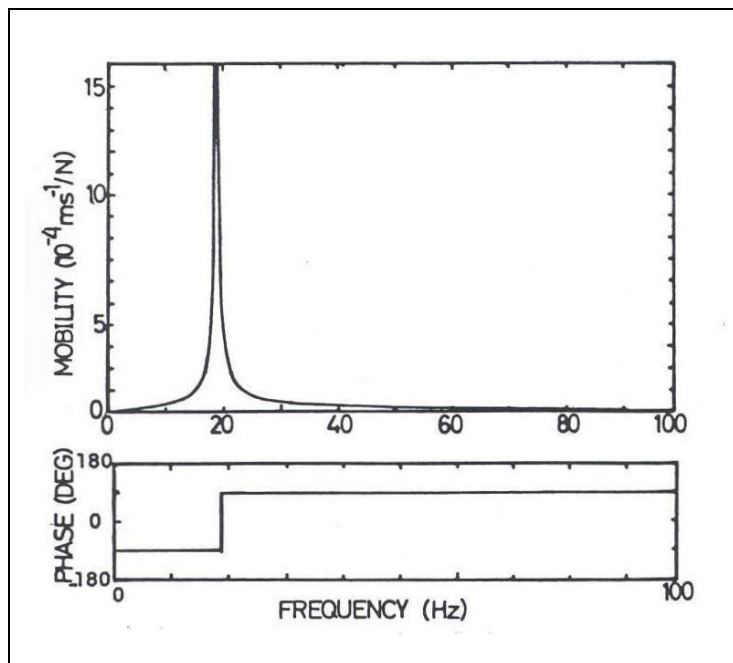


Figure 2.18. Mobility FRF plot for undamped SDOF system.

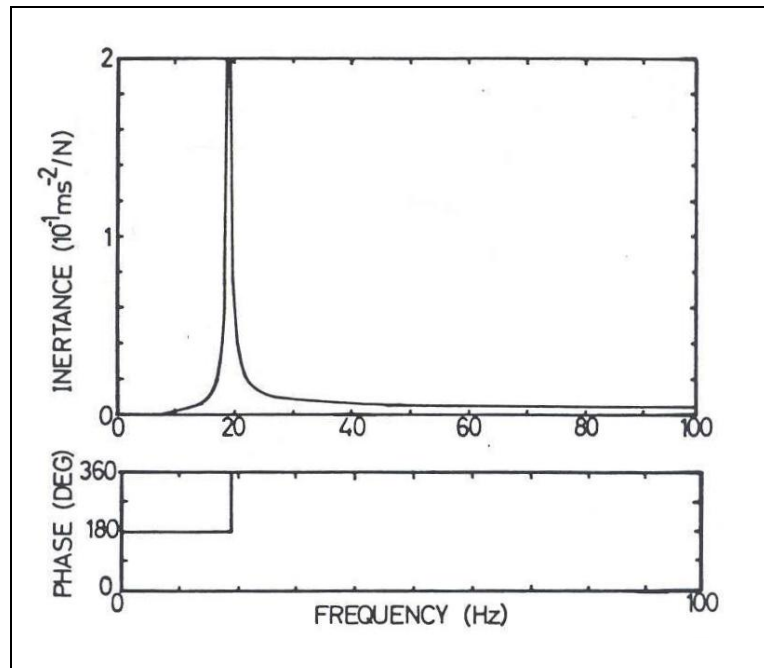


Figure 2.19. Inertance FRF plot for undamped SDOF system.

2.3 Fundamentals of Vehicle NVH

Up to now, theoretical basis of general acoustics and vibration is explained. It is now intended to cascade the fundamentals of noise and vibration to vehicle NVH fundamentals, and the principles in vehicle NVH design and development.

There are plenty of mechanisms and systems used in daily life, which are served for the comfort of humans. This means there are various areas of application like HVAC, biomechanics, aeronautics, automotive and etc. All of those applications include complex systems and sub-systems that need to be engineered in several perspectives. In automotive design, there are several of attributes and sub-attributes, which affect customer impression of a vehicle. Those can either be soft attributes like the interior trim quality, multi-functionality, interior view and etc. or attributes like vehicle dynamics, performance feel, packaging & ergonomic, NVH, fuel economy and etc. It is for certain that engineering one of those attributes individually may not always work, due to the fact that changing a parameter to optimize one may harm the other. This means, there will be a strict trade-off in the vehicle development phases, and it is one of the most challenging business of automotive engineers to balance the performance all attributes.

During its operation, a vehicle is under the effect of various environmental and operational excitations. Therefore, in vehicle NVH development, engineers need to

take into account of variety of parameters and optimize vehicle vibration and acoustics accordingly, such that vehicle will comfortably be usable in all driving conditions (Like cruising on a highway, a city drive will lots of drive-aways or low speed creeping, coarse road drives, etc.)

When the decision is given by the senior management to kick-off a new vehicle program, one of the most important and strategic phases during development is benchmarking. In this phase, the competitor products are investigated attributes wise, and based on the marketing strategy, the DNA of the vehicle under development becomes clearer. Following step, in terms of NVH, is to determine the vehicle level NVH targets, which will lead all program decisions till the launch of the vehicle. Although, vehicle NVH quality will be judged by the vehicle level sound and vibration performance at the end of the day; it is essential firstly to identify system and sub-system level targets properly, so that the vehicle level targets would be achieved.

Overall vehicle NVH performance is considered in the combination of three main sub-attributes, which are; Powertrain NVH, Road NVH and Wind NVH. All these sub-attributes need different design methodologies and strategy and developed by addressing mostly different system and sub-systems in the vehicle. As an example, developing wind NVH requires a well aerodynamic body design providing less turbulent air flow, very good sealing strategy from proper door settings to proper expandable sealers positioning. Whereas, in Powertrain NVH development, the engine mount dynamic stiffnesses, noise transfer functions, body sensitivity etc will be more effective parameters. Similarly, road NVH development requires proper dynamic stiffness selection for the suspension bushes and a good body sealing. As it is seen, all sub-attributes in vehicle NVH development are affected mainly from different aspects; therefore not only a balance is needed for other attributes like NVH, vehicle dynamics, performance et, but also in between the sub NVH attributes.

Sound wave propagation from the noise source to our ears occurs in two different ways such as structure-borne and air-borne. In vehicle acoustic and vibration development studies, it is essential to determine the contribution of both structure and air-borne noise to be able to have the knowledge of which noise source is a strong contributor, or which region of the body/chassis is weaker, or is it worth to improve sound radiation from the vehicle panels, or addressing the absorption of the noise in the interior. This fundamental knowledge enables developing vehicle NVH in a more effective and in a cost-optimized way.

2.3.1 Structure-borne and Air-borne Sound

In most of the acoustical events, sound is radiated by vibration of solid objects. During combustion, internal combustion engines radiate sound by the vibrations of the engine block and this noise is transferred in air, through the dash panel and the openness of the vehicle to the driver's or passenger's ear. This way of sound transmission is investigated under the subject air-borne sound. Another way of sound radiation from the internal combustion engine, for example, is due to structure-borne excitations. Combustion and the free moments of inertia of the rotating objects produce certain amount of force within the engine structure in a wide frequency range, which is transferred to the body panels through the engine mounts and chassis. This results in low frequency sound radiation from the body panels and driveline and is referred as structure-borne sound. The word "borne" is the past participle form of the verb "to bear" which means "to carry" [3]. This would be a good expression to the actual meaning of air-borne and structure-borne sound in acoustics. The intention here is to provide a fundamental background of what air-borne and structure-borne is and the transfer path to the vehicle interior. Therefore, the details of wave propagation and stress distributions in solids will not be explained in details. Interested readers may refer to references [4] and [5] for further details on structure-borne sound.

2.3.2 Noise Sources and Transfer Paths in Vehicles

As already mentioned in the previous sections, vehicle during its operation is subjected to various excitations from either its components or the environment. While those excitations would be harmonic in the simplest way, they are usually random depending on the operating conditions. This study is based on Powertrain NVH development; therefore the focus point will be the contribution of Powertrain components and the impact of the main powerplant transfer paths. Road and wind NVH is not the main intent but general aspects will be pronounced.

2.3.2.1. Powertrain NVH

Powertrain NVH covers all of the noise and vibration generation mechanisms on the power unit/gearbox through to the final drive ratio to the wheels. Noise is pronounced as the engine draws air in for the combustion process. The explosion in the cylinder is transmitted through the surface of the engine block into the air and to the vehicle interior [9]. The forces generated by the rotating parts of the engine and

due to combustion are radiated from the power unit, most dominantly from the engine mounts and sub-frame attachments. The release of the burnt exhaust gases results in a pulsating noise at the exhaust outlet. In figure 2.20 the noise and vibration sources in a vehicle is shown.

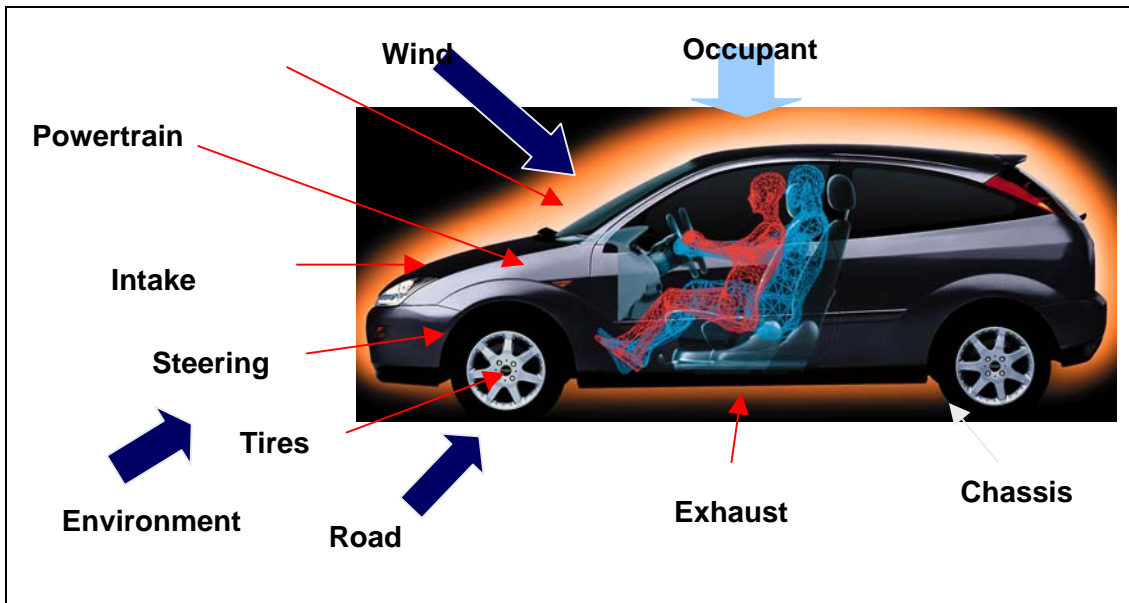


Figure 2.20. Noise and Vibration Sources in a Vehicle

2.3.2.2. Intake System Radiated Noise

The vehicle intake system performs the following main functions [9]:

- **Channel to the engine:** Intake orifice is mostly placed in the air mainstream, for example the engine bay. The intention is to provide the suction air as cold as possible for increased volumetric efficiency.
- **Filter Particulates:** The filter element filters the required volume of the air from particles and therefore it must be large enough and should achieve necessary service life.
- **Dynamic performance tuning:** The length of the pipes, connecting the valve face to the plenum, is tuned to improve the engine breathing at the desired engine speed.
- **Reduce Noise:** The air induction system should be tuned to the critical frequencies to have reduced radiated noise levels. Especially in Petrol engines, air intake system is experienced as one of the dominant contributors in overall interior noise.

A schematic of an intake and exhaust system is shown in figure 2.21.

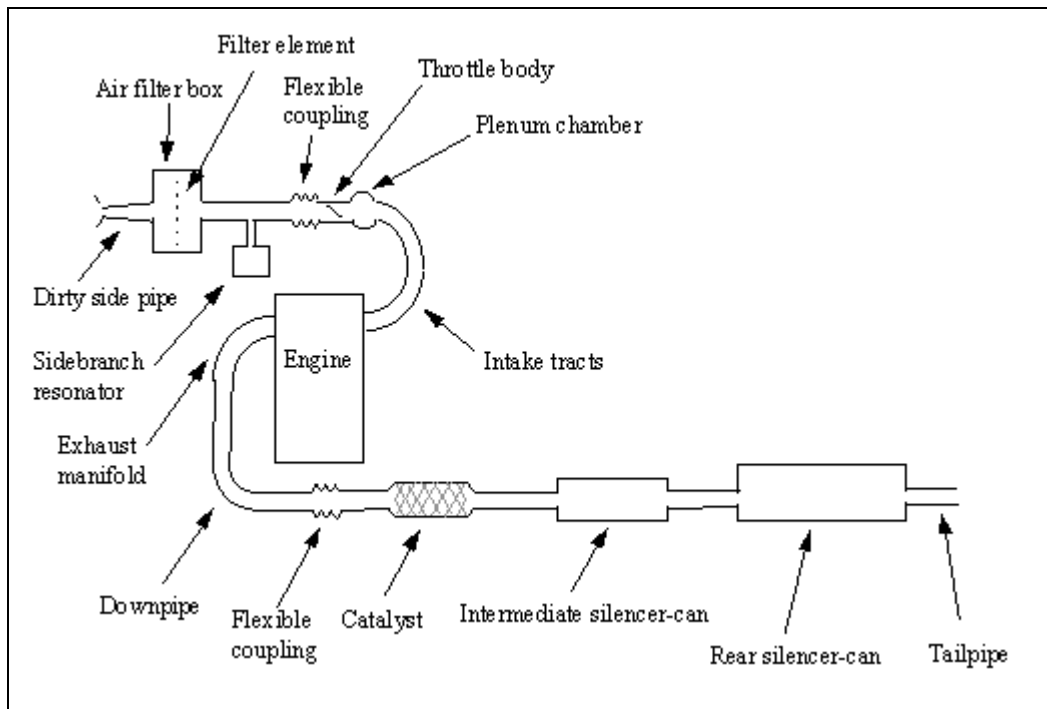


Figure 2.21. Intake and Exhaust System Components

The noise sources and transfer paths of the intake and the exhaust system are illustrated in figure 2.22.

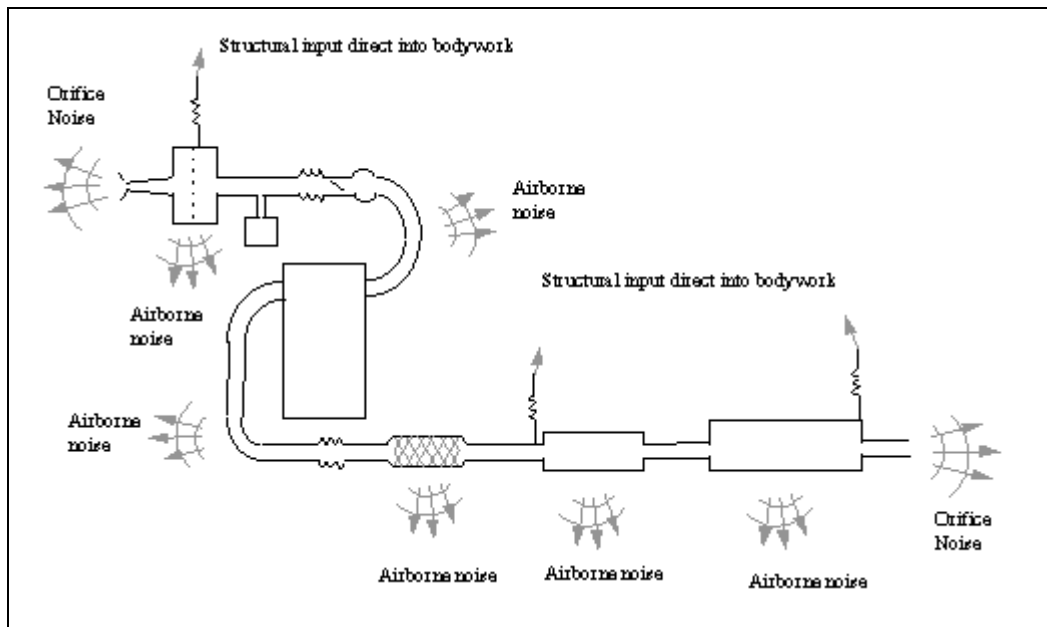


Figure 2.22. Intake and Exhaust System Noise Transfer Paths

The attenuation of the air intake noise is provided by the air filter box, which basically works as an expansion box covering a wide frequency range and a resonator, which is addressed to a specific frequency. Due to the fact that air intake system is mounted on the body at some certain locations, the oscillations during air suction causes vibrations and excite the vehicle body. In order to minimize the structure-borne excitations, flexible coupling in the air intake system is used, which provides some isolation between the engine and the point, at which the air filter box is attached to the body.

2.3.2.3. Exhaust System Radiated Noise

Exhaust system in vehicles are used to let burnt gases inside the cylinders out of the vehicle. Although, extraction of the burnt gases is the main function of the exhaust system, it should be designed in such a way that the gas emissions should be limited to a certain level directed by the legislations; vehicle performance should be less affected by proper exhaust gas back pressure tuning. In addition, the noise from the exhaust system should be treated for attenuation in several aspects due to being a strong noise contributor.

As schematically seen in figure 2.22, exhaust noise contribution to the overall vehicle interior noise happens in two ways. One is the air-borne content, which is related to gas rush out of the exhaust orifice; and also the sound radiated by the vibrations of the exhaust muffler system. Secondly, portion comes from exhaust system vibrations as they are transferred to the vehicle body from the attachment points and builds up the structure-borne contribution. Therefore, the exhaust hangers have a significant importance in the isolation of the exhaust system vibrations before exciting the body.

2.3.2.4. Engine Radiated Noise

Internal combustion engine is one of the biggest excitation sources in the vehicle. Both air-borne and structure-borne engine noise radiation should be considered in vehicle powertrain NVH development, as both occur in different frequency ranges, contribute through different paths and therefore need different NVH treatment. Noise radiated from the main structure of the engine (block, cylinder head etc.) can be split into two main categories. First can be referred as the mechanical noise generated by the engine structure response to movements of components, such as the cam shaft, valves, pistons, crankshaft etc. Second proportion comes from the engine

structure response to the gas pressure profile during combustion [9]. The air-borne noise radiation from the mentioned components is transferred in the air through the bulkhead and the openness of the vehicle body to the vehicle interior and finally to the ear of the driver and passenger. Especially in diesel engines, combustion characteristics have a great importance on the engine radiated noise. There are certain parameters, which affect the combustion efficiency such as rail pressure in common rail diesel engines, start of main injection timing, pre-ignition timing etc. Details of combustion fundamentals will not be explained in details in this thesis. Other very important engine noise contribution comes from the structure-borne excitations most dominantly through the engine and transmission mounts. Vibrations are damped up to a certain level by the rubber or the hydraulic engine mounts; however, the remaining portion of the force is transferred to the body and turned into sound pressure in the vehicle interior by the excitations of the vehicle panels and driveline components. Figure 2.23 briefly illustrates the structure-borne and air-borne engine noise paths.

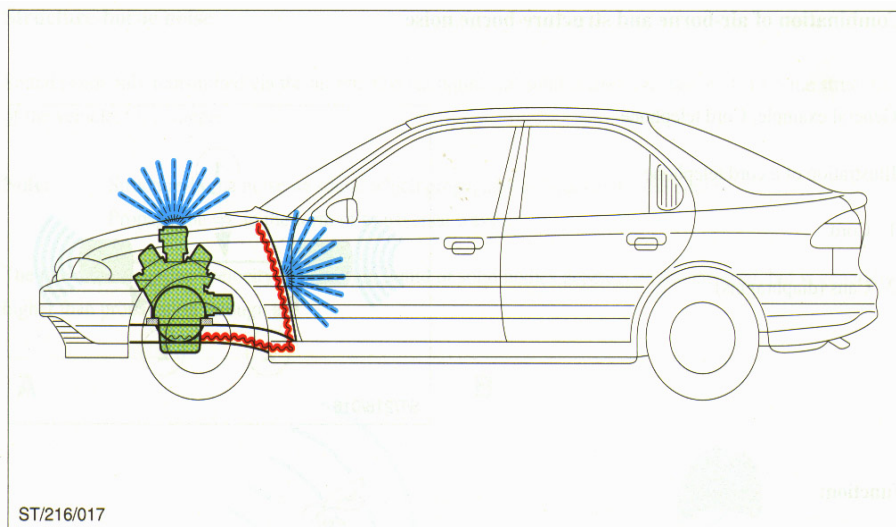


Figure 2.23. Structure-borne and air-borne engine noise radiation.

In order to understand the impact of internal combustion engine, being the most important exciter, on overall vehicle NVH; the fundamentals of engine excitation mechanism and the engine order definition will firstly be considered.

An NVH order can simply be defined as a sinusoidal phenomenon at a frequency, which is a fixed multiple of the rotation speed of the rotating source producing the phenomenon [9]. Order is expressed in multiples of the underlying periodic process such as engine crank shaft rotation speed, gear mesh, firing frequency and etc.

The n^{th} order rotation frequency of a rotating component can be calculated by;

$$f_n = \frac{\text{RPM}}{60} s \quad (2.32)$$

where,

f_n is the rotation frequency

s is the number of order

RPM is the revolution of the rotating component per minute.

In order to determine the amplitude of these sinusoids as the rotation speed varies, the method order tracking is used. A typical three dimensional waterfall plot enables seeing all the spectra simultaneously as seen in figure 2.24.

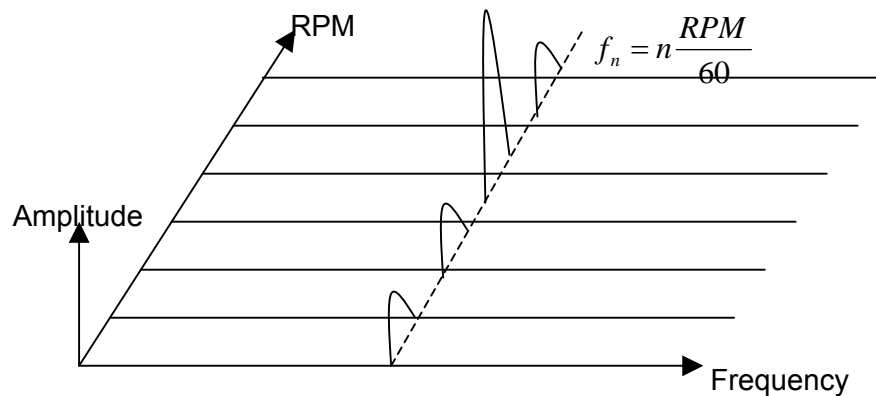


Figure 2.24. Waterfall Diagram

Engine orders have a very important role in the identification of most of the engine-driven NVH issues. Engine orders are referenced to crankshaft rotation speed. In case of an N cylinder four-stroke internal combustion engine, there will be $N/2$ firing pulses per revolution. The result of the firing pulses is a firing torque, which can be decomposed as sinusoids. The force due to the moment of inertia of the rotating components is also an important contributor to engine excitation.

Multiples of 0.5 orders are produced due to cylinder to cylinder firing pressure variations. Deflections of the engine block or bearing produce the primary orders 0.5. Also the deflection of the crank produce primary orders 1 [9].

Table 2.6 summarizes the engine orders and their sources briefly.

Table 2.6. Engine orders and their definitions/sources.

Engine Order	Source / Description
2, 4, 6...	Primary firing torque on I4s
3, 6, 9...	Primary firing torque on I6s and V6s
4, 8, 12...	Primary firing torque on V8s
½, 1, 3/2...	Orders due to cylinder to cylinder firing pressure variation, unequal exhaust or intake runners
Primary ± 1/2	Orders due to local deflection near a cylinder
Primary ± 1	Orders due to crankshaft bending

In order tracking it is essential to determine the important orders, which are relevant to the mechanics of the system under investigation. For example, one must know that a four cylinder four-stroke internal combustion engine produces its most dominant excitation forces in 2nd order and its multiple orders. Another important subject is to accurately select the order resolution, so that important orders will be separated from each other. Moreover, setting the appropriate frequency resolution provides accurate measure of the peaks in the important orders together with an appropriate sweep rate selection. That is, the optimum sweep rate should be selected to be able to analyse the data accurately. For example, if the vehicle is subjected to wide open throttle acceleration in 1st gear, then the RPM sweep will be so fast that it is going to be difficult to determine and separate the peaks in the frequency domain data. Whereas, if for example, 2nd or 3rd gear wide open throttle acceleration is selected as the measurement method, each important peak can be detected more accurately.

2.3.2.5. Body Air-borne Noise Transparency

Although it is of great importance to decrease the excitations in the source, it is also very beneficial to have a less transparent body. That is, decreasing the interior noise levels for good sound quality design depends also on the level of body noise attenuation. If the engine is very quiet but the transmission loss of the bulkhead is poor especially at high frequencies, than the benefit of a quieter engine will not be

reflected to the overall interior noise level due to poor attenuation provided by the body. Therefore, vehicle NVH engineers pay special attention to cascade appropriate noise reduction targets. Noise reduction can be defined as the attenuation factor for air-borne noise and is given versus frequency, which is very similar to noise transfer function development for structure-borne noise attenuation. Noise reduction levels in the frequency range of interest should be defined for local regions of the vehicle where the noise source exists. In powertrain NVH development; intake, exhaust and engine noise reduction levels are measured from the relevant locations, such as the engine compartment, the intake and exhaust orifices.

3. VEHICLE INTERIOR NOISE SOURCE CONTRIBUTION AND TRANSFER PATH ANALYSIS

Up to now, the fundamentals of acoustics and vibration is described to provide a basic background. The steps and the important parameters in vehicle NVH development are discussed. In this study, the intention is to calculate the overall interior noise of a vehicle by summing up the individual contributions from each sub-system, with the interior noise source contribution and transfer path analysis. Vehicle overall noise level is also actually measured to make a comparison of the measured and the calculated values. Finally, a critical peak level is investigated throughout the measured and calculated data and potential solution opportunities are discussed including the approximate investment costs and development time requirements.

The interior noise contribution and transfer path analysis method starts off with sub-system level investigations in the relevant operating condition and extends the individual contributions to vehicle level. Therefore in this study, each structure-borne and air-borne noise sources and paths are investigated by certain measurements, which are going to be discussed in details in the following sections.

3.1 Selection of the Test Method

Selection of the appropriate test methodology is as important as selecting the correct data acquisition and signal processing parameters in vehicle NVH testing. In this study, the intention is to calculate the interior noise of a commercial vehicle during its worst case road load condition, which is the wide open throttle acceleration. As it enables sweeping all engine speeds, from creeping to the maximum, the powertrain NVH performance of the vehicle is measured in the complete operating RPMs. Third gear is selected for the wide open throttle run-up test. At this point, the question of “Why specifically 3rd gear is selected?” can be raised in a common sense. This can be explained in the simplest way, that 1st gear run-up would be completed unnecessarily fast and similarly the 2nd gear. This means the time resolution would be insufficient for identification of the critical peaks

and engine orders. 4th gear would be the one to pronounce the critical order excitations and the peaks; however, this time the acceleration of the vehicle would be too slow causing increase in measurement time, larger files to be stored in the hard disk driver; and most importantly difficulties in keeping the vehicle creep properly in its idling RPM. In order to eliminate the errors mentioned, 3rd gear is selected for this full throttle acceleration test.

3.2 Test Environment

In addition to the measurement technique, the test environment that the measurements will take place plays an important role in gathering accurate and reliable measurement data. It is obvious that the background noise level should be under certain values (at least 10 dB lower than the minimum level under investigation to have ~0.5 dB measurement tolerance), so that it will not influence the actual vehicle noise to be measured.

In this study, the vehicle NVH testing took place in a semi-anechoic chassis dynamometer. It is of great importance to highlight that the customer is going to hear and feel what the vehicle performs on the road. However, in vehicle NVH development phases, semi-anechoic chassis dynamometers are widely used, as they permit keeping all other parameters constant such as humidity, ambient temperature, wind – road irregularities, etc. and let us determine the NVH impact of different designs. In addition, there is certainly a good correlation between the dynamometer and the test track measurements from Powertrain NVH point of view.

3.3 Data Acquisition Hardware and Software

In this study, the NVH measurements are conducted with the Head Acoustics 20-channel data acquisition system. Brüel & Kjaer's transducers are used such as microphones, accelerometers and the impact hammer. The main signal processing software used is Head Acoustics Artemis; and additional in-house software is used for the calculation of interior noise contributions.

To be able to calculate the noise reduction spectra, high frequency and the low frequency sound sources of ISVR, University of Southampton are used.

3.4 Measurement Details

Overall vehicle level interior noise levels, under certain operating conditions, are derived from the summation of the excitation from each sound source multiplied by a transfer function. The measurement tree used in this study can be seen in table 3.1

Input x Transfer Function = Output

- Radiated Noise x Noise Reduction = Air-borne Output
- Excitation Force x Noise Transfer Function = Structure-borne Output

Table 3.1. Interior Noise Contribution and Transfer Path of Each Source

OVERALL INTERIOR NOISE			
Structure-borne Noise Contribution			
		POWERPLANT	Powerplant Mount Forces in x, y, z
			Powerplant Mount Noise Transfer Functions in x, y, z
		SUSPENSION	Suspension Attachment Point Forces in x, y, z
			Suspension Attachment Point Noise Transfer Functions in x, y, z
		EXHAUST	Exhaust Hanger Attachment Point Forces in z (Only z direction is critical)
			Exhaust Hanger Attachment Point Noise Transfer Functions in z
Air-borne Noise Contribution			
		ENGINE	Engine Radiated Noise
			Engine Noise Reduction
		INTAKE	Intake Orifice Radiated Noise
			Intake Orifice Noise Reduction
		EXHAUST	Exhaust Orifice Radiated Noise
			Exhaust Orifice Noise Reduction

It is of great importance to determine independent structure-borne and air-borne noise sources and their transfer paths in the vehicle. In most of the cases, omitting a noise source is not a common issue in NVH testing. Whereas, the critical issue, which would affect the accuracy and the correlation, is the selection of the correct independent excitation sources. Unless independent points are investigated, two or

more sources can be summed up affecting the correlation of the calculated levels, which is usually a matter of structure-borne excitation points. Not to make any confusion, it is important to underline that in this study, the suspension system is not considered for road induced excitations, as the investigations are relevant to powertrain NVH. However, since the suspension system has a rigid connection to the engine block from the half shafts, it has to be considered precisely even if it is not directly a structure-borne noise source. Therefore, when determining the independent suspension excitation points, special attention should be paid in order not to make any double counts. The suspension system components (such as the sub-frame, L-arm and the shock tower) are attached to the body with two or more bolts; therefore, all connection points are subjected to relevant measurements and the average of all is considered in the model not to double an individual contribution.

3.5 Air-borne Noise Contribution Measurement Technique

As discussed in the previous sections, a transfer function and an actual input is required to calculate the contribution of any source. For air-borne noise contribution, the local noise reduction spectrum and the actual noise radiation from the system is required. In order to calculate the intake, exhaust and engine system air-borne noise contributions, the following steps are followed:

3.5.1 Intake System Air-borne Noise Contribution

Intake system radiated noise is measured during 3rd gear wide open throttle acceleration with a microphone, which is located perpendicularly and 10 cm. away from the centre of the orifice. The intake orifice is raised; however it is still made to keep its connection to the engine. Therefore, the background noise from the engine had to be attenuated in a certain level such that its influence on intake orifice noise should be neglected. For this matter, engine top surface is covered with heavy layers and thick felt absorbers. Keeping the measurement location as 10 cm from the intake orifice also provided good conditions to attenuate the engine radiated noise as much as possible. Moreover, a microphone windscreen is used in order to eliminate the contribution of high-frequency air flow noise. Figure 3.1 illustrates the intake orifice radiated noise measurement location.

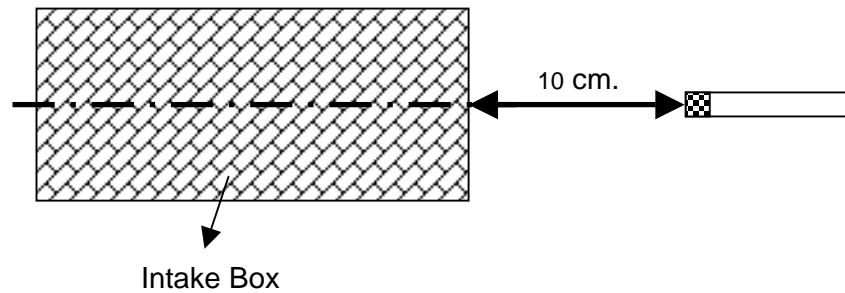


Figure 3.1. Intake Orifice Radiated Noise Measurement Location

Second step in the calculation of the intake system radiated noise is to measure the intake orifice noise reduction spectrum. For this purpose, a high frequency point source is used to excite the vehicle body with white noise from the intake orifice position. The noise radiated from the point source is measured in the interior of the vehicle (driver outer ear position) and the attenuation of the vehicle to intake system is calculated as a spectrum.

3.5.2 Exhaust System Air-borne Noise Contribution

Exhaust system radiated noise measurements are carried out with a similar technique as it is in intake system measurements. In practice, exhaust system air-borne noise radiation is both from the muffler shell and the orifice outlet. However, in this study only the exhaust orifice outlet noise radiation is taken into account, due to being more dominant compared to muffler shell noise radiation.

As seen in figure 3.2, exhaust orifice radiated noise is measured with an angle of 45° from the centre axis, and the microphone is located 25 cm. away from the exhaust orifice. Location of the microphone is not kept closer and perpendicular to the orifice as it is the case in intake measurements, because hot exhaust gases may damage the microphone structure. Again a microphone windscreen is used to eliminate the high frequency air flow noise.

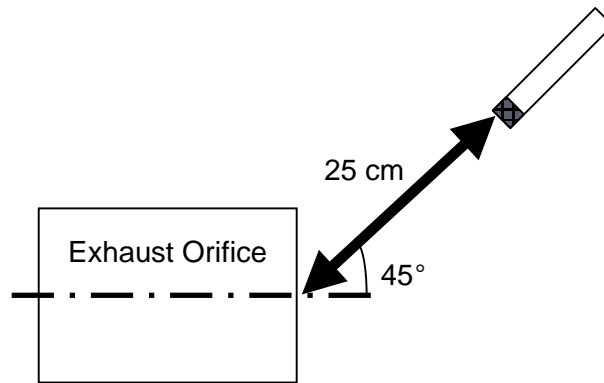


Figure 3.2. Exhaust Orifice Radiated Noise Measurement Location (Top View)

To measure the exhaust noise reduction, ISVR high frequency sound source and amplifier device is used. The point source is placed by the exhaust orifice, and white noise is played simulating the orifice input. The radiated white noise is then recorded via the interior microphone in order to calculate the difference between the output and the reference input. The schematic view of the source and the interior microphone is shown in figure 3.3.

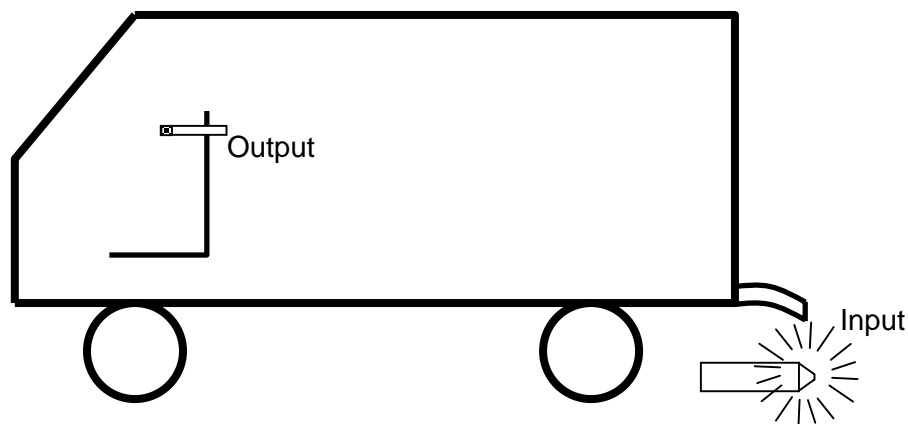


Figure 3.3. Exhaust Noise Reduction Measurement Method

The ISVR high frequency sound source and amplifier device provides near omnidirectional sound radiation up to around 8 kHz. The 1/3 octave band sound pressure levels at 1 meter are 55 dB at 200 Hz; and 85 dB at 3150 Hz. Further details can be found in the reference [10].

3.5.3 Engine Air-borne Noise Contribution

Based on the methodology used so far for identification of air-borne noise contribution of a source, the intention is to measure the actual radiation and the air-

borne transfer function (Noise Reduction) from the source to the vehicle interior. However, applying the same technique for engine system yields some difficulties and is subject to several restrictions. First of all, it is not a very applicable way to measure the engine air-borne radiated noise while the engine remains installed in the vehicle. This is because there is usually not enough space to install microphones around the engine and to satisfy free-field condition in the engine compartment for accurate measurements. For this matter, engine noise radiation is usually measured in semi-anechoic engine rigs. Opposite of this, engine noise reduction spectrum can be measured while the engine is installed. The practical method is to use the rule of reciprocity, in which the noise source is placed in the vehicle while microphones are positioned around the engine. Sufficient number of microphones should be used around the surfaces of the engine and the most appropriate data analysis should be applied, which may be the statistical average of all. However, if the engine radiated noise levels are not available from a semi-anechoic engine rig; the engine noise reduction spectrum would not be enough itself for the determination of engine system air-borne interior noise contribution.

In this study, the method used is to identify the engine noise contribution by using a low frequency sound source device's diaphragm surface acceleration as reference and combining the reference input/output with the actual engine input.

In this method, the engine volume acceleration and volume acceleration transfer function is measured, and engine air-borne contribution is calculated. In figure 3.4, the measurement devices and locations are illustrated.

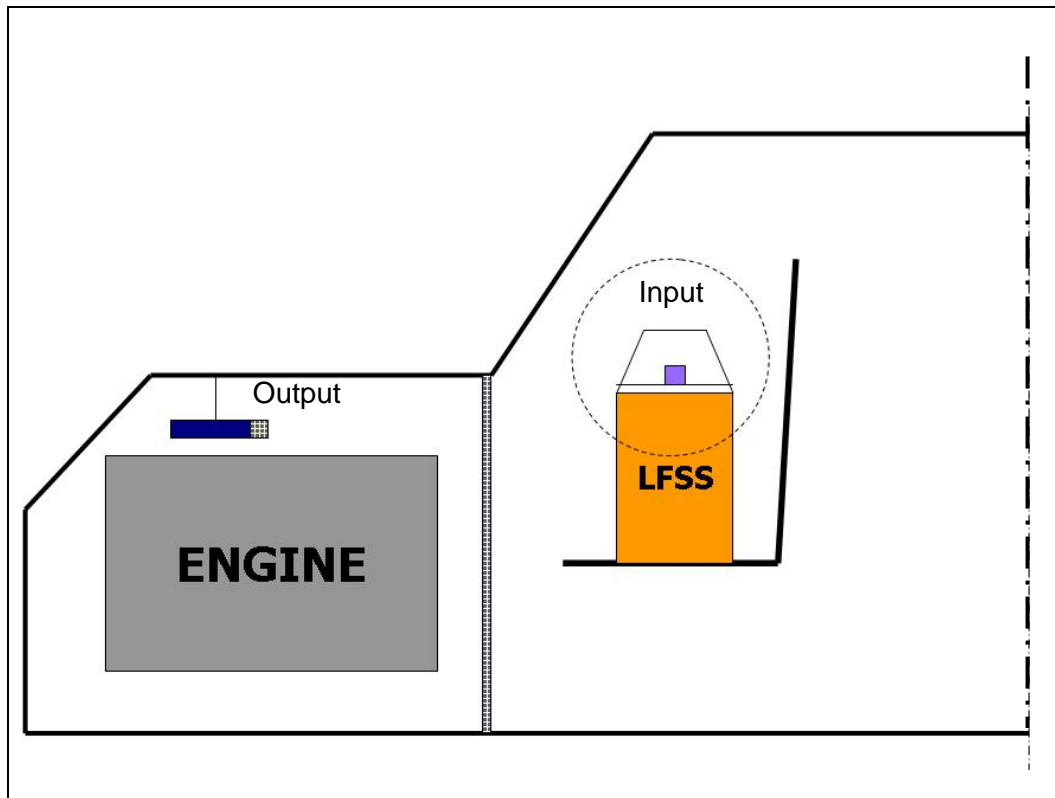


Figure 3.4. Volume Acceleration Transfer Function Measurement.

As briefly illustrated in figure 3.4, ISVR low frequency sound source (LFSS) is placed inside the vehicle and three microphones are located inside the engine compartment around the engine. More accurate values can be obtained by using more reference microphones such as two per engine surface; however, this is a time consuming technique and therefore simplified with upper and lower engine surfaces. From the rule of reciprocity, the noise radiated by the LFSS is measured via the engine compartment microphones and the values are averaged to determine the noise transmission from engine compartment to vehicle interior. Since the surface acceleration and the surface area of the diaphragm of the LFSS is known, it enables calculation of the noise transfer per reference surface acceleration. It should be noted that the LFSS diaphragm is assumed to make pure rigid body motion. Unless the diaphragm behaves rigidly and local modes contribute to noise radiation, it would not make sense to combine this input with the engine block volume acceleration. Second step is to measure the surface acceleration of the engine block for determining the actual noise contribution from the engine. While the noise transfer from the reference surface area and actual acceleration is known, the multiplication of the engine surface acceleration with the reference will yield the actual noise transmission from the engine as in equation (3.1).

$$\text{Engine air-borne noise contribution} = \frac{P_{\text{Ref}}}{a_{\text{Dia}} \cdot A_{\text{Dia}}} (a_{\text{Engine}} A_{\text{Engine}}) \quad (3.1)$$

In figure 3.5, the air-borne noise sources and their transfer paths to vehicle interior are illustrated schematically.

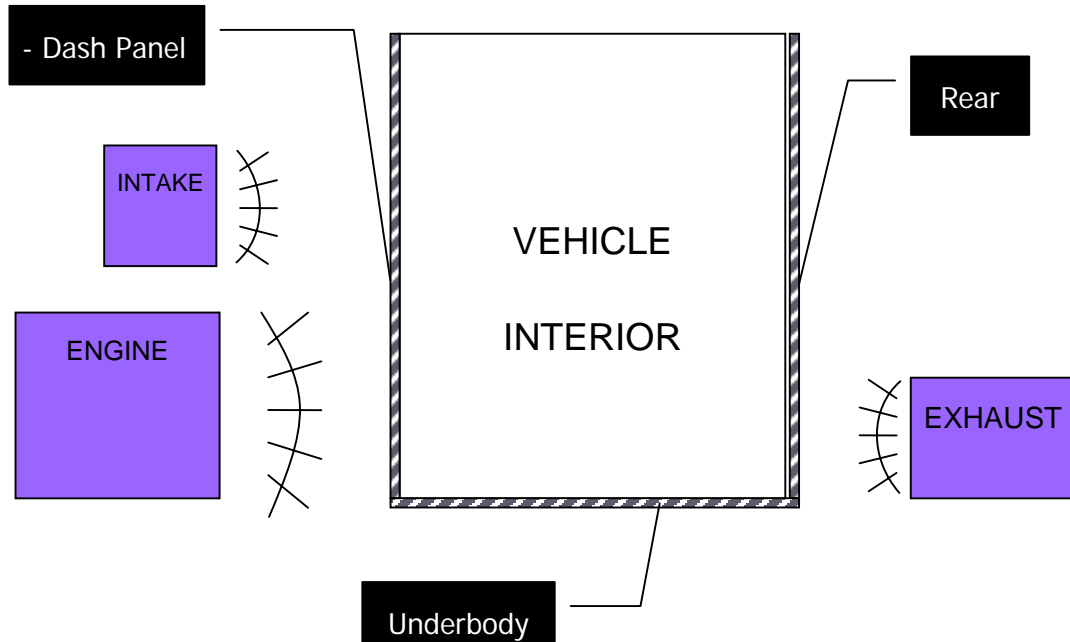


Figure 3.5. Air-borne Noise Sources and Transfer Paths to the Vehicle Interior

3.6 Structure-borne Noise Contribution Measurement Technique

As mentioned previously, each structurally transmitted contribution is the product of a force and sensitivity. Therefore, the force acting on each excitation point and the noise transfer function from each of the attachment points is measured in all directions, in order to calculate the structure-borne noise radiation. For this purpose, an impact hammer, three-axial accelerometers and a microphone is used. In point intertance and noise transfer function measurements, the calibration of the accelerometers and the impact hammer is important to be able to have correct amplitude levels. For this purpose, the accelerometers are calibrated before the measurements with B&K's vibration calibrator; and the impact hammer calibration is conducted by a hanged-mass system of 20 kg. In this system, the mass and the acceleration levels are known accurately enabling the correction of force level from dynamics basic rule of Force equals Mass times the Acceleration.

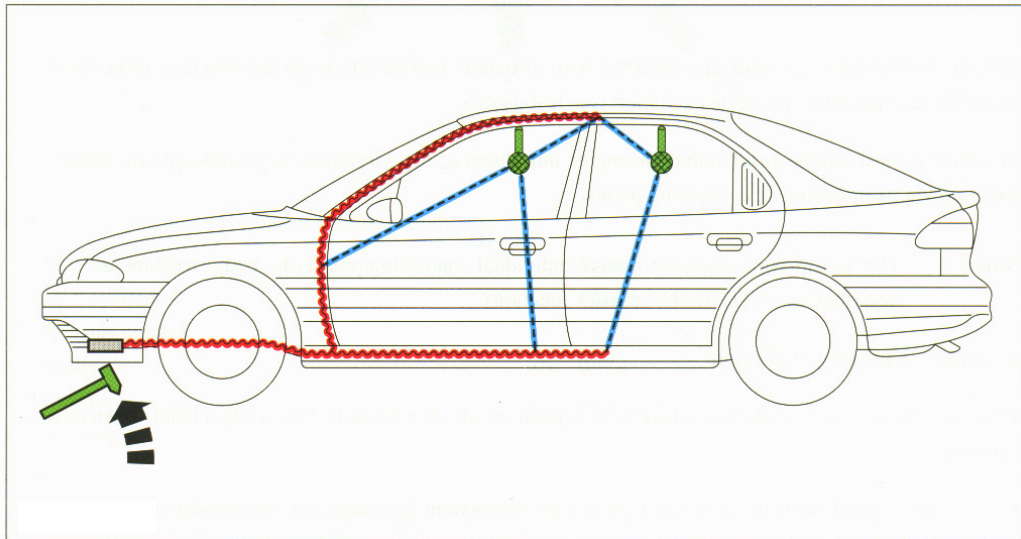


Figure 3.6. Structure-borne Noise Contribution

Noise transfer functions (NTF) in all directions are measured via the reference input applied from the impact hammer. The noise response of the vehicle is measured with the interior microphones and the ratio yields the noise transfer function from the relevant excitation point and direction. To be able to measure the force, both the accelerations and the point inertances are measured on the excitation points. Since the acceleration response of the point to a unit force is known and the actual acceleration in the operating condition is measured, the actual force can be calculated as shown in equations (3.2) – (3.4). The structure-borne noise contribution will then be the product of the force and the NTF as shown in equation (3.5).

$$\text{Point Inertance}_{Ax} = \frac{a}{F} \text{ in } \frac{\text{m/sn}^2}{\text{N}} \quad (3.2)$$

$$\text{Operational Acceleration}_{Ax} = a \text{ in } \frac{\text{m}}{\text{sn}^2} \quad (3.3)$$

$$\text{Noise Transfer Function}_{Ax} = \frac{P}{F} \text{ in } \frac{\text{Pa}}{\text{N}} \quad (3.4)$$

Structure-borne Noise Contribution from point A in x direction can be calculated as;

$$L_{Ax} = \left(\frac{a}{F} \right)^{-1} a \frac{P}{F} = P_{Ax} \text{ in Pascal} \quad (3.5)$$

3.7 Measured and Calculated Levels and Root Cause Identification

Vehicle overall sound pressure level is the sum of all contributions from each of the noise sources. By individually calculating the contribution of each noise sources and applying logarithmical addition, the overall vehicle noise can be achieved. Figure 3.7 shows the comparison of the calculated and the measured sound pressure levels in all engine RPMs swept during the 3rd gear wide-open throttle acceleration in the semi-anechoic chassis dynamometer.

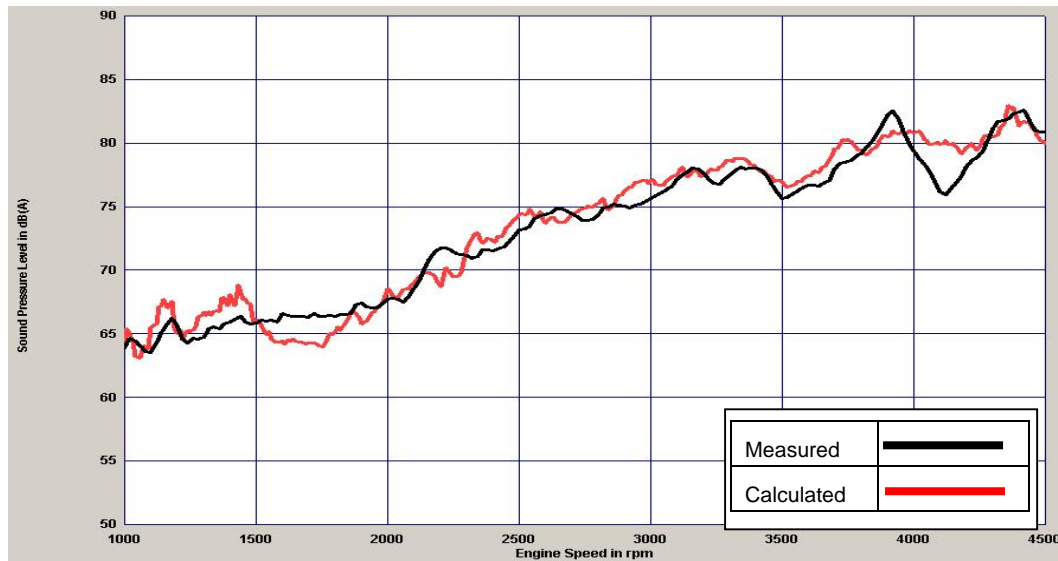


Figure 3.7. Calculated vs. Measured Sound Pressure Levels.

By comparing the measured and the calculated levels, it is concluded that good correlation is achieved with the calculation method, and therefore the interior noise contribution and transfer path analysis can be used as an efficient tool for vehicle NVH development purposes.

In the following paragraphs, one of the critical peak levels in the overall sound pressure level will be investigated. By applying the interior noise source contribution and transfer path analysis, the possible root causes of this phenomenon will be questioned and contribution in the critical points and transfer paths will be identified. Next statement will address the potential solutions to be able to overcome this vehicle level problem with approximated cost levels.

Theoretically, an ideal sound pressure level curve shall follow the pattern of a line. However, in practice this will never be the case as none of the noise sources in a vehicle will have linearly increasing noise contribution as the engine speed increases.

As illustrated in figure 3.8, both the measured and the calculated levels show a peak value at around 3900 RPM, which might be a potential indicator of a customer complaint due to poor sound quality.

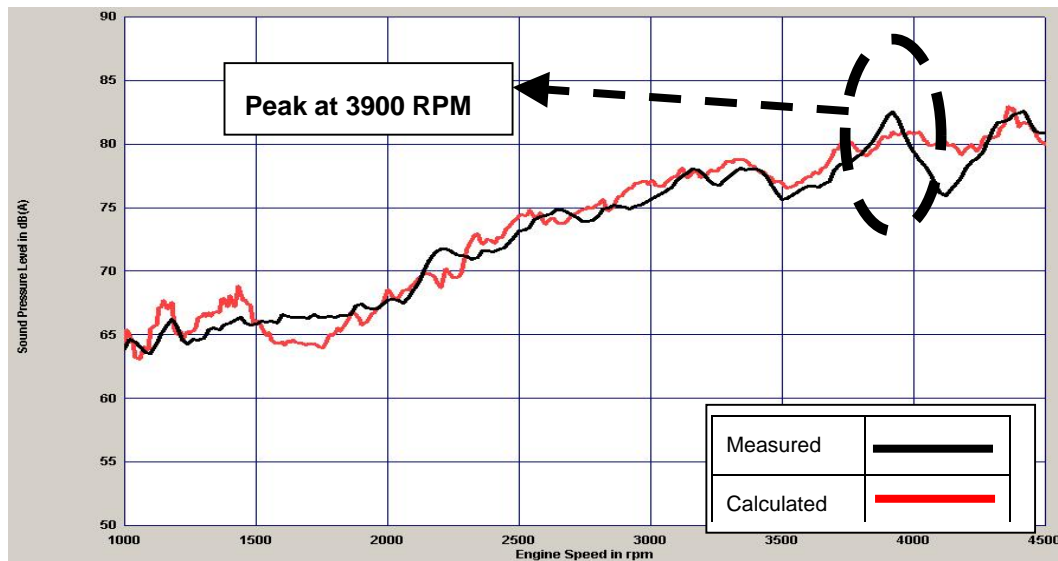


Figure 3.8. Identification of a potential NVH problem at 3900 RPM.

Since there is a good correlation between the measured and the calculated levels, it will not be a bad approach to further investigate the system and sub-system level data to search for the root cause of the problem by the interior noise contribution and transfer path analysis. However, investigating the frequency spectra of the measured data firstly would yield a better understanding on the phenomenon before digging into noise contribution and transfer path analysis. Figure 3.9 is the frequency spectra of the measured overall interior noise versus engine speed, which is gathered by applying Fast Fourier Transform.

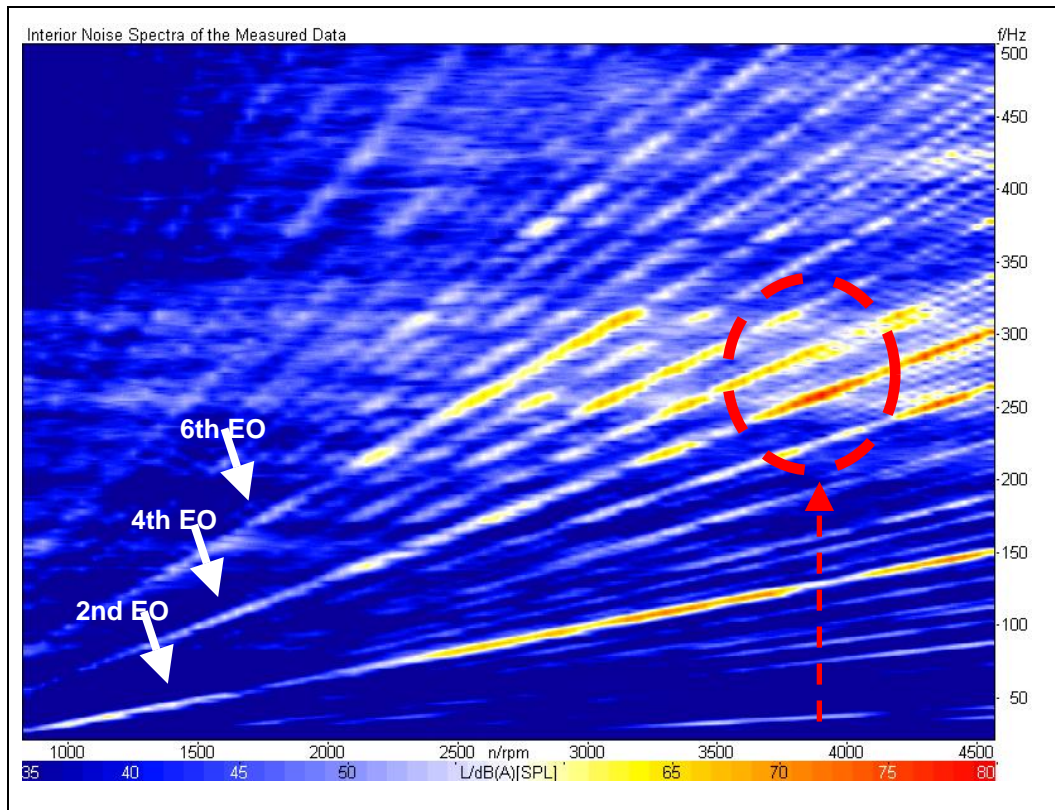


Figure 3.9. Frequency Spectra of the Measured Interior Noise

The interior noise spectra of the measured data show that the peak in the overall level at 3900 RPM is mainly dominated by the 4th Engine Order, which corresponds to 260 Hz. (can be calculated from the equation 2.32). By applying the Fast Fourier Transform and investigating the noise data in frequency domain, we now have more clues about the issue under investigation. From now on, it would be worthwhile to check whether or not the similar pattern would be reflected in the calculated data, before getting into detailed analysis on this 4th engine order phenomenon. Figure 3.10 indicates how much contribution each critical engine order has on the overall interior noise level according to the calculated data. As it seen, the overall interior noise level at 3900 RPM is mainly dominated by the 4th engine order, which is also the case for the measured values.

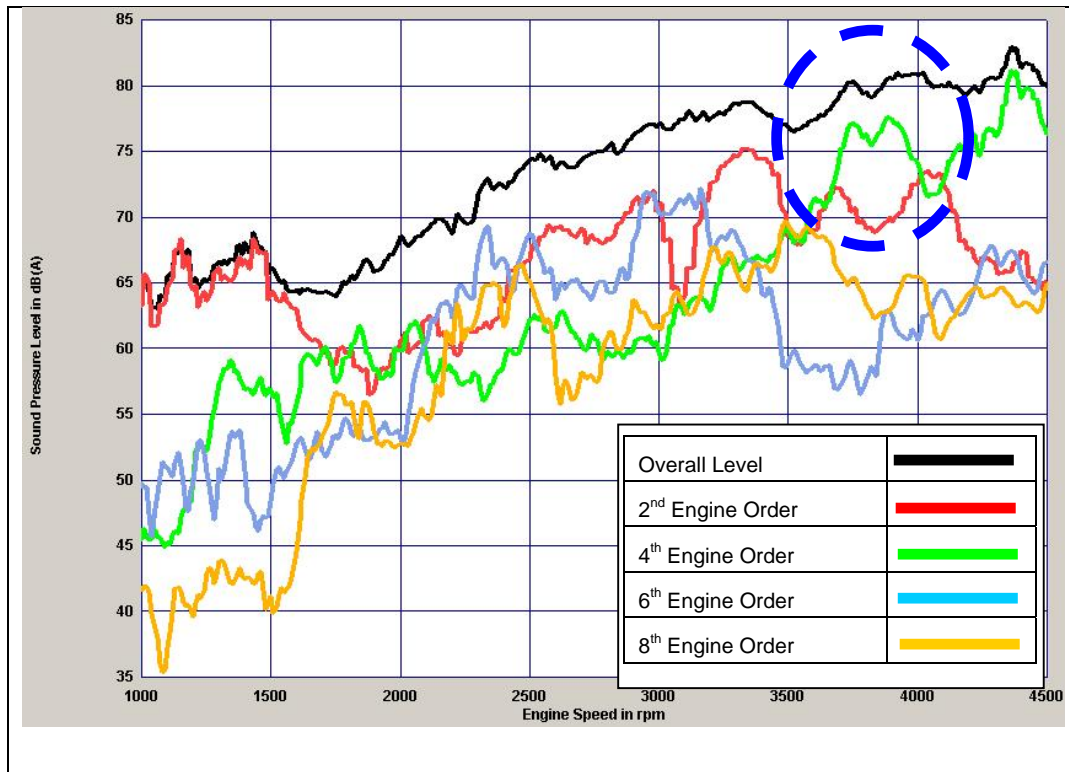


Figure 3.10. Contribution of critical engine orders on overall interior noise level – from the calculated data.

Due to the fact that the main contributor on the 3900 RPM overall interior noise level is the 4th engine order, the investigations will be focused on 4th engine order excitations from each of the noise sources and the body sensitivity around 260 Hz. As the problem is now cascaded further, a beneficial approach on the phenomenon will be to check out either the issue is radiated as structure-borne or air-borne excitation and / or transfer path. This investigation will enable eliminating some of the noise contributors, and therefore following steps in the identification of the problem will be much more clarified.

Figure 3.11 shows the structure-borne and the air-borne noise contribution in all engine speeds. To be able to calculate the air-borne contribution; the engine radiated noise and noise reduction; exhaust orifice radiated noise and exhaust noise reduction and lastly intake orifice radiated noise and intake orifice noise reduction are taken into account only and summed up. In structure-borne contribution analysis, each excitation point force is multiplied by the relevant points' noise transfer function and then each individual attachment's contribution is summed up.

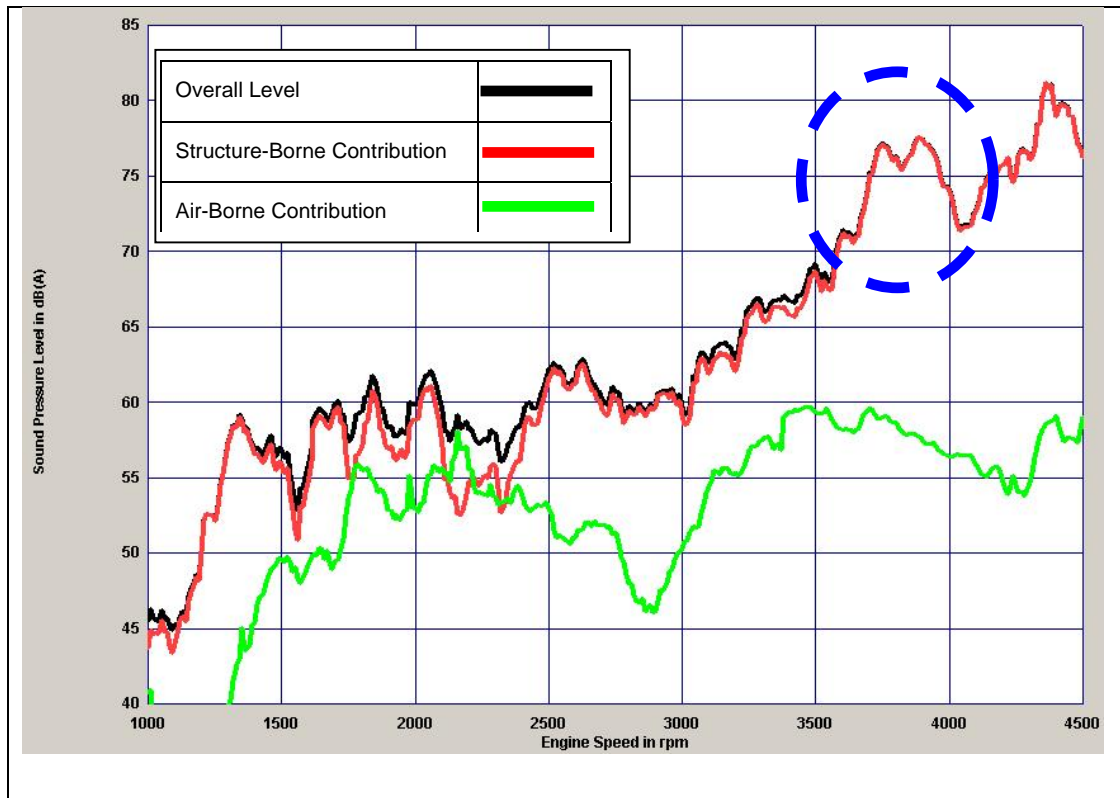


Figure 3.11. Structure-borne and Air-borne Noise Contribution

It is clearly seen in figure 3.11, that the 3900 RPM phenomenon is completely radiated as structure-borne sound. This knowledge will be very time saving as the air-borne noise sources and transfer paths will not be considered in the following investigations due to having negligible dominance. At this point, it is now well known that the efforts should be spent for identification of the structure-borne excitation points and noise paths to be able to have a better understanding.

First of all, it is essential to make a contribution analysis, which will enable determining the main contributors on the 4th order structure-borne noise. The vehicle under investigation has three points for engine – transmission system attachment; six points for suspension system attachments (such as the L-Arms, sub-frame, and shock towers) and finally three attachments for the exhaust system. Identification of the contribution of the mentioned sub-systems is highly required, in order to focus on the most dominant noise source and transfer path. In figure 3.12, the overall 4th engine order contributions of the mentioned systems are displayed against the overall structure-borne 4th engine order level.







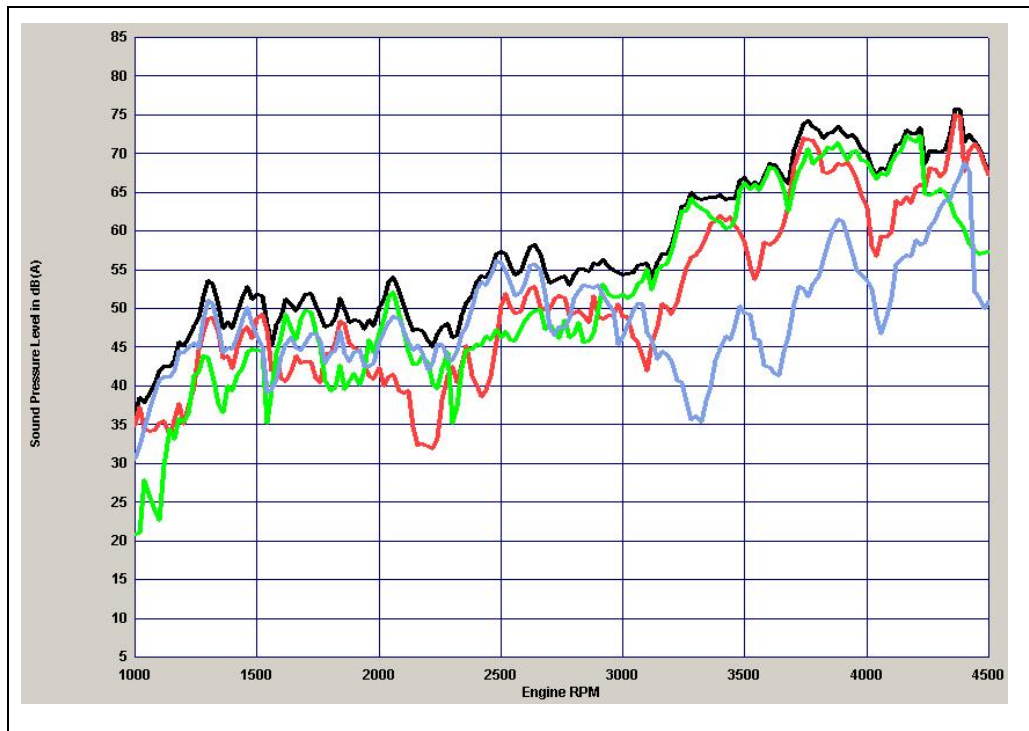
Overall 4 th EO Structure-borne Contribution	
Powerplant Mount 4th EO Structure-borne Contribution	
Suspension Attachments 4th EO Structure-borne Contribution	
Exhaust Attachments 4th EO Structure-borne Contribution	

Figure 3.12. 4th Engine Order Structure-borne contribution of all systems.

When each system's 4th engine order structure-borne contribution is considered, it is observed that at around 3900 RPM, exhaust system attachment's impact is negligible; whereas, powerplant mounts and suspension system attachments have a considerable effect on the overall 4th engine order level. This means, suspension and powerplant systems shall be investigated in sub-systems in more details. Although the mentioned systems have a bigger impact on the noise contribution, this does not mean that each of the engine mounts or the suspension attachments would have higher excitation forces acting on, or does not mean that the body is weak for the excitations from each of those attachments. There may be a single point and direction, where either most of the excitation energy is turned into structure-borne noise radiation from the body panels or there is relatively high force levels compared to the other attachment points. One other possibility would be that both phenomena could be acting together and dominating the overall 4th engine order level at this specific engine RPM. In figures 3.13 and 3.14, powertrain system

attachments and suspension system attachments are investigated in between, in order to figure out the strongest link to the interior noise and also to determine the percentage effect of each contributor.

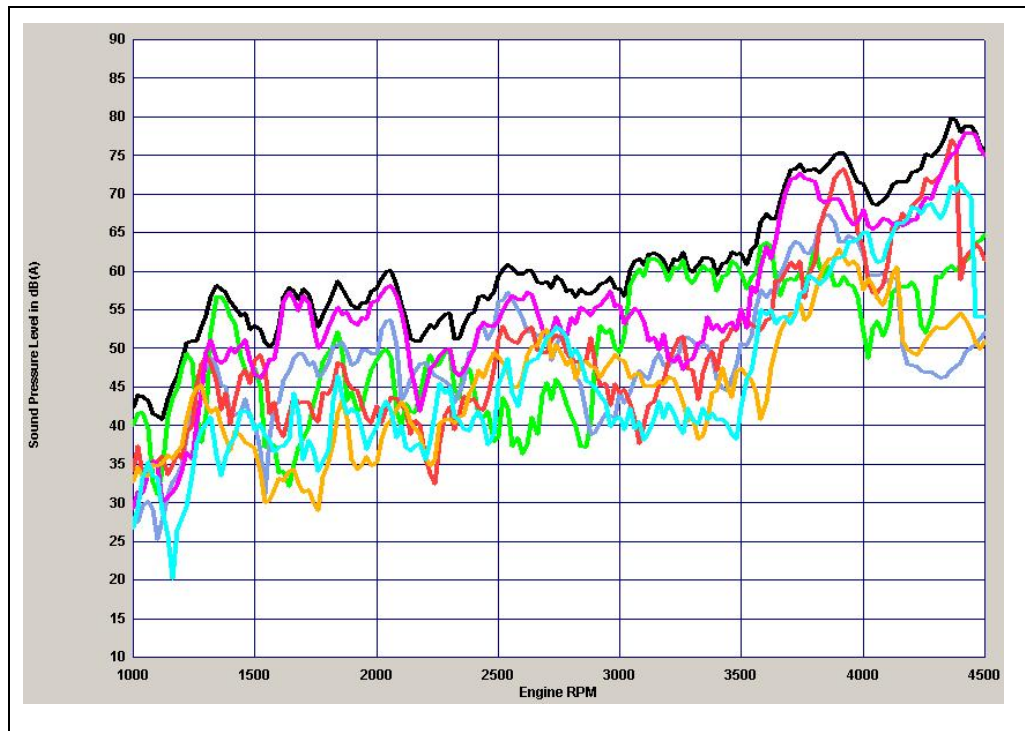


Powerplant Mount Overall 4 th EO Structure-borne Contribution	—
LH Engine Mount 4 th EO Structure-borne Contribution	—
RH Engine Mount 4 th EO Structure-borne Contribution	—
Transmission Mount 4 th EO Structure-borne Contribution	—

Figure 3.13. Powertrain sub-system level 4th EO Contributions

From figure 3.13, it is observed that the Powerplant 4th Engine Order structure-borne contribution is mainly dominated by the RH engine mount and LH engine mount sub-systems, the impact of transmission mount is not considerably higher, therefore it is not worthwhile to work on transmission mount as a sub-system.

Another very important parameter, which will reduce the efforts in providing a solution to the problem, is to find out the direction of the high contribution. This will also be investigated in details, once the main contribution or transfer path is identified.










Suspension attachment overall 4 th EO Structure-borne Contribution	
LH Shock Tower 4 th EO Structure-borne Contribution	
RH Shock Tower 4 th EO Structure-borne Contribution	
LH L-arm Front 4 th EO Structure-borne Contribution	
RH L-arm Front 4 th EO Structure-borne Contribution	
LH L-arm Rear 4 th EO Structure-borne Contribution	
RH L-arm Rear 4 th EO Structure-borne Contribution	

Figure 3.14. Suspension attachments sub-system level 4th EO Contributions

From the suspension system attachments, both LH and RH L-Arm front attachments seem to have high contribution relative to the other points; however, the noteworthy contribution comes from the LH L-Arm attachment.

At this level, the main effect from the suspension and powertrain sub-systems is identified. Next step will be to investigate whether the noise is radiated from all directions, or there are one or more directions, which play a dominant role in this specific noise problem. Analysing the overall 4th engine order structure-borne contribution versus the most dominant sources identified, we will be in a position to determine the percentage of each source, which will enable finding out the priorities to be focused on. The overall level would obviously be lowered by decreasing the

impact of each of the contributors; or stiffening and damping each of the transfer paths. However, this would lead to an over-engineered design, which will not be optimal due to the fact that there will be unnecessarily spent money for the solution of this NVH problem.

One must keep in mind that there will not be unlimited resources in development studies; therefore, in order to avoid over-engineered designs, the most dominant root causes should be attacked. From figure 3.15, the percentage effects of the critical sources are illustrated.

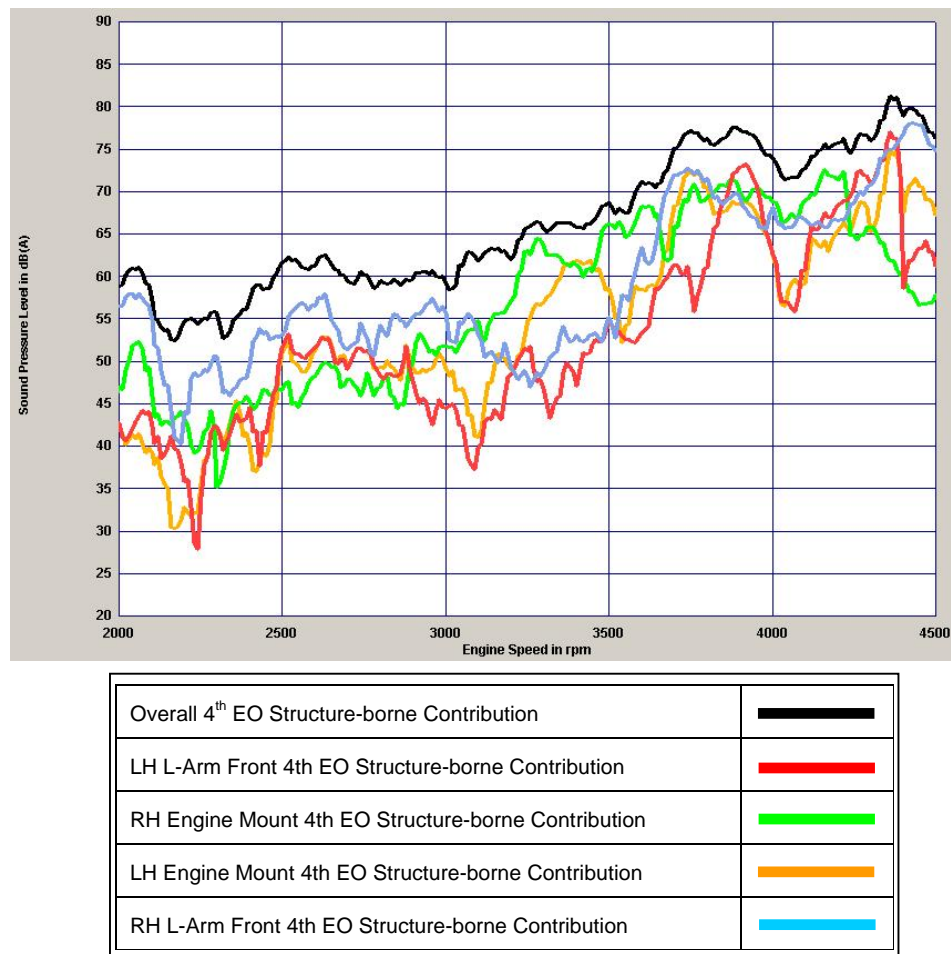


Figure 3.15. The most dominant sources 4th Engine Order Contributions

The maximum sound pressure level at 3900 RPM and the percent contribution of each source are listed in table 3.2.

Table 3.2. Percentage of Each Contributor

CONTRIBUTOR	SOUND [dB(A)]	% CONTRIBUTION
LH L-Arm Front 4 th EO	73	35.48
RH Engine Mount 4 th EO	70	17.78
LH Engine Mount 4 th EO	68.5	12.58
RH L-Arm Front 4 th EO	69.5	15.84
All Other	69.3	18.32

From the contribution analysis, it is observed that the 4th engine order interior noise at 3900 RPM is mainly radiated over the LH L-Arm front attachment point with 35.48 %. Therefore, it makes sense to focus on LH L-Arm front attachment and investigate which direction is more effective in creating the peak at 3900 RPM.

A more detailed approach to the problem is shown in figure 3.16, where overall 4th engine order structure-borne contribution from the LH L-Arm front and the contribution of all directions are displayed. As seen in the red curve, the peak at 3900 RPM is mainly because of the contribution from x and y direction.

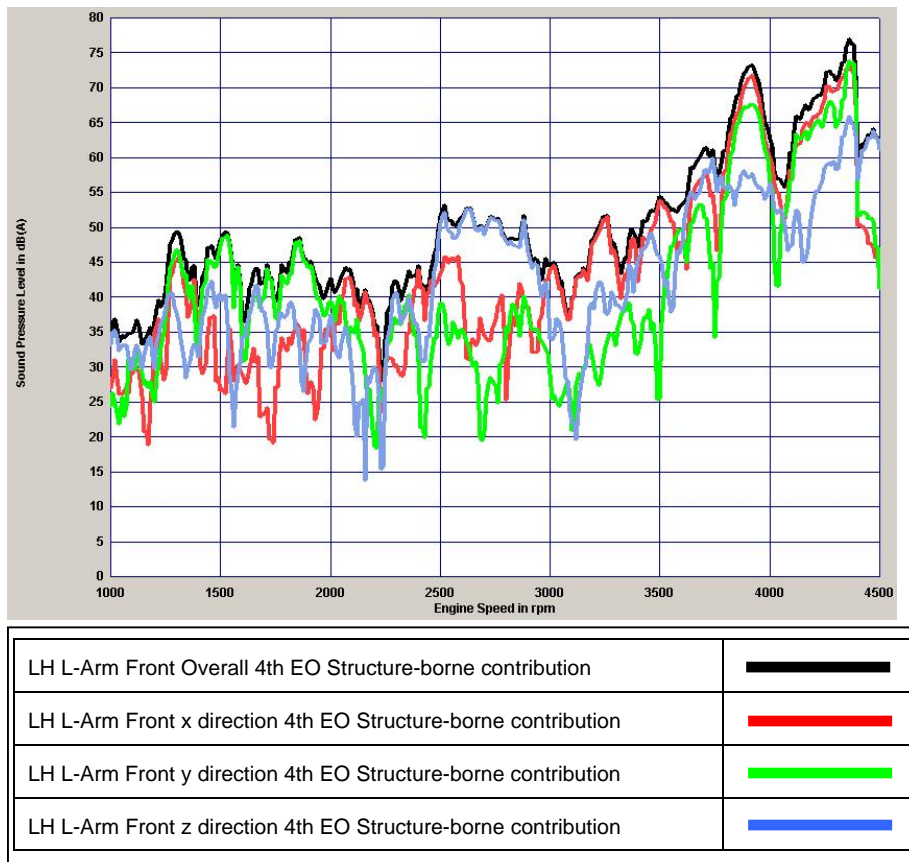


Figure 3.16. Effect of all directions on LH L-Arm front 4th engine order structure-borne contribution.

As the main root cause of the NVH issue is cascaded to the most dominant sub-system and even to the direction of the excitation; from this point forward, the reasons dominating LH L-Arm front interior noise contribution from x and y direction will be investigated. This would either be due to high excitation forces on the attachment point, or high body noise response or both.

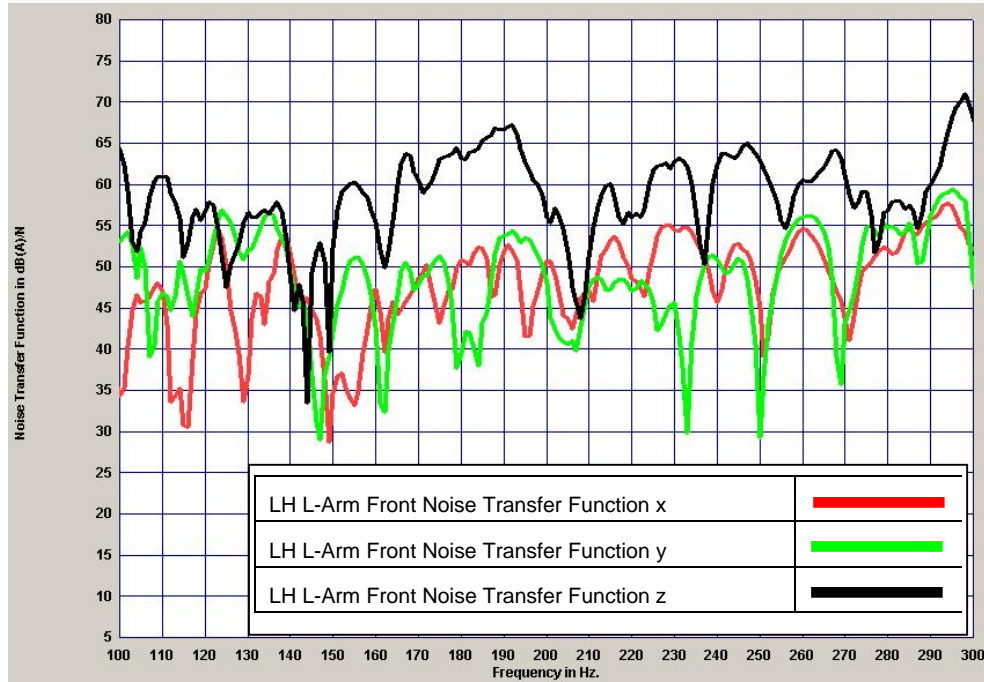


Figure 3.17. LH L-Arm Front Noise Transfer Functions in all Directions

In figure 3.17, the noise transfer functions in all directions from the LH L-Arm front attachment is displayed. Since the NVH problem observed in the vehicle is figured out as high 4th engine order interior noise at 3900 RPM, the 4th engine order frequency at 3900 RPM can be calculated in equation (3.6) by putting the values in the equation (2.32);

$$f_{4^{th}EO} = \frac{3900}{60} 4 = 260 \text{ Hz.} \quad (3.6)$$

This gives the fundamental information that the body sensitivity, or in other words noise transfer function, should be investigated precisely at 260 Hz.

In figure 3.17, it is observed that both x and y directions show a resonant peak at 260 Hz. When NTF response amplitudes are compared, z direction has relatively a small peak; however, its level is nearly 5 dB(A) higher compared to x and y directions. However, it should be kept in mind that, the other important parameter in addition to the noise transfer function is the force levels.

In figure 3.18 the overall forces and the 4th engine order forces are displayed for each direction from LH L-Arm Front attachment point.

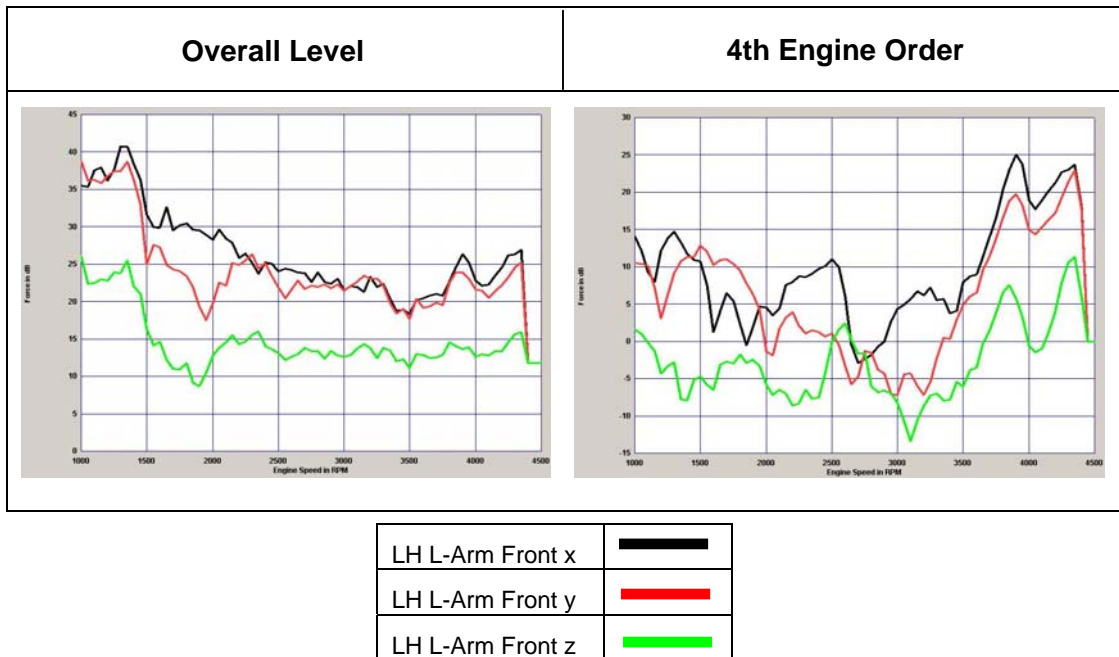


Figure 3.18. Overall and 4th Engine Order Force Levels in x, y, z directions from LH L-Arm Front attachment point.

From the noise transfer function plots, z direction seemed to be more critical at 260 Hz. due to having a higher response level compared to x and y directions. However, when force levels are taken into account, z direction has a much lower force contribution compared to other directions both in overall and 4th engine order levels. Since the structure-borne interior noise contribution is the product of the noise transfer function and the excitation force in the corresponding frequency, the reason of z direction being less dominant is because of the lower force level. However, in x and y directions both noise transfer function and the force levels can be considered as the reasons of higher noise contribution at 3900 RPM, or in other words at 260 Hz.

To be able to overcome this specific noise problem, the interior noise contribution and transfer path analysis played an important role in identification of the main contribution point and the transfer path to the interior. The next steps would be to address several development opportunities for this phenomenon, which can either be to lower the excitation forces in the critical attachment points and the critical directions; or it can be to eliminate the weakness of the body for excitations from the LH L-Arm front attachment point especially in x and y directions.

Choosing the development opportunities is mainly a matter of cost and a trade-off with other attributes' performance. In one hand, we may have the opportunity to improve the NVH performance with certain design changes; whereas on the other hand, it may have a harmful effect on one of the other attribute's performance. As an example, if the dynamic stiffnesses of the chassis bushes are decreased, road NVH performance tends to get better due to the fact that body will be less excited. However, this change may lead a poor vehicle dynamics performance. In vehicle programs, such design changes are always a matter of trade-off, which needs to be decided by the program.

3.8 Potential Solution Methods

In this section, the possible development opportunities will be mentioned with the approximated cost and timing estimations for this particular NVH issue.

As discussed in the previous paragraphs, there are two main solution approaches for this NVH problem. Either the excitation forces acting on the attachment point would be reduced by a new design, or the body noise response at the critical frequency (noise transfer functions) would be attenuated.

3.8.1 Reducing the Excitation Forces

The main excitation source in vehicles is the engine system. During its operation, the engine excites the body with the combustion forces and the forces due to the rotating masses in the engine system (Refer to section Engine Radiated Noise for details). There are certain parameters contributing the forces generated by the engine system. One of the options addresses the very beginning of the force generation as to have less torque and therefore lower force levels from the engine mechanisms. However, this is generally not a very practical solution, because in engine design, the torque requirements are defined based on the operating conditions targeted for the vehicle program. As an example, a medium commercial vehicle, which is designed to carry goods, is expected to show a certain performance, such that it should carry certain weight of goods; reach to a certain maximum speed and has to climb a hill with a certain slope. All these performance metrics can be satisfied by designing the engine in a way that it would produce enough torque and power. As a conclusion, reducing the engine excitation within the

engine system for NVH purposes can be in a limited way in order not to decrease the performance requirements.

Second and the most applicable approach would be to address the engine attachments for force attenuation. Most of the excitation sources in the vehicle are not mounted on the vehicle directly, but generally with rubber/hydraulic or similar isolators. Internal combustion engines are one of the main excitation sources, which are installed in the vehicle via the engine mounts. Depending on the weight, position, balancing requirements, the number of engine mounts would be three or four in most of the commercial vehicles. Engine mounts are built up of a main carrier part and an isolator part, with one end connected to the engine side and the other end to the vehicle body usually by bolts. In engine mount design, one of the most important criteria is the force levels that the engine will apply on the mount, which concerns both NVH and Durability attributes. Second point, which is mostly NVH related and concerns most of the vehicle NVH problems, is to isolate the engine mount active side forces with an appropriate design. Active side refers to the section where the forces from the source are not yet attenuated; and the passive side refers to the region where the mount is attached to the body. It is therefore obvious that the engine mount dynamic characteristics significantly influence the forces transmitted to body. In order not to have unwanted NVH issues such as error state noises or high noise levels, the engine mount active side forces, should be reduced by the mounts in a certain way that the body will not be subjected to too high dynamic forces.

There are certain important physical parameters that have an influence in the force levels transmitted from the active to the passive side of an engine mount. First of all, the engine mount dynamic stiffness should be defined in such a way that the active side forces can be attenuated as demanded before passing to the passive side. Theoretically, the lower the dynamic stiffness, the lower the force levels in the body side; however, it can not be designed to have very low stiffnesses due to durability reasons. Secondly, If the engine mount has its resonances in the frequency range, where the engine has its critical excitation frequencies, the inputs from the engine at this mount resonance frequencies would be amplified and transferred to the body side with very high amplitudes. Especially in commercial vehicles, engine mounts are usually designed with conventional rubber isolators; or in some cases with hydraulic isolators, which are relatively more expensive compared to rubber. Changing the engine mount design requires new tooling, and it would have nearly

70.000 – 100.000 € cost increment. Moreover, the engineering studies and the component testing would approximately cost 100.000 – 150.000 € excluding the vehicle level durability testing, which roughly costs 145.000 € and lasts 7 months. If it is concluded to change the design from rubber mounts to hydraulic mounts, the additional piece cost increment would be 3 € per vehicle.

While reducing the engine mount passive side forces works well by an appropriate mount design; another important method is to select the most suitable locations in the body for engine mount installation, which is relevant to the engine block rigid body movements during its operation. The mount location selection should be made considering the engine torque roll axis, because engine mount active side forces will be much less if the engine is attached from the proper locations, theoretically from the points on its torque roll axis. However, changing the engine mount locations to solve an NVH issue is nearly not a possible method, and it needs to be optimized at the beginning of the vehicle design by integrated CAE and test methods.

Besides the engine mounts, this force reduction methodology can be applied to any systems that have attachments in the excitation source and attachments in the passive side, such as the suspension bushes, shock towers, exhaust hangers and etc. In this study, it is observed that the suspension system attachment points such as the LH and RH L-Arm attachment play an important role in the 3900 RPM 4th engine order structure-borne noise contribution. This means similar treatment would be required as in engine mounts, to be able to decrease the force levels through the suspension system attachments, by changing the dynamic stiffness of the suspension bushes. Approximately, 100.000 € would be required for new tooling and engineering studies for all suspension bushes; 150.000 € for the tooling of shock tower bushes if required. One other point is that those changes may have an impact on durability performance of the components; therefore, vehicle level durability testing would be required, which would roughly costs 145000 € and 7 months time.

As another solution approach, the torsional irregularities of the engine should be attenuated, which is transmitted through the fly-wheel, the half shafts to the suspension sub-systems. From the nature of combustion and rotating masses in the engine system, the torsional vibrations play an important role in NVH contribution in addition to the translational vibrations. Unless the torsional vibrations are attenuated by the fly wheel, they will be transferred by the rigid half shaft connections and suspension attachments to the vehicle body, which would be one of the cases in this

issue. For a detailed root cause identification method, the torsional vibrations on the crank and half shafts should be measured as next steps.

In the reduction of torsional vibrations from the engine system, the most effective solution is introduced by fly wheel manufacturers as DMF, which stands for dual-mass fly wheel. The DMF working principle is to absorb the engine vibrations before they are transmitted to driveline. This is achieved by splitting the conventional flywheel into two sections: a primary section, which bolts to the crankshaft, and a secondary section, onto which the clutch is bolted [11]. Roughly, switching from split-mass flywheel to DMF would lead to 60 – 80 € add-on cost per vehicle excluding the engineering, tooling and testing requirements.

In figures 3.19 and 3.20 the comparison of a vehicle equipped with SMF and DMF are illustrated in 3rd gear wide open throttle acceleration. It is seen in the figures that the usage of dual mass fly wheel provides improvement especially in the mid frequency range (200 Hz. – 500 Hz.) due to better attenuation of the torsional irregularities [9].

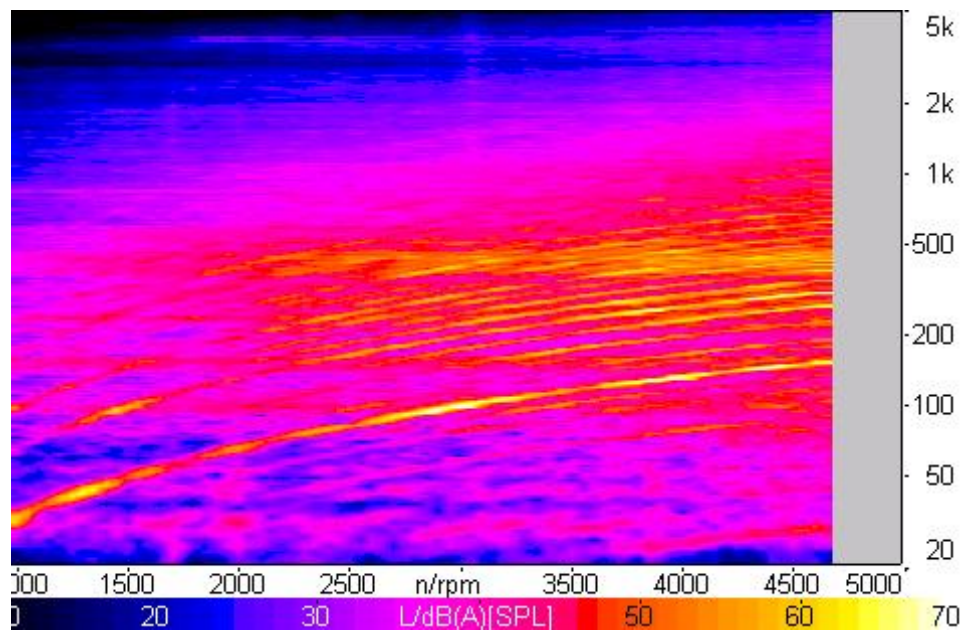


Figure 3.19. 3rd gear WOT acceleration with SMF

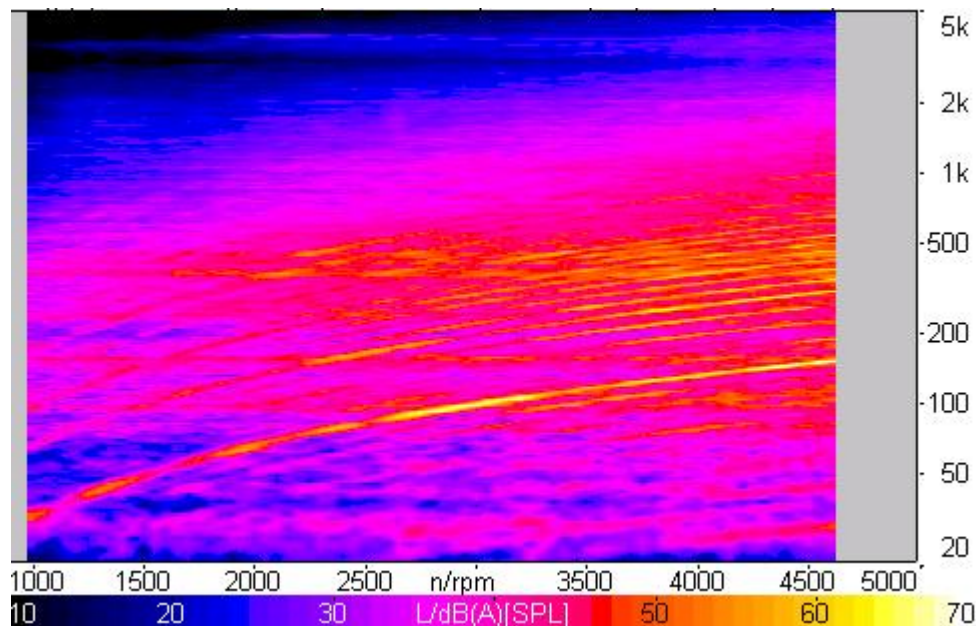


Figure 3.20. 3rd gear WOT acceleration with DMF

If it is not possible for some reasons to modify the engine system related design parameters, one other solution to reduce the excitations in the suspension system would be to improve the half shaft design from NVH perspective. A torsional vibration damper on the half shafts would theoretically work well in reducing the vibration levels such as the DMF; however it is usually very difficult to fulfil durability targets and needs big engineering efforts. Moreover, it would have high add-on cost to implementing a driveline torsional vibration damper.

Since it is not possible to damp the vibrations, other approach would be to shift the resonance frequencies of the half shafts to a range, where the engine excitations are not dominant. This can be satisfied by the use of different shaft designs (such as hollow shafts) rather than conventional solid shafts. This would again put a piece add-on cost of 4€, roughly 70.000 € tooling and 100.000 € engineering and testing cost incremental and the vehicle durability testing.

3.8.2 Decreasing Noise Transfer Function Amplitudes

There may be cases where decreasing the excitation force levels may not be possible due to too high investment and design costs; or modifying the parameters which will attenuate engine excitation forces may affect other attributes such as performance, fuel economy, durability, package and etc. Even combination both of these reasons may force the NVH engineer to re-design relevant parts of the body in order to decrease the noise transfer functions from the critical excitation points. In

this case, it is of great importance to decide on what to do in order to take the noise response of the body to lower values especially in the critical frequency range (260 Hz. in this case). While trying to find solution opportunities to reduce the NTFs, it must be kept in mind that the intention here is not to decrease the response levels in all frequencies, but the specific frequency or frequency range creating the phenomenon needs to be reduced in response amplitude. In this solution approach, the NVH engineer can either stiffen the relevant noise paths, so that the resonance at 260 Hz. would be shifted, or damp the noise transfer function resonance at this frequency.

To be able to have a more detailed understanding on which part of the body contributes more on the relevant noise transfer functions CAE tools can effectively be used. This would enable determining the contribution of the body panels such as body sides, dash panel, doors, underbody and roof on the noise transfer functions from the engine mounts and L-Arm front attachments, since they are observed as the main noise transfer paths.

As short term development opportunities, the surface accelerations of large body panels can be damped by using damping pads or equivalent damping materials. This sort of solutions usually do not require big investment costs and much time; however, the piece cost increment per vehicle would be much higher compared to a long term solution, which requires higher investment costs and timing. Concerning body panel design changes for stiffening or damping of the structure to be able to decrease the NTF amplitudes, such long term and short term opportunities can be addressed opposite of the engine mount or suspension bush design changes. The overall decision should be given by the senior management by considering the overall incremental cost, the pay-back duration, timing required to implement the change and even the money spent for the solution of possible customer complaints due to the NVH problem.

In table 3.3, the approximate cost increments and the required timing to implement the design change is summarized.

Table 3.3. Solution Opportunities of the Observed NVH issue with costs and required timing

Solution	Tooling Cost	Engineering and Component Testing	Vehicle Durability	Piece Cost Increment	Timing (Years)
RH Engine Mount Design	70.000 €	100.000 €	145.000 €	3 €	1 – 1.5
L-Arm Bushes Dynamic Stiffness Change	100.000 €		145.000 €	-	1
DMF Introduction	250.000€	250.000 €	145.000 €	80 €	1.5
Hollow half shafts introduction	70.000 €	100.000 €	145.000 €	4 €	1 – 1.5
Body Stiffeners or Dampers	40.000 €	30.000 €	-	3 – 5 €	0.5

From the interior noise contribution and transfer path analysis conducted in this study, it is observed that the most dominant noise path is the LH L-Arm front attachment, and it is followed by the RH engine mount and other contributor. In table 3.3, the development opportunities are listed both for the main transfer path as the L-Arm and engine mounts; for the main source as the engine system and half shafts, and the last contributor being the body panels. As it can be seen, all mentioned design changes to overcome this NVH problem do not seem to be classified as short-term actions. Moreover, if it is not possible to implement the changes by modifying an existing design, but completely new investments are required; the tooling and engineering costs will be higher in addition to the implementation timing. Therefore, a proper strategy has to be set considering all timing, cost as well as customer satisfaction for image brand DNA, to be able to decide on the most proper solution.

It is of great importance to highlight that the development studies carried on the last contributor as the body panels seem to be the quickest and the cheapest solution. However, it should be kept in mind that the tolerance in the body sheet metals, engine mount and suspension bush stiffnesses, and also engine system itself may put certain limitations for the improvements achieved to overcome the noise problem. This is because of the fact that the after production vehicles would have a certain variation both for air-borne and structure-bore NVH issues, due to the mentioned part/component tolerances and also the assembly processes conducted by different operators.

4. CONCLUSIONS & DISCUSSION

In this study, it is intended to calculate the vehicle level interior noise by the interior noise source contribution and transfer path analysis. Further investigations are carried out to identify the root cause of a certain noise problem. For this purpose, vehicle structure-borne and air-borne noise contributions are identified in each individual sub-system. For structure-borne noise contribution, the forces acting on each of the attachment points and the noise transfer functions from those points in all directions are measured. For air-borne noise contribution, the radiate noise of each source is measured during the operating condition. Additionally, the air-borne noise reduction spectrum is measured for engine, exhaust and intake systems. By summing up the contributions of each of these noise sources, vehicle level interior noise levels are calculated in all RPMs.

Based on the outcome of the studies conducted, it is concluded that; when the structure-borne and air-borne noise sources and transfer paths are identified correctly and accurate measurements are carried out, good correlation between the measured and the calculated values can be achieved. At this point, it is of great importance to select independent excitation points for structure-borne noise contribution analysis. In most of the cases, it is not a big issue to omit the contribution of an attachment; however, it is likely to take into account of two or more excitation points, which may be dependent on each other. That is, the contribution to the interior noise will be summed up twice or more according to the number of dependent points, which will lead to poor correlation. It is pronounced in this study, that each structurally transmitted contribution is the product of a force and sensitivity (or noise transfer function); and each air-borne noise contribution is the difference of the radiated sound and body local noise reduction. Vehicle level overall noise is the sum of all contributions from the relevant sources.

It is finally concluded that, interior noise contribution and transfer path analysis is a very efficient tool in vehicle NVH development, as it can be used to identify the root causes of NVH problems; to set vehicle level NVH targets; and to cascade vehicle level NVH targets to systems, sub-system and even to component levels.

REFERENCES

- [1] **Ewins, D.J.**, 2000. Modal Testing Theory and Practice, Research Studies Press LTD., Hertfordshire.
- [2] **Kinsler, L.E.**, 1982. Fundamentals of Acoustics, Wiley, New York.
- [3] **Fahy, F.**, 2001. Foundation of Engineering Acoustics, Academic Press, San Diego.
- [4] **Möser, M.**, 2004. Engineering Acoustics, Springer, Berlin.
- [5] **Lothar, C.**, 1988. Structure-borne Sound, Springer, New York.
- [6] **Randall, R.B and Tech, B.**, 1987. Frequency Analysis, Brüel & Kjaer, Denmark.
- [7] **Steidel, R.F**, 1989. An Introduction to Mechanical Vibrations, Wiley, New York.
- [8] **Wikipedia Encyclopedia**, <http://en.wikipedia.org>, 14.11.2005.
- [9] **Ford Motor Company**, Internal Documents, 10.10.2005.
- [10] **ISVR**, <http://www.isvr.co.uk/adau/adausource.htm>, 04.03.2006.
- [11] **LuK**, <http://www.lukclutch.com/products/flywheels.phtml>, 16.04.2006.

AUTOBIOGRAPHY

Fuat Okan TANDOĞAN was born in Ankara in 1980. He received his primary degree from Salih Alptekin Primary School (1986-1991); and high school degree from Ankara Yıldırım Beyazıt Anatolian High School (1991-1998). He started his B.Sc. education at the Mechanical Engineering Department of the Istanbul Technical University in 1998. After receiving his B.Sc. degree in 2002; he has started his M.Sc. education at the Automotive Engineering Programme of the Mechanical Engineering Department in Istanbul Technical University. Since 2002, Tandođan has also been working in Ford – Otosan A.S. Product Development Department as an NVH Engineer.