Basics of Vehicle Acoustics

Dr. T. Langhans

11/2021

Agenda:

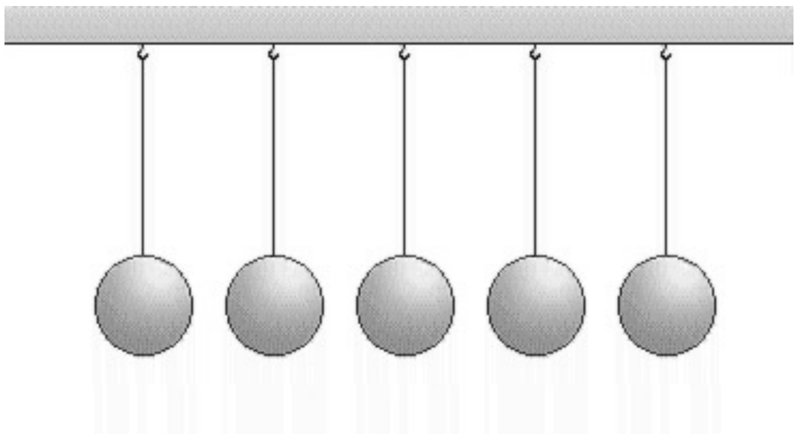
• Fundamentals of Acoustics	30 min
• Noise & Vibration Reduction Methods	30 min
Break	
• NVH in Vehicles	30 min
• Main Insulation Concepts	30 min
Break	
• SEA Prediction	30 min
• Q&A	15min

1.0 Speed of Sound and Light

- **In air, sound propagation is slower than that of light.**
- **One notices this very clearly during thunderstorms ->**

1.1 Propagation of Sound in Air

- **An acoustic wave propagates by small pressure changes that are transmitted locally through the medium, e.g. air.**
- **The propagation speed of these waves is called the speed of sound. For air at room temperature the speed of sound is 340 m/s.**
- **This value must not be confused with the speed of the particles in the medium: only the pressure fluctuations (or acoustic pressure) propagate. In fact, the molecules of the medium keep, on average, the same position when oscillating around their equilibrium position (i.e. small pressure fluctuations). After a sound wave has passed, the molecules return to their initial position.**



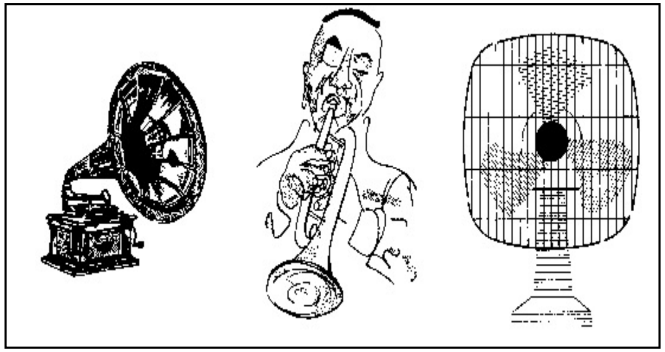
1.2 Emission, Propagation & Reception

- **Emission** is the mechanism by which a sound source causes an oscillatory movement in the ambient medium.
- **Propagation** is the phenomenon by which this movement is transmitted through the medium.
- **Reception** is the phenomenon by which sound is detected. Such a device could be, for example, a microphone or a human ear.

In applied acoustics, one is interested in these 3 phenomena for:

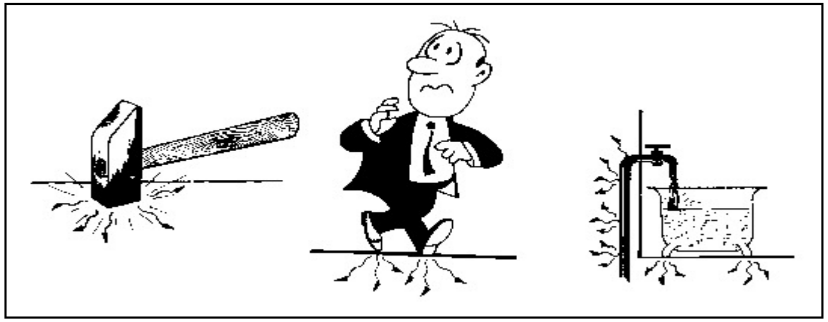
- reducing noise at source
- modifying a propagation path
- measuring noise

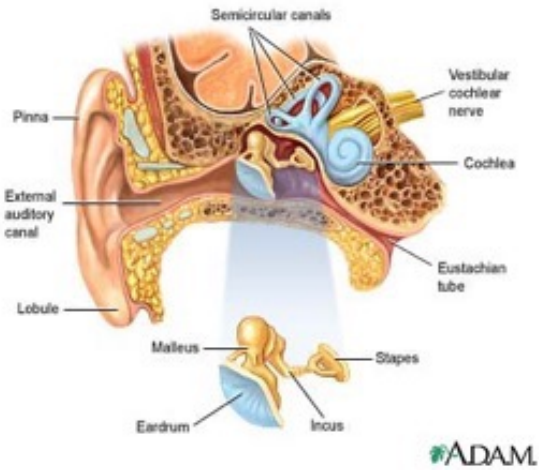
1.3 Airborne versus Structureborne Sound Transmission

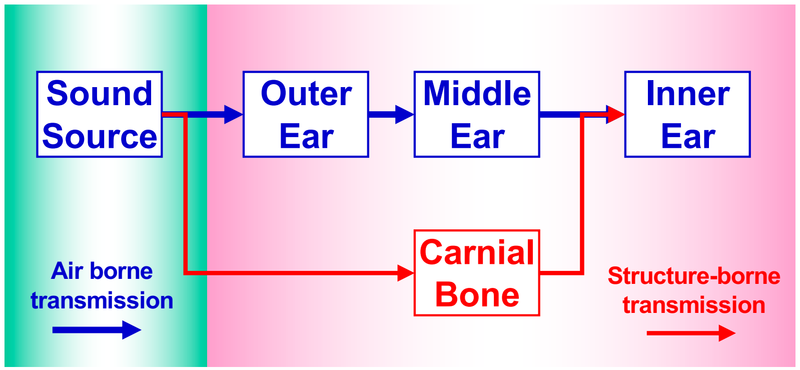


Airborne sound is transmitted via fluids (gases & liquids) and can be perceived with the human ears

Structureborne sound or vibrations are transmitted in solids and can be perceived with the human body







2 Acoustic Quantities

Sound pressure (p): is what the ear detects as noise. Units are Pa or N/m^2

Sound power (P): the amount of noise emitted from a source:

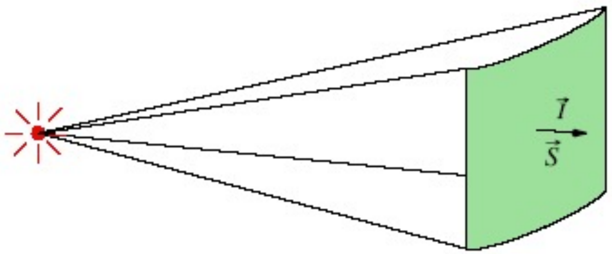
$$P_I = \sum_{i=1}^{N} P_i$$

Sound Intensity (I): the sound intensity of a sound wave describes the direction and net flow of acoustic energy through an area.

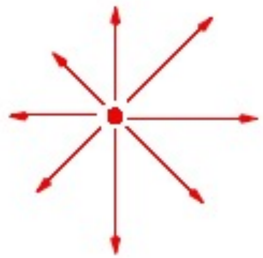
$$Total\ power\ P_I = \int \vec{I}.d\vec{S}$$

- Intensity is also the time-averaged rate of energy flow per unit area. If the energy is flowing back and forth resulting in zero net energy flow then there will be zero intensity:

$$\vec{I} = \frac{1}{T}\int_0^T \vec{I}(t)dt$$

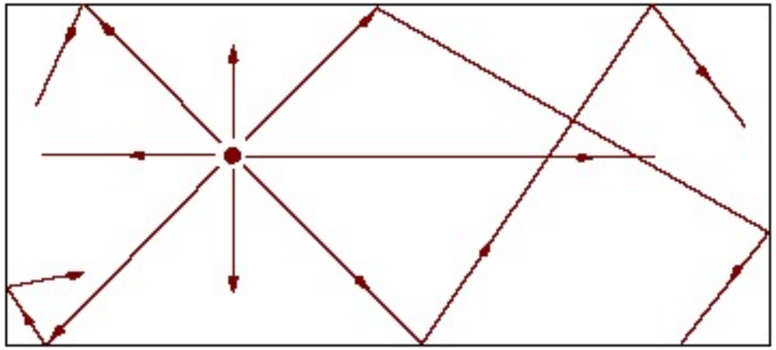


Free field: refers to an idealized situation where the sound flows directly out from the source and both pressure & intensity drop with increasing distance according to the inverse square law.



2.1 Acoustic Quantities

Diffuse field: in a diffuse field the sound is reflected many times such that the net intensity can be zero. E.g. reverberant room.



Particle velocity: pressure variations give rise to Movements of the air particles.It is the results of Pressure and particle velocity that results in the Intensity:

$$\vec{I} = p\,\vec{v}$$

where p= sound pressure (Pa)
$\vec{v}$= particle velocity (m/s)

Acoustic impedance:

Is defined as the product of the mass density of a medium and the velocity of sound in that medium:

$$Z = \varrho \, . \, c$$

where
ϱ = mass density (kg/m^3)
c = velocity of sound in the medium (m/s)

2.2 Acoustic Quantities

Reference conditions:

the density of air ϱ_o = 1.21 (kg/m^3)
the velocity of sound in air c = 343 (m/s)
the acoustic impedance $\varrho_0.c$ = 415 rayls (kg/m^2s)

Sound power level:

$$L_W = 10\log_{10}\left(\frac{|P_I|}{P_0}\right)$$

The reference sound power *is* $P_0 = 10^{-12}$ (W)

Particle velocity level:

$$L_v = 20\log_{10}\left(\frac{v}{v_0}\right)$$

The reference particle velocity is $v_0 = 50\ 10^{-9}$ (m/s)

Sound pressure

p_0 = 20uPa:

$$L_p = 10\log_{10}\left(\frac{p}{p_0}\right)^2$$

$$= 20\log_{10}\left(\frac{p}{p_0}\right)$$

Sound intensity

I_0=1pW:

$$L_I = 10\log_{10}\left(\frac{|\vec{I}|}{I_0}\right)$$

2.3 Sound Pressure Level & Decibel

The **decibel (dB)** is a logarithmic unit which is used in science to compare the quantity of interest with a reference value, often the smallest likely value of the quantity. Sometimes it can be an approximate average value.

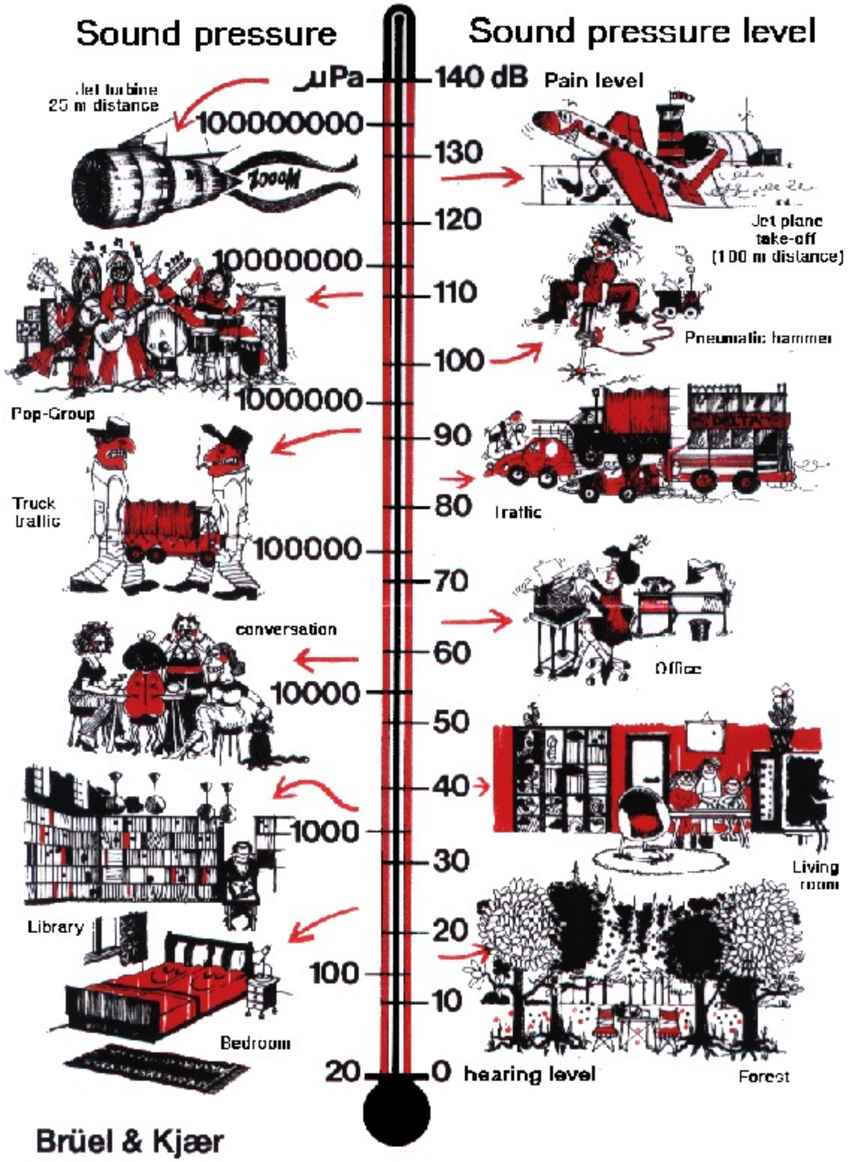
In acoustics the decibel is most often used to compare sound pressure with a reference pressure. The reference sound pressure (in air) = 0.00002 = 2E-5 Pa (rms)

Acousticians use the dB scale for the following reasons:

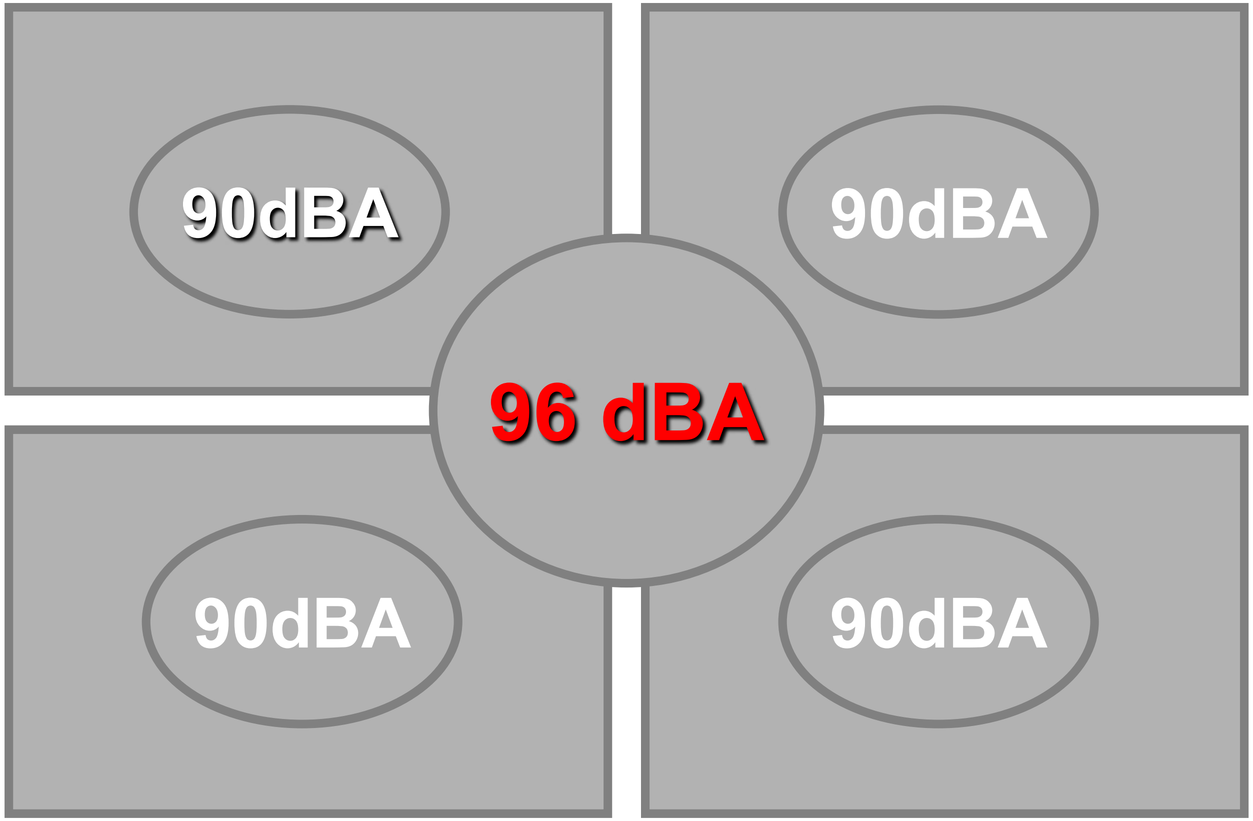
- Quantities of interest (e.g. sound pressure) often exhibit such huge ranges of variation that a dB scale is more convenient than a linear scale.
- The human ear interprets loudness more easily interpreted with a logarithmic scale than with a linear scale.

The **sound pressure level (SPL)** indicates the sound pressure, p, as a level referenced to 0.00002 Pa, calibrated on a decibel scale

$$SPL = 20x\lg\left(\frac{p}{0.00002}\right)(dB)$$

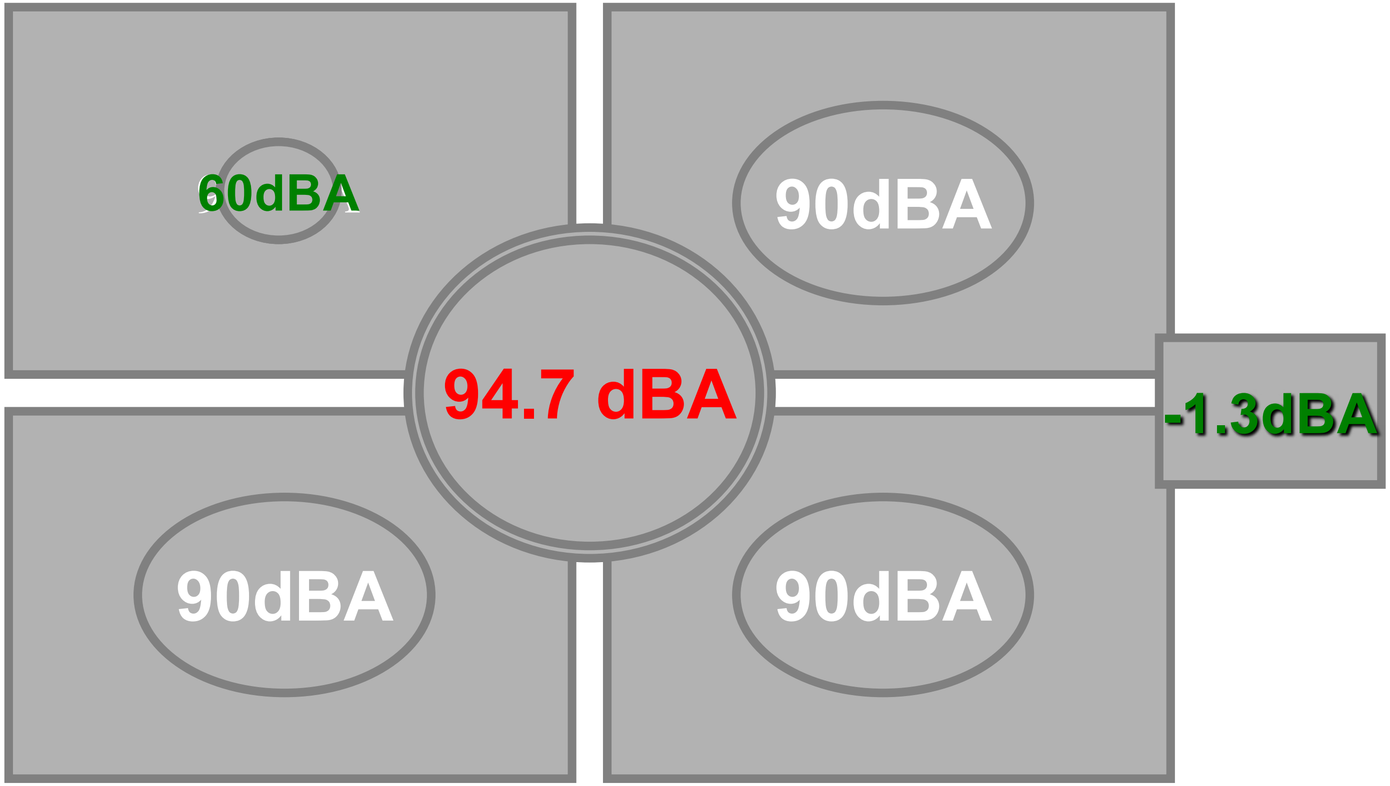


3. Calculation of Overall Level



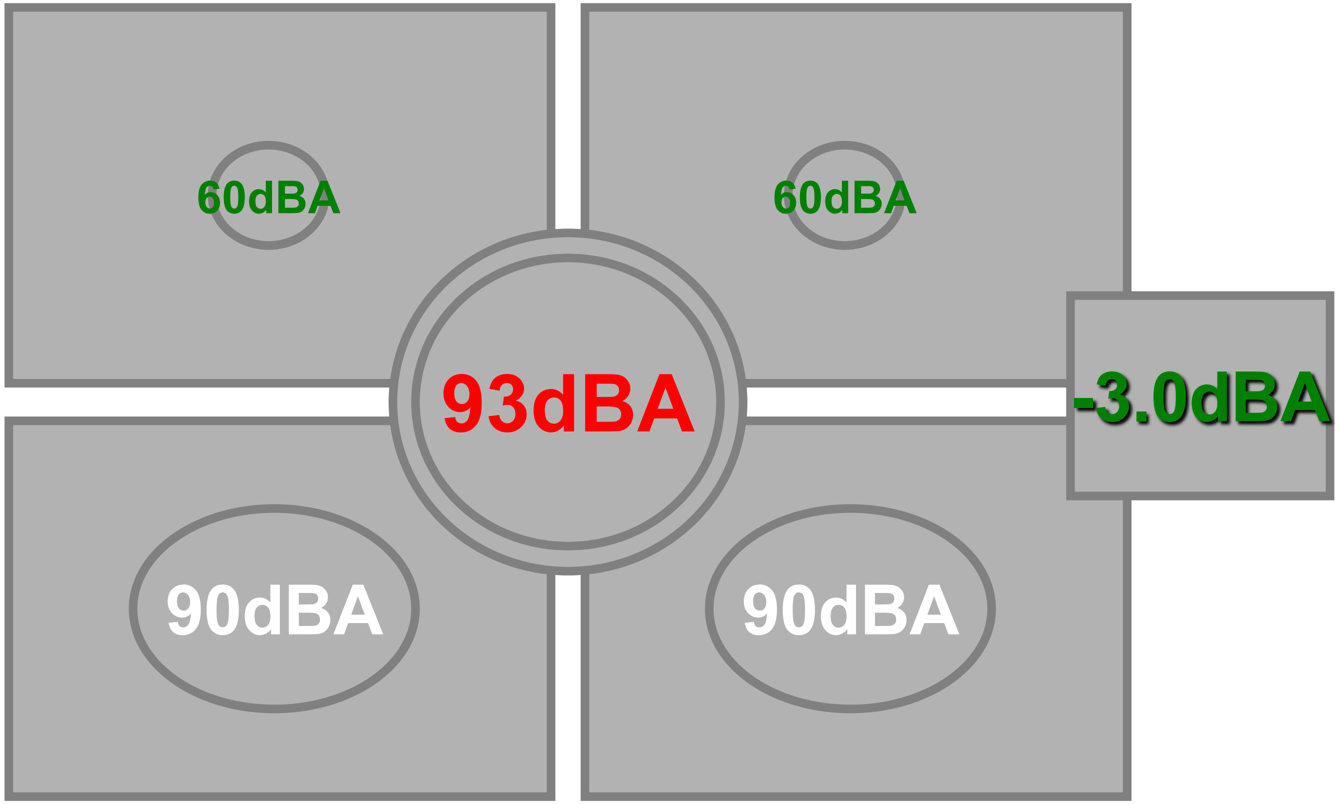
4 sources (or windows) with 90dB sound intensity each will sum up to 96dB overall !

3.1 Calculation of Overall Level



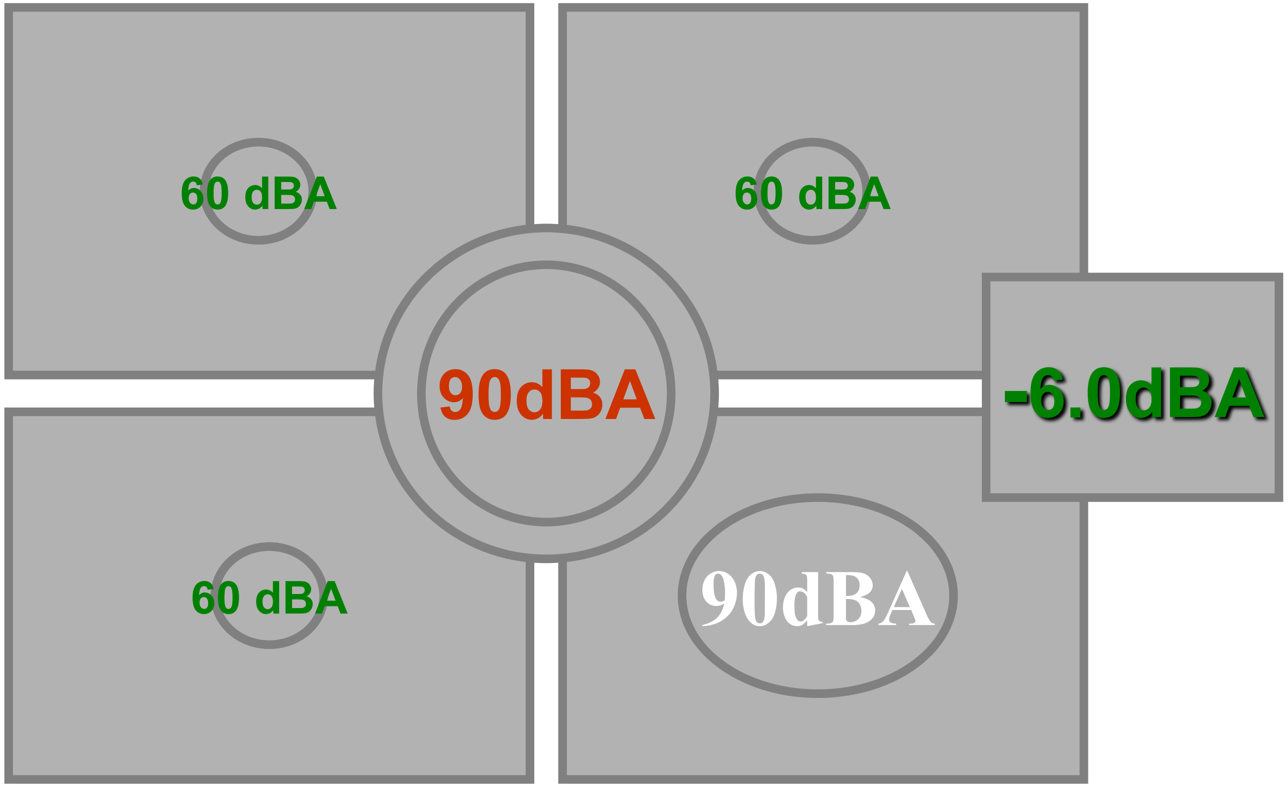
3 sources (or windows) with 90dB sound intensity and one source with 60dB will sum up to 94.7dB overall !

3.2 Calculation of Overall Level



2 sources (or windows) with 90dB sound intensity and two sources with 60dB will sum up to 93dB overall !

3.3 Calculation of Overall Level



1 source (or windows) with 90dB sound intensity and three sources with 60dB will sum up to 90dB overall !

4. Frequency

- Any motion that is repeated is called a periodic motion. Examples of periodic motions are the earth's rotation, a beating heart, the wings of a hummingbird and the vibration of a music instrument's string.
- A period is the time required for one complete cycle. **The frequency** is the number of cycles occurring in a given time period.
- Thus the frequency is the inverse of the period:

$$f = \frac{1}{T}$$

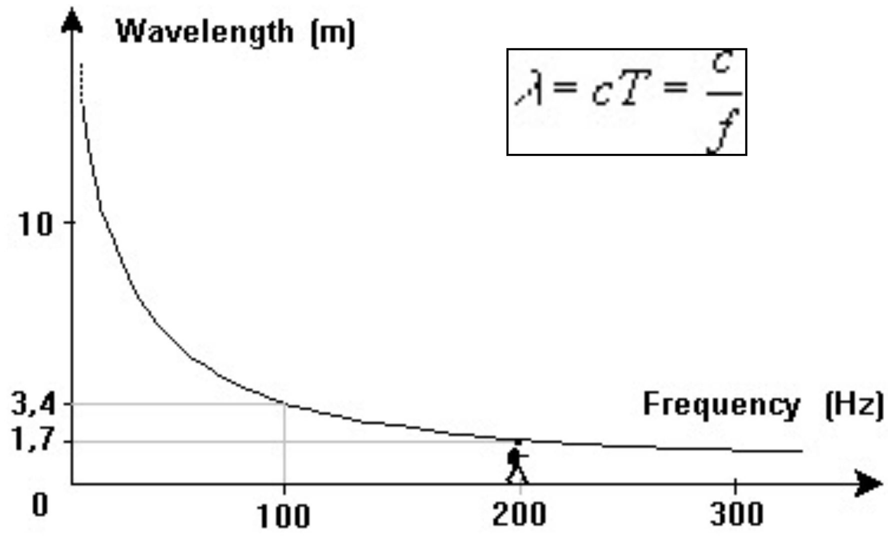
The units used in the past (cycles per seconds = cps) are nowadays more commonly expressed in **Hertz** (Hz) => 1 cps = 1 Hz

Motion	**Period (sec)**	**Frequency (Hz)**
Earth's rotation	24x60x60 = 86400	0,0000115
Heart beat	1	1
Hummingbird's wings	0.016	62.5
Concert A	0.0022727	440

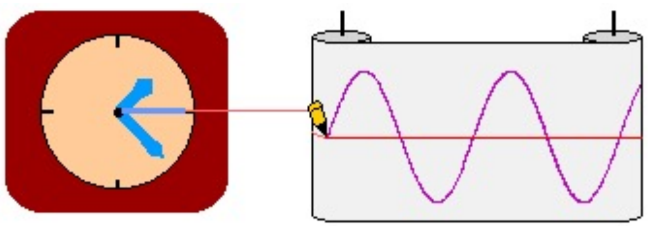
Table 1

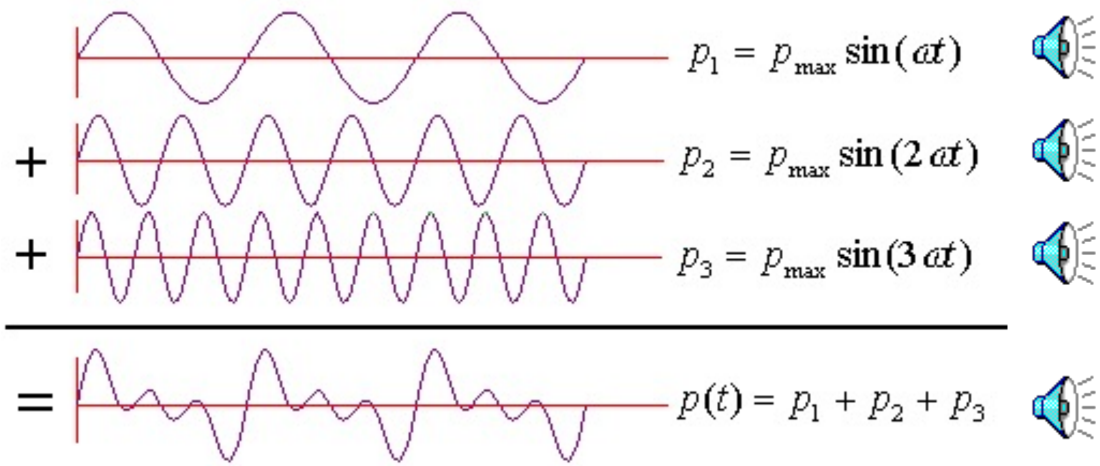
4.1 Wavelength, Pure + Harmonic Sounds

It can be shown that a sound that has a time periodicity, is also periodic in space. The time period T corresponds to the wavelength in the direction of propagation. The **length of the wave** (or **wave-length**) is one of the most important parameters in Applied Acoustics and must be taken into account for many applications. Our character shown opposite is 1.7m high and this corresponds to the wavelength at the frequency of f = 200 Hz - using a speed of sound of c=340 m / s.



Pure Tone





Harmonic Sounds

4.2 Random Sounds

Real sound waveforms are not as simple as that of pure tones and harmonic sounds. Very often, no repetitive pattern can be identified in a signal. One cannot predict the signal behavior from its time history: It has a random waveform. Such random sounds are called "noises" because they are often more disturbing than periodic or almost periodic sounds. The audio recordings below allow you to experience the complexity of such time histories.

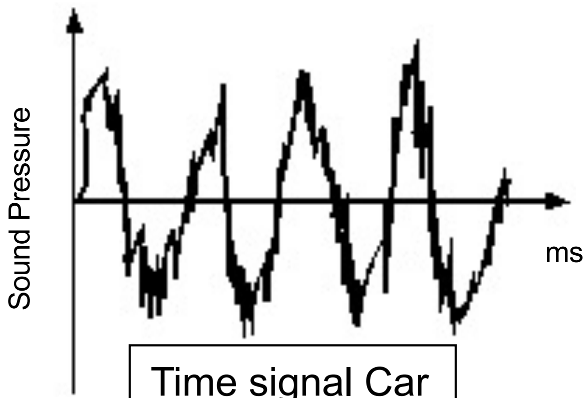
Car

Wind

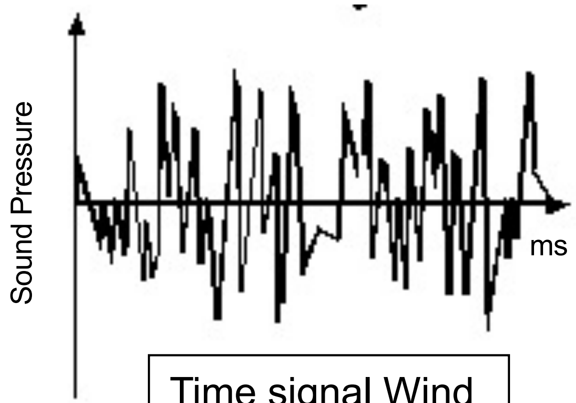
Train

Drill

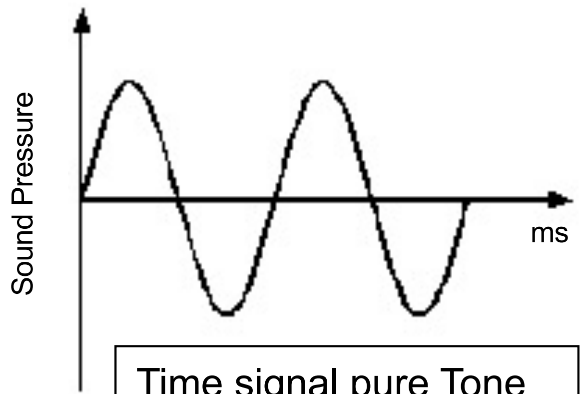
Pure Tone



Time signal Car



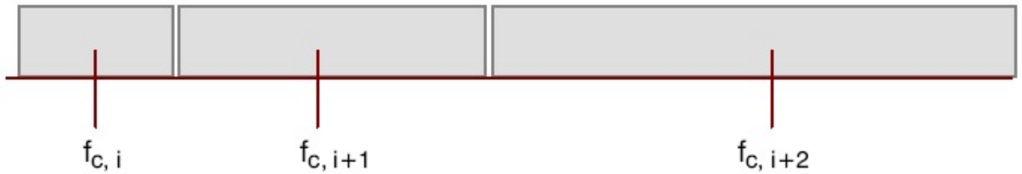
Time signal Wind



Time signal pure Tone

5. Octave Bands

Complete (1/1) octave bands represent frequency bands where the center frequency of one band is approximately twice that of the previous one:

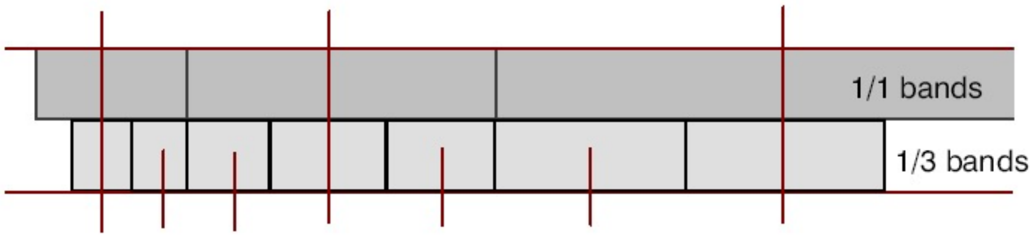


$$f_{c,\,i+1} = 2.\,f_{c,i}$$

Partial octave bands (1/3, 1/12 1/24 . . .) represent frequency bands where

$$f_{c,\,i+1} = (\,2^{1/x}\,).\,f_{c,i}$$

and where x = 3,12, 24 . . .

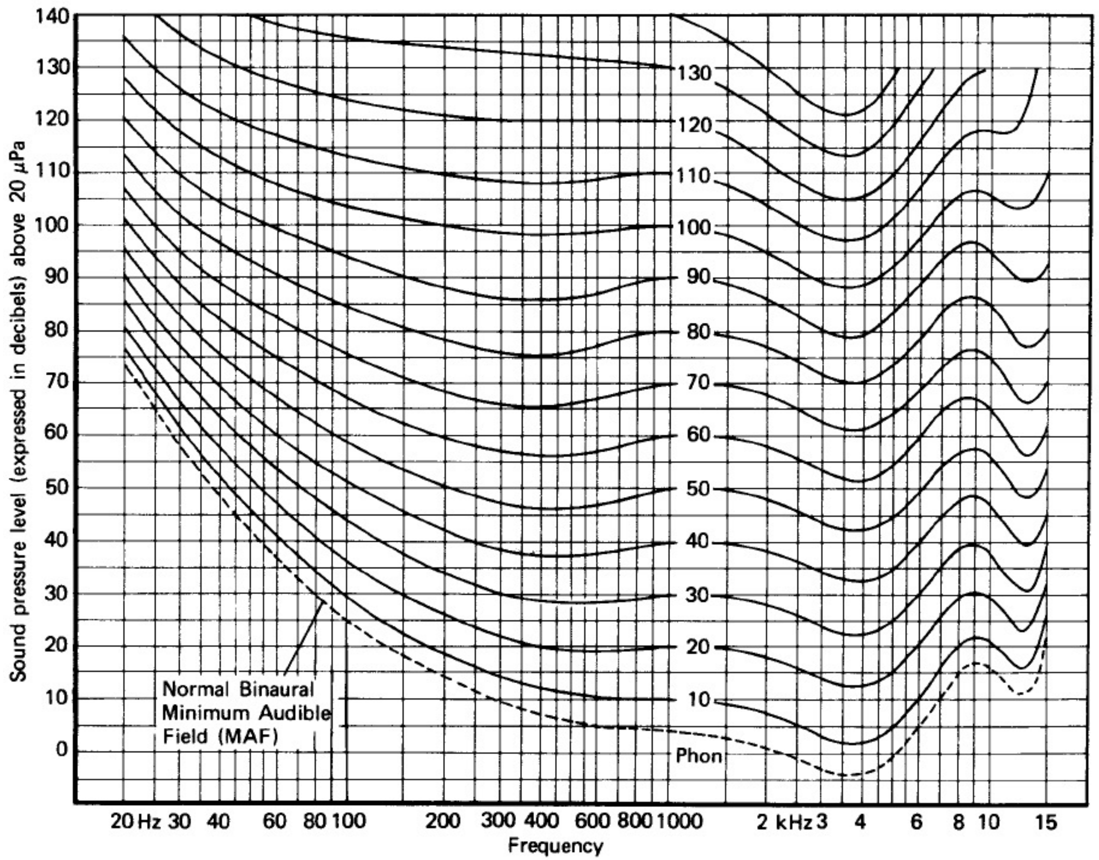


The **Lower** band limit of a 1/x octave band is $f_c.2^{-1/2x}$

The **Upper** band limit of a 1/x octave band is $f_c.2^{+1/2x}$

5.1 Frequency Weighting

The human ear has nonlinear, frequency dependent characteristics, which means that the sensation of loudness cannot be perfectly described by the sound pressure level. To derive an experienced loudness level from the SPL, the spectrum is multiplied by a weighting function. A number of equal loudness contours are shown below:

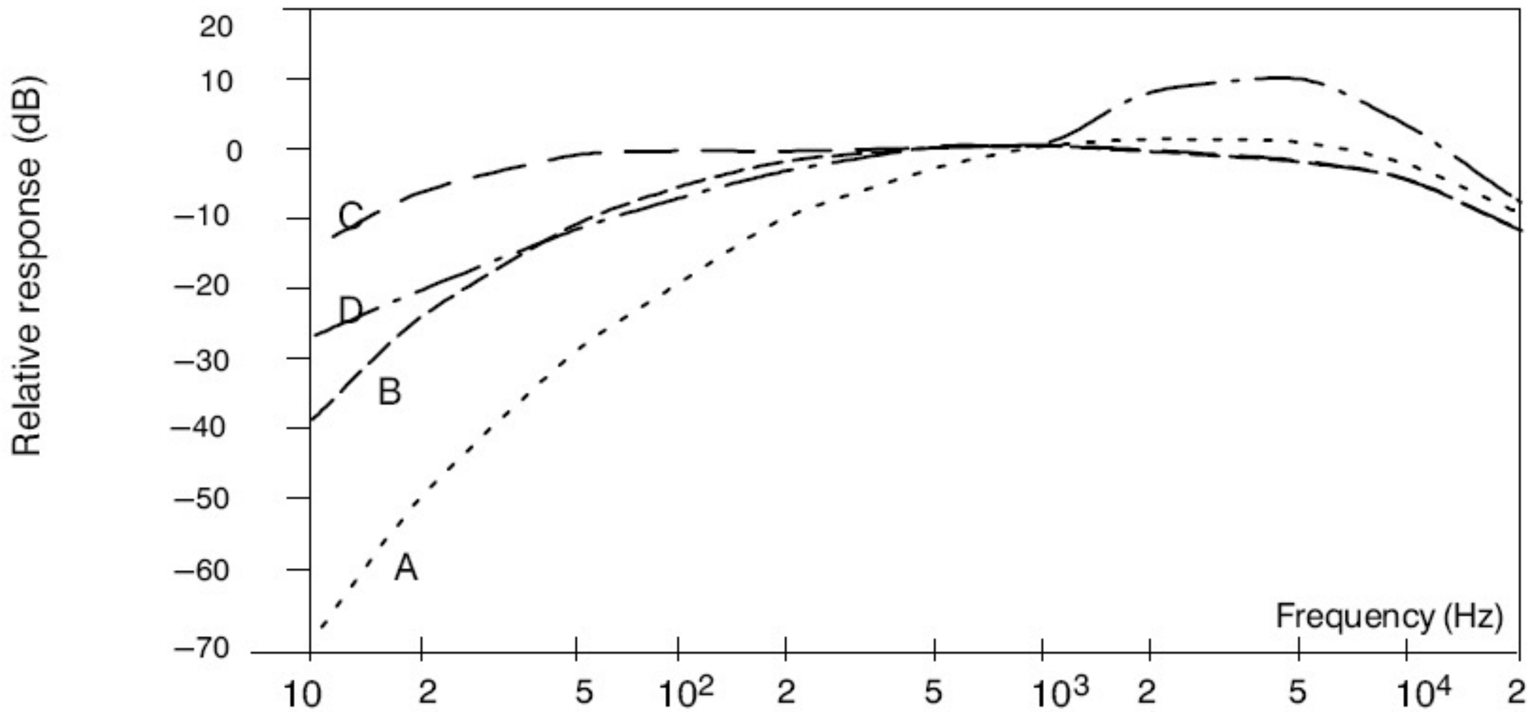


5.2 Frequency Weighting: A,B & C

A-weighting modifies the frequency response such that it follows approximately the equal loudness curve of 40 phons.

The **A-weighted** sound level has been shown to correlate well with subjective responses.

The **B and C-weighting** follow more or less the 70 and 100 phon contours:



6. Basic Concepts of Sound Quality

Sound signals: The characteristics of a sound as it is *perceived* are not the same as the characteristics of sound being emitted !

Sound power & sound pressure: The effect of sound power emanating from a source is the level of sound pressure. Sound pressure is what the eardrum detects.

Sound pressure level: The sound pressure level of 20uPa is known as the standardized normal hearing threshold and represents the quietest sound at 1kHz that can be heard by the average person. Due to the large dynamic range the values are normally expressed in Dezibels (dB).

Hearing frequency range: The threshold frequency for human hearing is appr. 20kHz (depends on age).

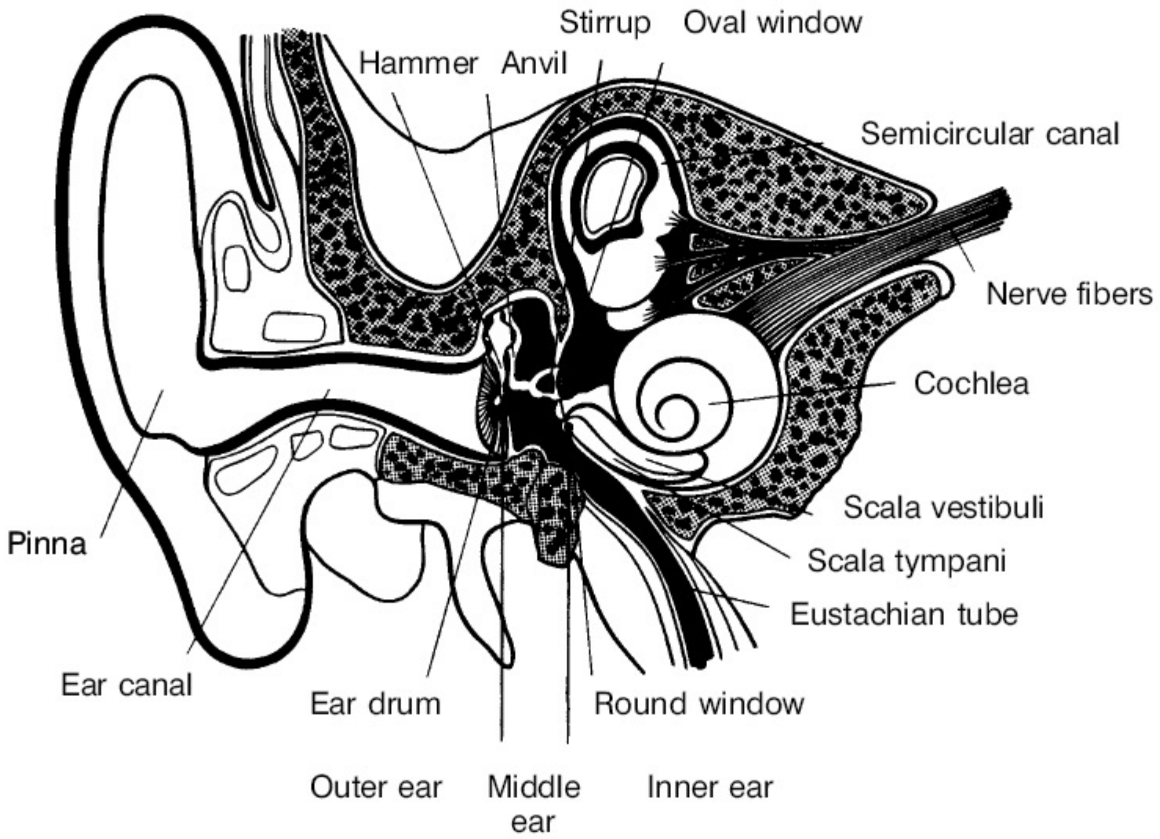
Loudness & pitch: A sound can be charcterized by ist loudness and ist frequency content. The common term for describing the frequency content of a sound is „pitch“. However, pitch is very much a *perceived frequency sensation* & depends on ist frequency and sound pressure level.

An important element in explaining why two sounds with an equal dB level may have a totally different subjective quality is related to the physics of the human hearing process. The human ear is a complex, nonlinear device, with specific frequency dependent transmission characteristics.

In additon, the fact, that hearing usually involves two ears („binaural“) has a considerable influence on sound perception.

7. The Hearing Process

Before reaching the eardrum, an incident acoustic signal is considerably modified by the spectral & spatial filtering characteristics of the human body and the ear.



7.1 Binaural Hearing

The sound signals received by the left and right ear show a relative time delay as well as a spectral difference dependent on the direction of the sound.

Below about 1500Hz, the phase difference between the two signals will be the main contribution to localization, while above this frequency the interaural level difference and difference in spectrum will be the principal factors.

Processing in the human brain not only allows the sound to be spatially localized, but also to suppress unwanted sounds and to concentrate on a sound coming from a specific direction.

This is known as **„cocktail party effect“.**

Sound perception:

The effects of the inner ear are many, but the most important are its nonlinear characteristics.

This means that the auditory impression of sound strength, which is referred to by the term **„loudness“** is not linearly related to the sound pressure level.

In addition, the perceived loudness of a pure tone of constant SPL varies with ist frequency.

7.2 Pitch

• The perceived **„frequency sensation“,** referred to as pitch, is not directly related to the frequency itself.

• The pitch of a pure tone varies with both the frequency and the sound pressure level, and this relationship is itself dependent on the frequency of the tone. Pure tones can be used though to determine how pitch is perceived. One possibility is to measure the sensation of „half pitch“. In this case the subject is asked to listen to one pure tone, and then adjust the frequency of a second one such that it produces half the pitch of the first one.

• At low frequencies, the halving of the pitch sensation corresponds to a **ratio of 2:1** in frequency.

• At high frequencies however this does not occur and the corresponding frequency ratio is larger than 2:1.

For example a pure tone of 8kHz produces a „half pitch“ of only 1300Hz.

7.3 Critical Bands

- The inner ear can be considered to act as a set of overlapping constant percentage Bandpass filters. The noise Bandwidths concerned are approximately constant with a Bandwidth of 110Hz below 500Hz, evolving to a constant value (about 23%) at higher frequencies.

- This corresponds perfectly with the nonlinear frequency-distance characteristic of the cochlea.

- These Bandwidths are often referred to as **„critical bandwidths“** and a „bark“ scale is associated with them as shown in **Table 6.1**:

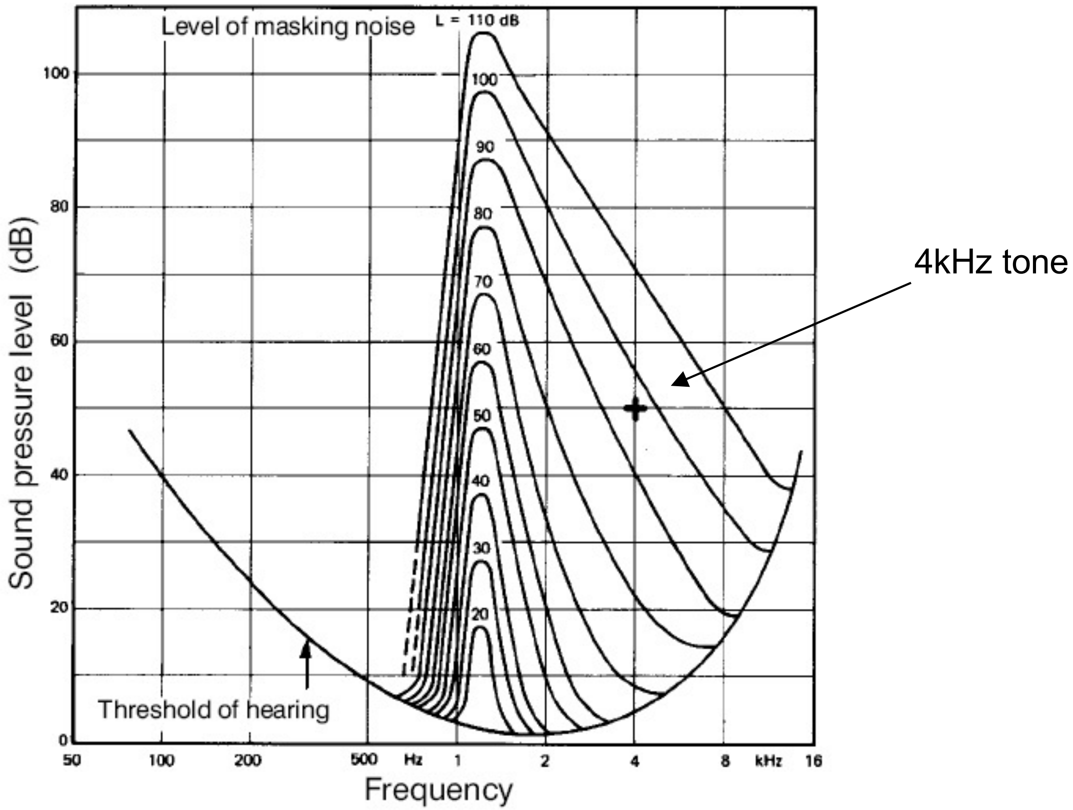
Critical Band (Bark)	1	2	3	4	5	6	7	8
Center Frequency (Hz)	50	150	250	350	450	570	700	840
Bandwidth (Hz)	100	100	100	100	110	120	140	150
Critical Band (Bark)	9	10	11	12	13	14	15	16
Center Frequency (Hz)	1000	1170	1370	1600	1850	2150	2500	290
Bandwidth (Hz)	160	190	210	240	280	320	380	450
Critical Band (Bark)	17	18	19	20	21	22	23	24
Center Frequency (Hz)	3400	4000	4800	5800	7000	8500	10500	13500
Bandwidth (Hz)	550	700	900	1100	1300	1800	2500	3500

Table 6.1 Table of critical bands

7.4 Masking

The critical bands have important implications for sounds composed of multiple components. Narrow band random sounds falling within one such filter bandwidth will add up to the global sensation of loudness at the center frequency of the filter. On the other hand, a high level sound component may „mask“ another lower level sound which is too close in frequency.

A 50dB, 4kHz tone (marked +) can be heard in the presence of narrow-band noise, centered around 1200 Hz, up to a level of 90 dB. If the noise level rises to 100 dB, the tone is not heard.



8. Sound Quality

The majority of problems or studies related to acoustics, the issue at hand is acoustic comfort.

The **acoustic pressure level** is by no means sufficient or even adequate to correctly represent the actual hearing sensations.

Auditory impressions can be annoying, in which case the sound is unwanted and is often referred to as „noise“. Typical examples are irritating engine, road or wind noise in a car, aircraft noise, machine or fan noise in the working environment.

Examples of vehicle noises which while being annoying do not contribute significantly to the SPL are:

Wiper noise, fuel pump noise, alternator whine, dashboard squeaks

To express this negative quality, a multitude of concepts are used:

Whine, rattle,boom,rumble,hiss,beat,squeak, speech interference, harshness, sharpness, roughness, fluctuation strength ..

But not everything you hear is either bad or unwanted. Examples are the solidity of a door slam, the feeling of sportiveness of a car engine (or exhaust) during acceleration, the smoothness of a limousine engine, the catching of a door lock or seat belt ... It only has to sound „right“ then !

8.1 .Typical Car Interior Noises

Noise description	Typical cause	Frequency Range in Hz
Slapping	Liquid in tank	< 20
Ticking	Speedometer cable	< 20
Bumps & thumps	Suspension, body	< 60
Humming	Tyres	< 60
Booming	Suspension	20 .. 60
Droning	Drive, exhaust system	< 250
Whine	Drive shafts	100 .. 500
Hammer	Diesel engine	> 300
Howl	Gearbox, drive train, servo pump	> 300
Roar	Exhaust system	< 500
Ringing	Petrol engine	> 500
Humming	Tyres, ball bearing	> 1k
Whistles	Catalytic converter	> 1k
Creaks	Cables, silencer system	0,3 .. 2k
Rattles	Drive, body	0,3 .. 2k
Squeaks	Brakes	1 .. 6k
Rustling	Body, dashboard	1 .. 10k
Banging	Suspension, body	1 .. 10k
Rushing, hissing	Wind noise, inlet	1 .. 10k

Acoustic treatment

Agenda:

- Fundamentals of Acoustics 30 min
- **Noise & Vibration Reduction Methods 30 min**

Break

- NVH in Vehicles 30 min
- Main Insulation Concepts 30 min

Break

- Pexmet & SEA Prediction 30 min
- Q&A 15min

9. Noise & Vibration Reduction Methods

Noise & Vibration in a dynamic system can be reduced by a number of means. These can be broadly classified into **active, semi-active and passive methods**.

Active Control involves the use of certain active elements such as speakers, actuators, and microprocessors to produce an "out-of-phase" signal to electronically cancel the disturbance. All other methods that do not include a real-time active algorithm can be grouped under **passive control options.**

The traditional passive control methods for airborne noise include the use of absorbers, barriers, muffler, silencers, etc. For reducing structure-borne vibration or noise, several methods are available. Sometimes just changing the system's stiffness or mass to alter the resonance frequency can reduce the unwanted vibration as long as the exciting frequencies do not change. But in most cases, the vibration need to be isolated or dissipated by using isolator or damping materials.

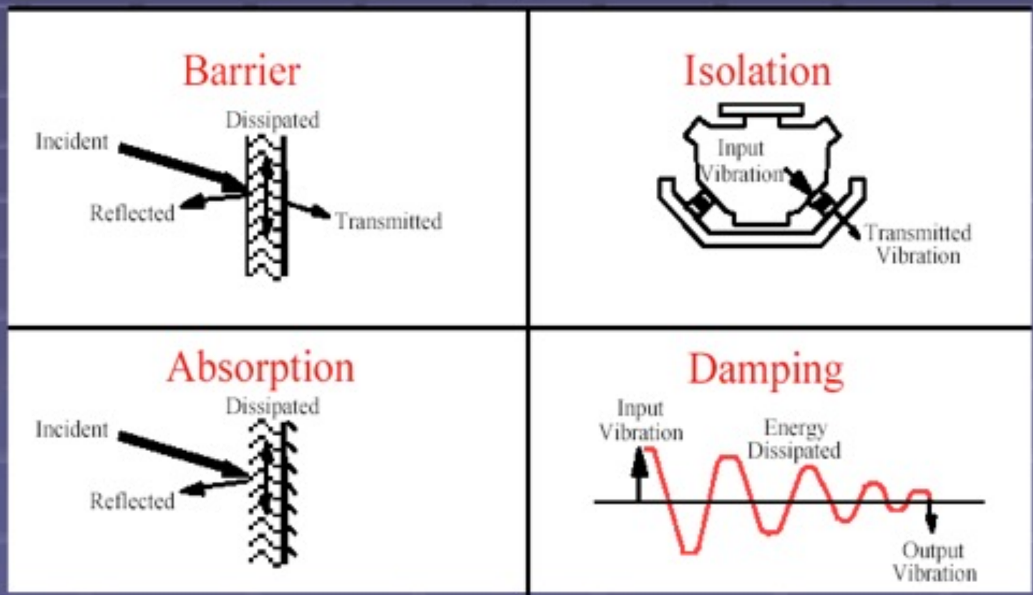
In semi-active methods, active control is used to enhance e.g. the damping properties of passive elements. Examples include Electro-Rheological and Magneto-Rheological fluids, and Active Constrained layer damping in which the traditional constrained layer is replaced with a smart material.

9.1 The four Noise & Vibration Reduction Methods

An acoustical material is any material that reduces noise and vibration

4 different materials are commonly used to control automotive noise and vibration

Noise | Vibration



Reflection:

Energy will be reflected at the point of impedance mismatch

Dissipation:

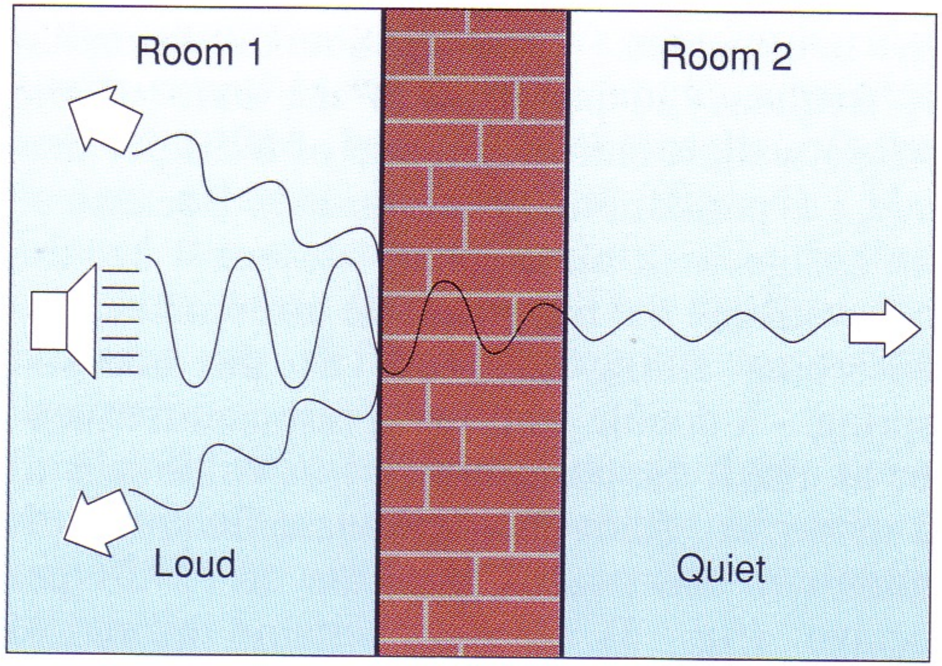
Energy will be transformed from internal friction into heat

Each material performs a special function to reduce noise or vibration

9.2 Airborne Noise: Transmission loss

- Sound transmission loss (STL) describes the ability of a component to reduce the transmission of sound from the source room to the receiving room.

- In case of cars, the source room is normally the **engine compartment**.



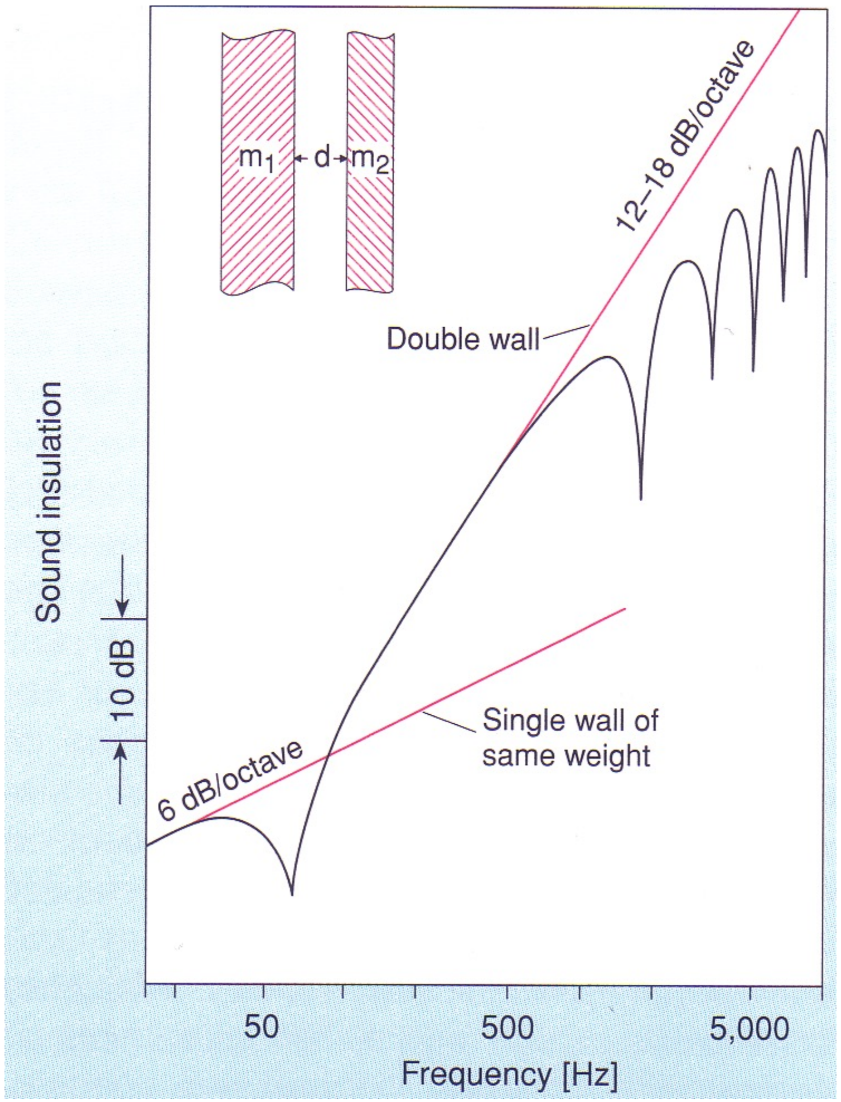
9.3 Airborne Noise: Double Walls

Acoustic engineers differentiate between single walls & double walls.

In both cases, airborne sound insulation is frequency dependent.

The STL for a **single wall** follows the mass law, e.g. 6dB/octave per doubling of the frequency and/or mass.

The STL of a **double wall** has a resonant frequency at which the transmission loss falls to a minimum, above this frequency the slope is between 12dB and 18dB/octave.

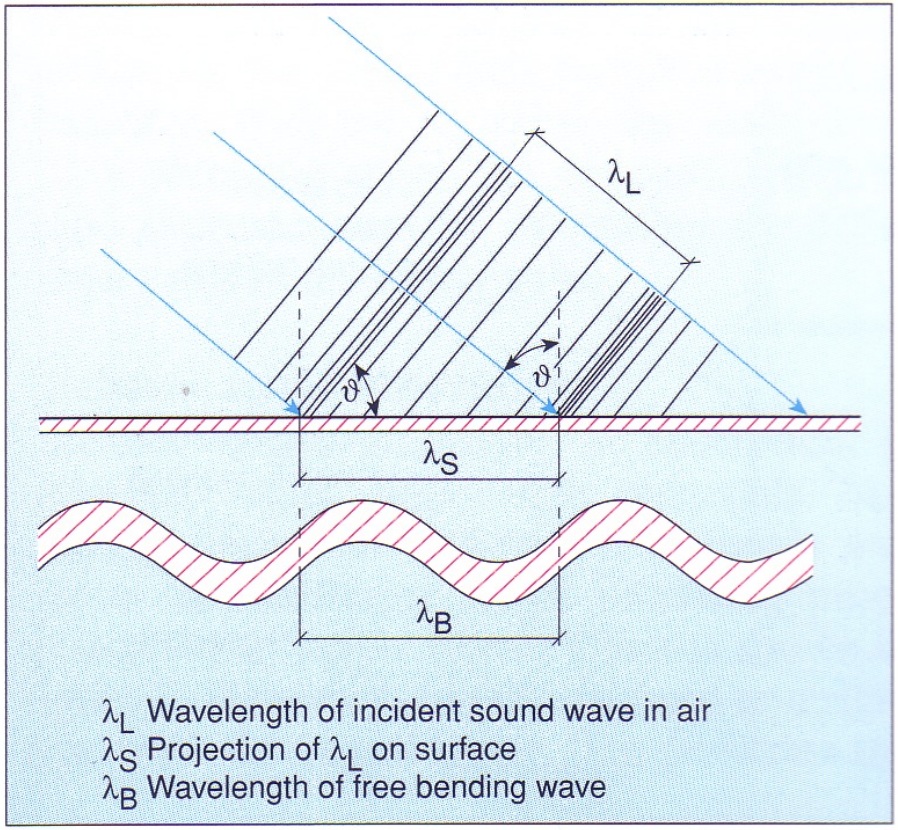


9.4 Airborne Noise: Coincidence

Coincidence is given when the projected wavelength of airborne sound incident at an angle correlates with the free bending wavelength of the material.

The greater the material´s flexebility, the higher the coincidence frequency!

At the coincidence frequency the STL has a minimum.



9.5 Sound Transmission Loss (STL) Test Methods

There are basically two methods to determine Sound Transmission Loss (STL) of a material or a component in the STL test suite:

1. **Sound Pressure Level (SPL) Method:**

$R = L_{SPLA} – L_{SPLE} - L_{K};\ L_K = 10*lg(S/A);\ A=0,163*V/T$

$L_{SPLA\,/\,SPLE}$: SPL in Source / Receiving Room

S: Area of test sample; A: equivalent absorption area

2. **Sound Intensity Level Method:**

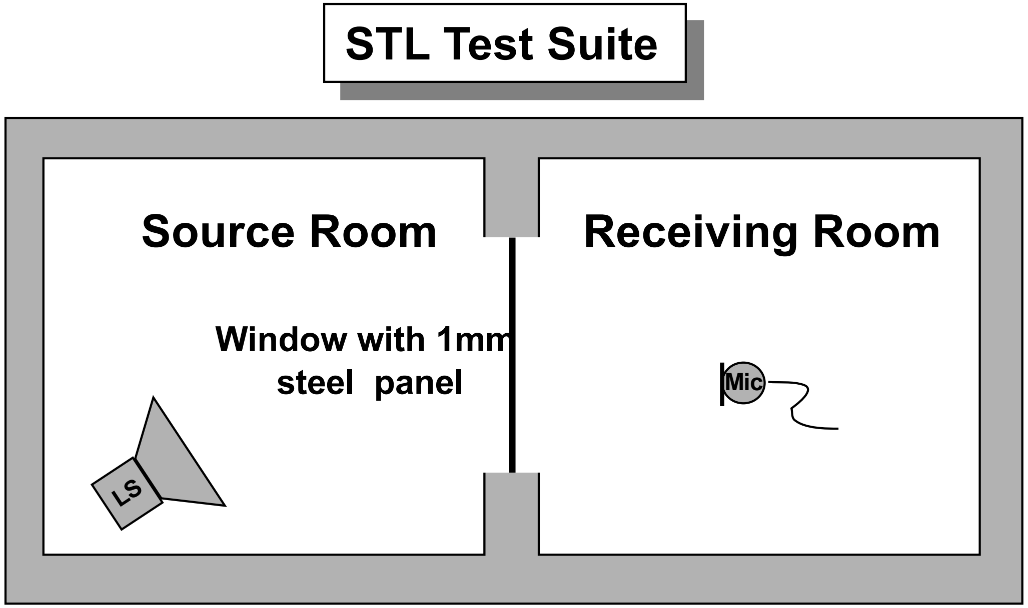
$R = L_{SPLA} - 6dB - L_{SIE}$

L_{SPLA}: SPL in Source Room

L_{SIE} : Sound Intensity Level in Receiving Room

- To determine the Sound transmission Loss (STL) according method 1 (SPL test) room parameter e.g test area, equivalent absorption area, and RT60 of the receiving room have to be known. Unfortunately RT60 of the receiving room has to be measured each time a new sample will be tested.
- To be independent of the room parameter usually method 2 (Intensity test) will be used.

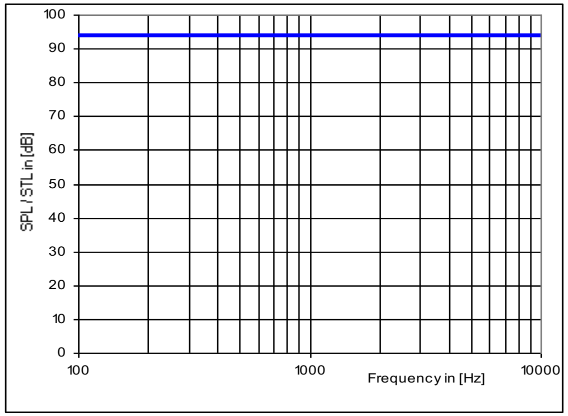
9.6 Sound Transmission Loss (STL)



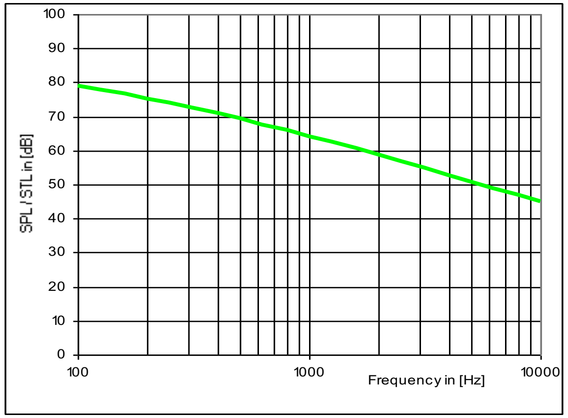
- Source room with loudspeaker and white noise excitation SPL > 94 dB(lin)
- Receiving room SPL (see graph below)
- Sound Transmission Loss (STL)

STL = SPL_{SR} – SPL_{RR} – Room constant

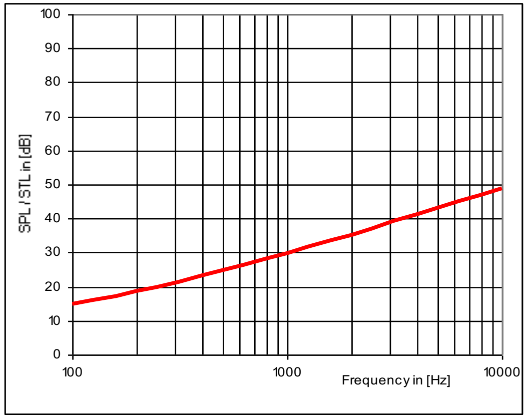
SPL in Source Room = 94 dB



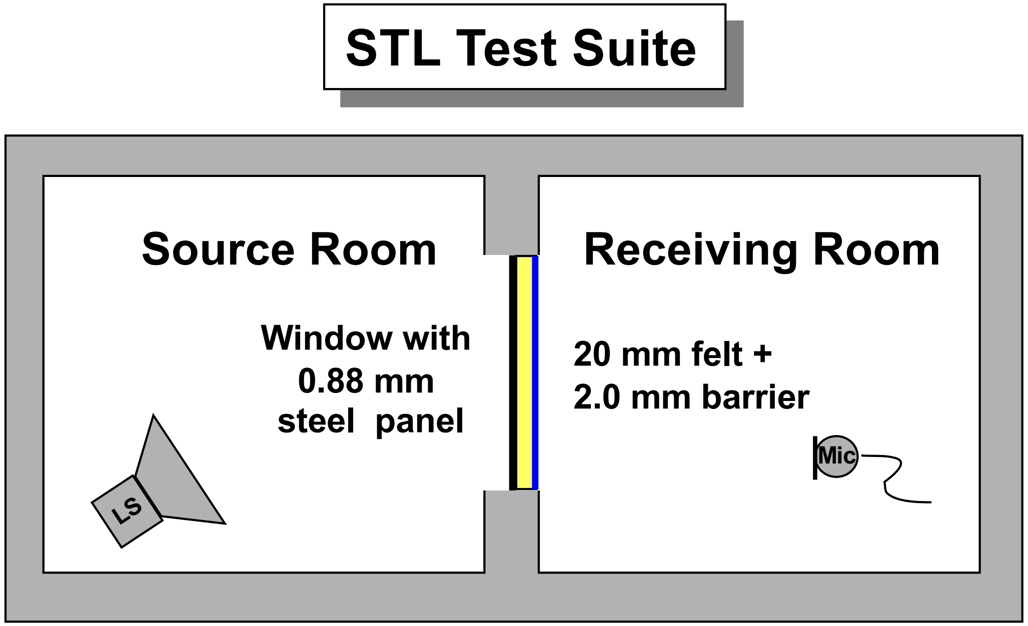
SPL in Receiving Room



Sound Transmission Loss (STL)



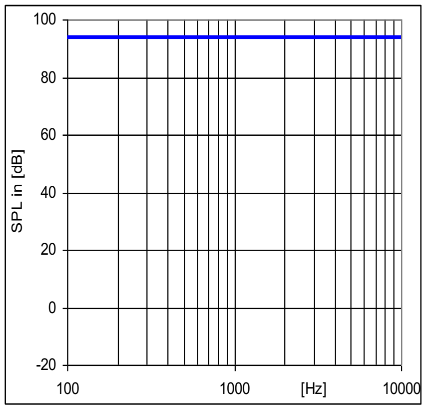
9.7 Sound Insertion Loss (SIL)



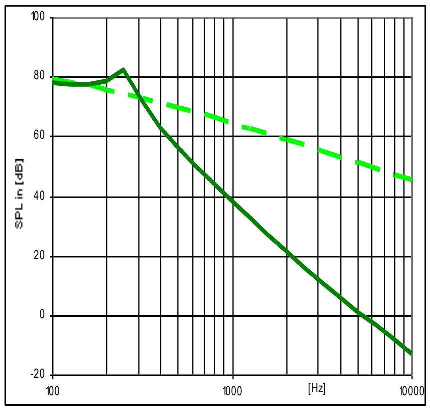
- Source Room (SR) with loudspeaker and white noise excitation SPL > 94 dB(lin)
- Receiving Room (RR) SPL (see graph below)
- **Sound Insertion Loss (SIL)** (effect of material treatment)

$$\mathbf{SIL = SPL_{RR2} - SPL_{RR1}}$$

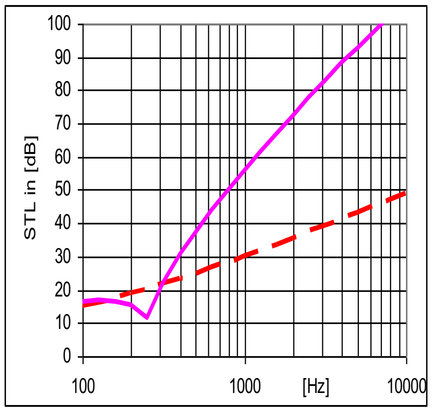
SPL Source Room



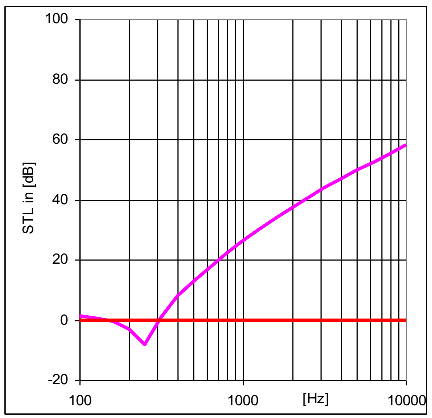
SPL Receiving Room



Transmission Loss (STL)



Insertion Loss (SIL)



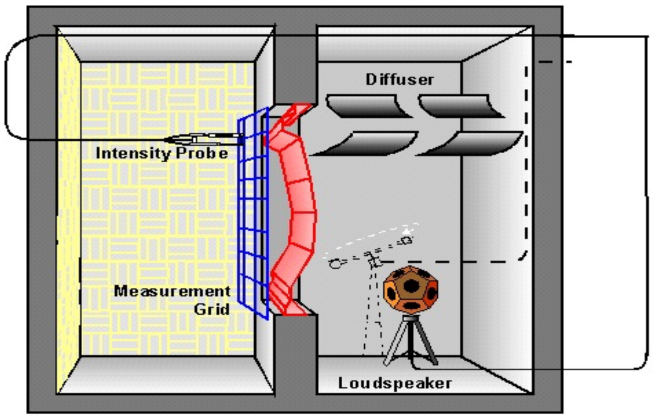
10. Methods to measure STL and SIL

Sound Transmission Loss (STL) and Sound Insertion Loss (SIL) can be measured in many different ways. The classical way in building acoustics is the measurement in the STL Test Suite according well known EN, ISO and SAE standards. To analyze smaller samples, several test methods have been introduced like Tower of Pisa…. A specific SIL test, common in the car industry only, with simultaneous structureborne and airborne excitation is the APAMAT test.

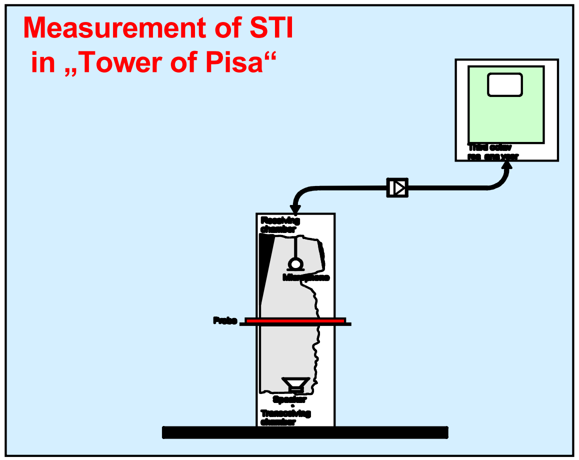
STL Test Suite
according
EN ISO 140 – 3

Tower of Pisa

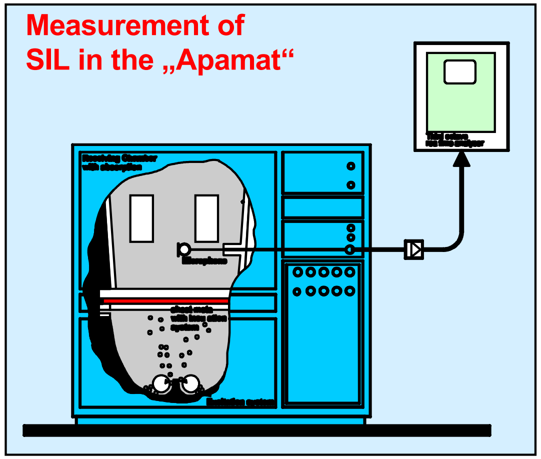
APAMAT
(Car Industry Standard)



Transmission Loss J1400



Measurement of STI in „Tower of Pisa“



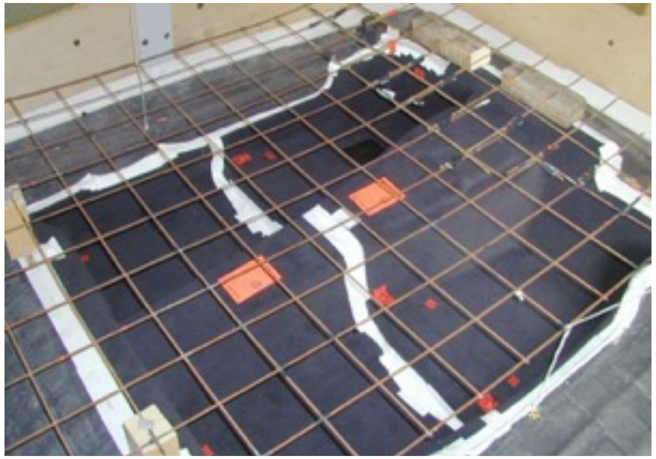
Measurement of SIL in the „Apamat“

10.1 Different STL Test Windows

Reverberation room with diffusors, small & large STL test window

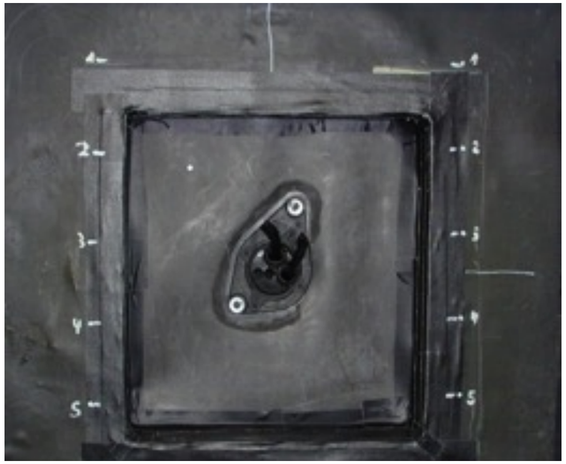
Vehicle door in STL test window (view from receiving room

Small STL test window for material & component tests



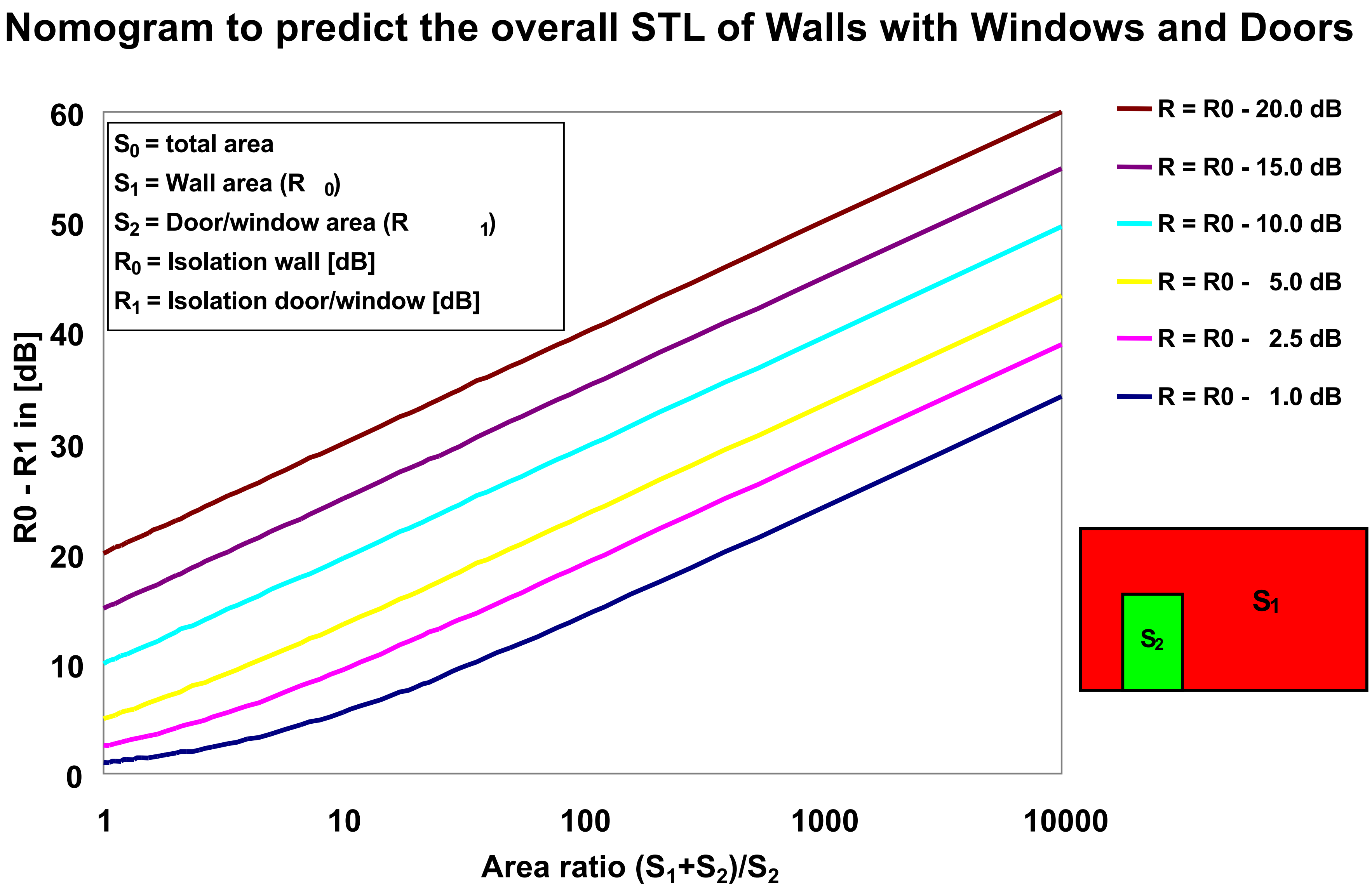
Vehicle floor with insulation & carpet assembly in horizontal STL test window

Complete vehicle front-end in STL test window (view from receiving room)



Cable grommet in STL test window

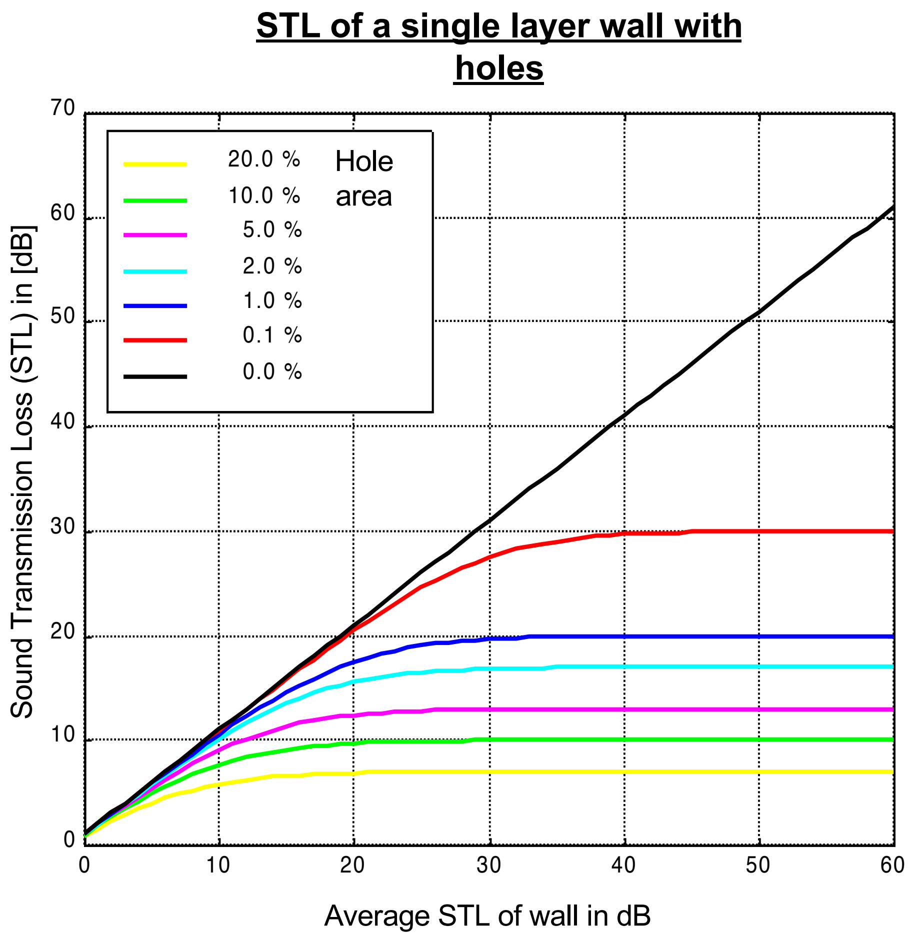
10.2 Influence of Areas with lower STL on overall STL

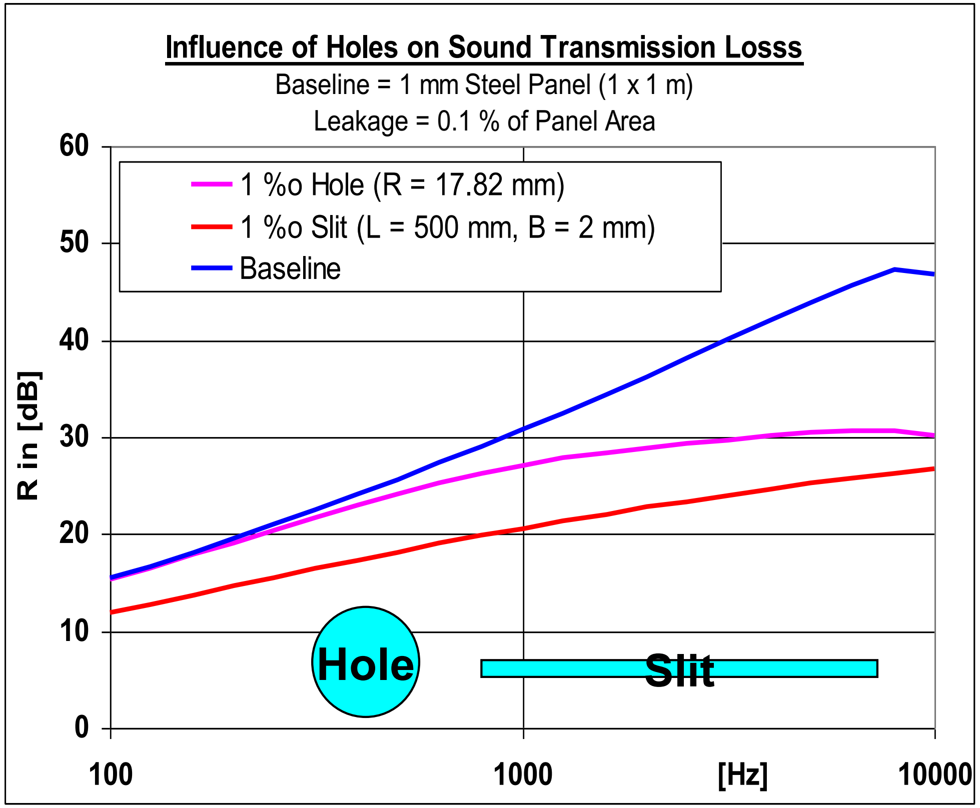


10.3 Influence of Leakage and Leakage Geometry

Influence of round holes in 1 mm steel panel:
Nomogram for theoretical prediction of STL reduction due to round holes

Influence of Leakage Geometry on STL Reduction:
Baseline: Steel panel,thickness = 1.0 mm, area = 1.0 m²
2 different Leakages with equal open area
(1000 mm² = 0.1 % of panel area)
a) Round hole with a radius of 17.8 mm
b) Slit with dimensions of length 500 mm, width 2 mm





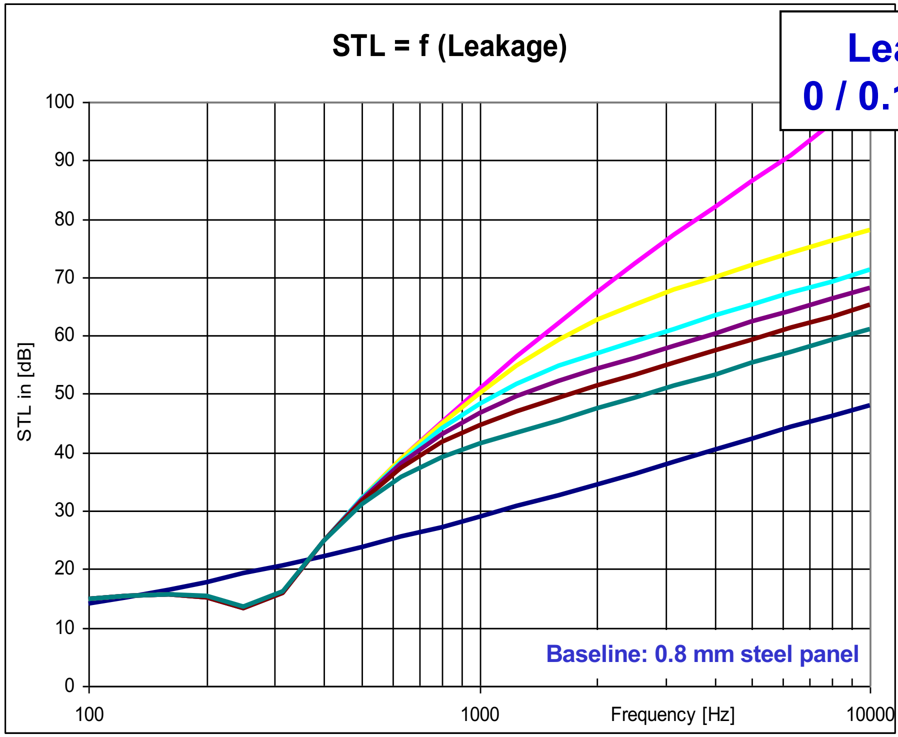
10.4 Sensitivity Analysis of Leakage

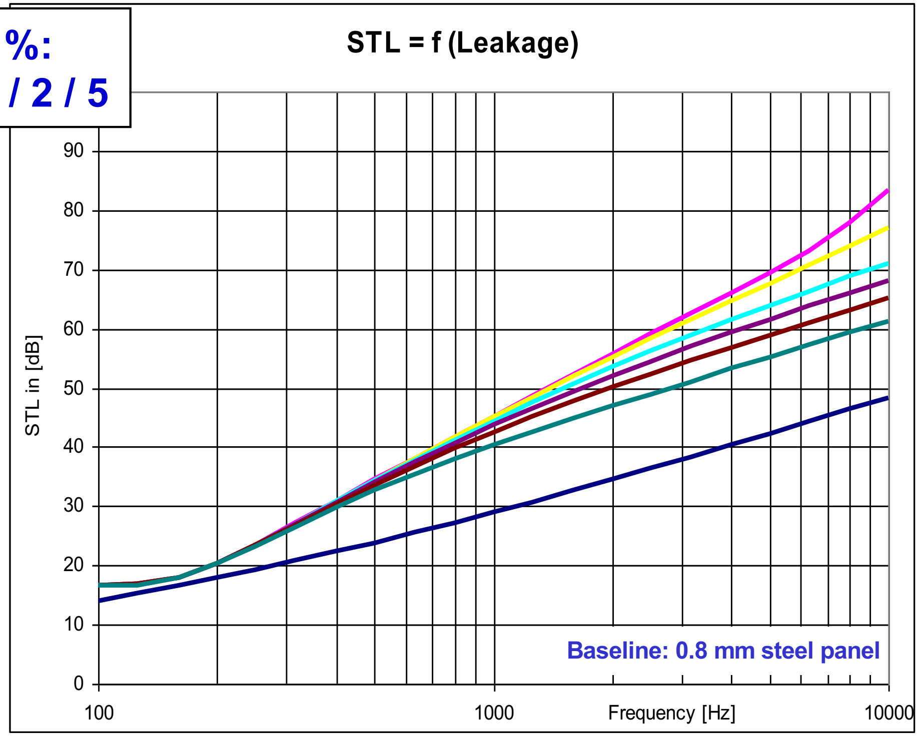
Comparison of Decoupler-Mass-System vs. Dual-Impedance-System

Leakage Sensitivity of Decoupler-Mass-Systems

Leakage Sensitivity of Dual-Impedance-Systems

Leakage in %: 0 / 0.1 / 0.5 / 1 / 2 / 5

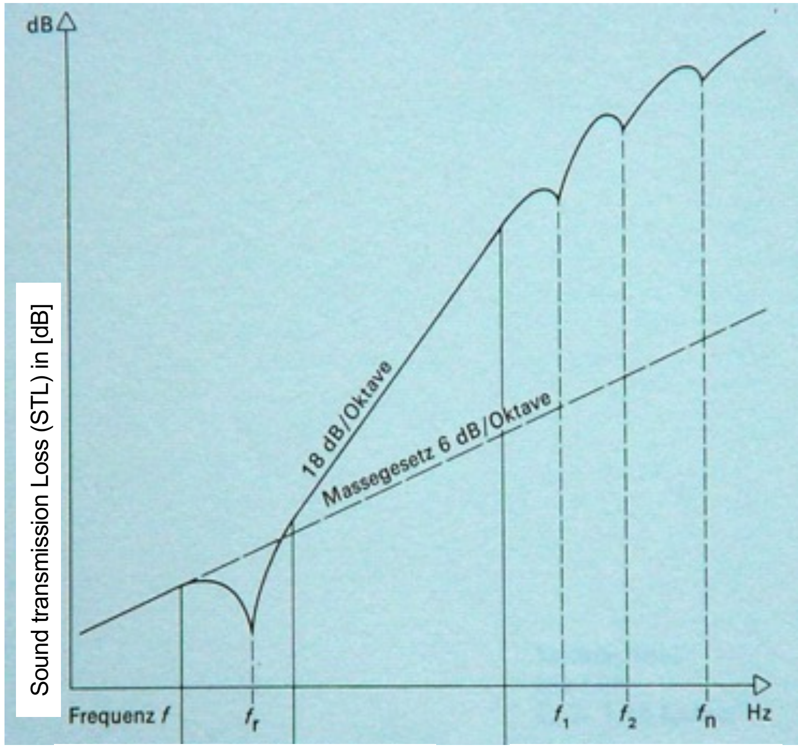




- **Decoupler-Mass-Systems are more sensitive to Leakage than Dual-Impedance-Systems**
- **Typical Leakage of NVH parts are in the order of magnitude of 0.1 to 5 % or even more .**

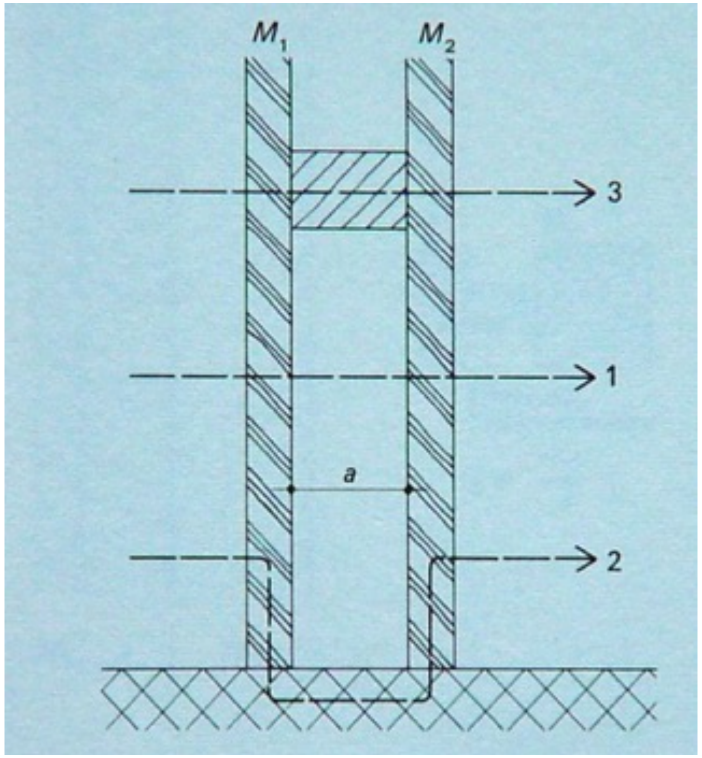
10.5 STL of Double Wall Systems

Principle STL characteristics of a double wall



Region 1	Region 2	Region 3	Region 4
Low frequ. Mass Law	Double Wall resonance	middle frequency	resonances in material depending on thickness

Transmission path in a double wall



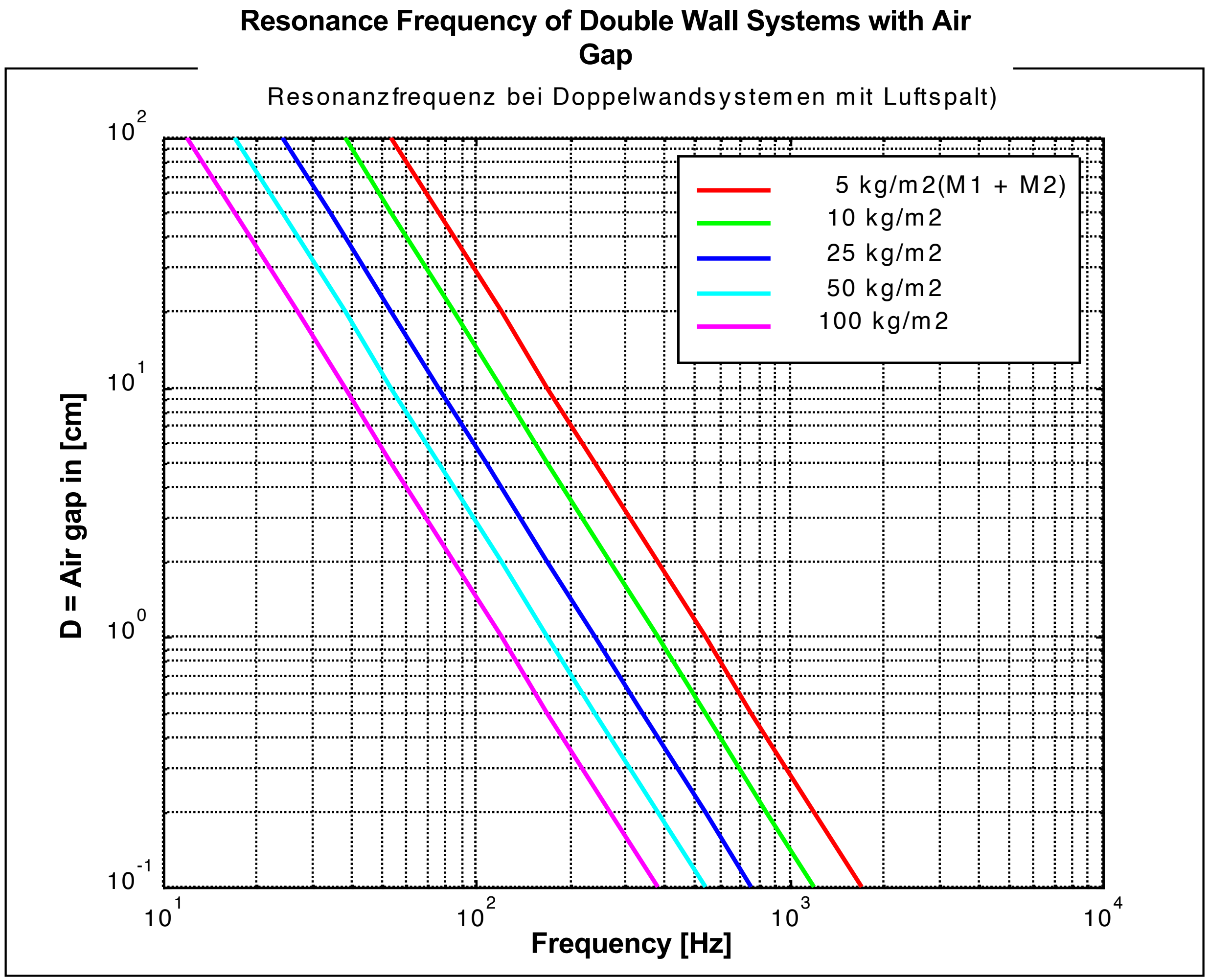
1. **Transmission through cavity (air or absorber)**
2. **Transmission through circumference attachments**
3. **Transmission through structure-borne short cut**

10.6 Prediction of Double Wall Resonance Frequency

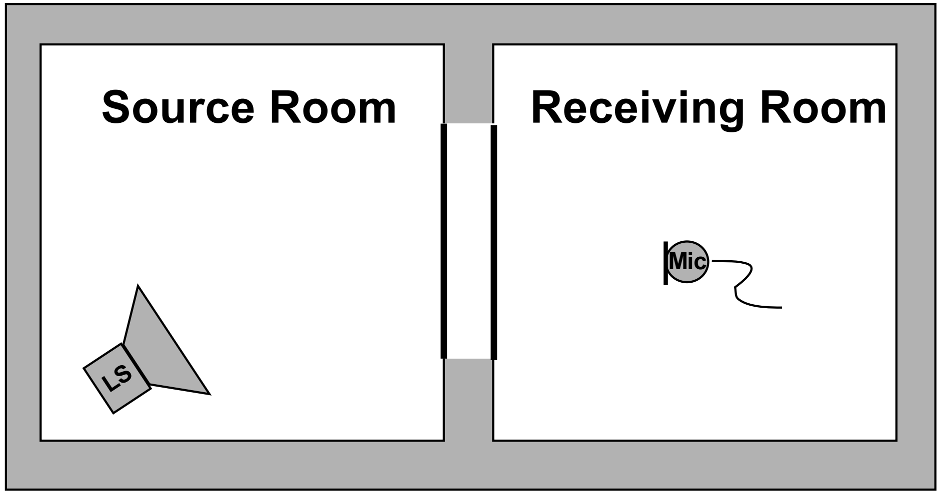
Sound Transmission Loss (STL) of Double Wall Systems:

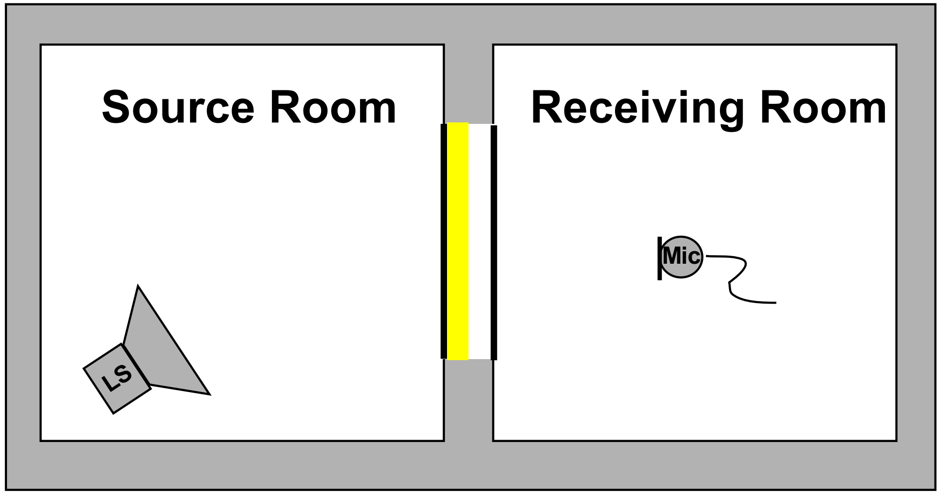
- STL below f_{res} = Mass Law (Mass = m1 + m1)
- STL above f_{res} ➔ + 12 db/Octave
- f_{res} = 500 * Kd / d * M [Hz]

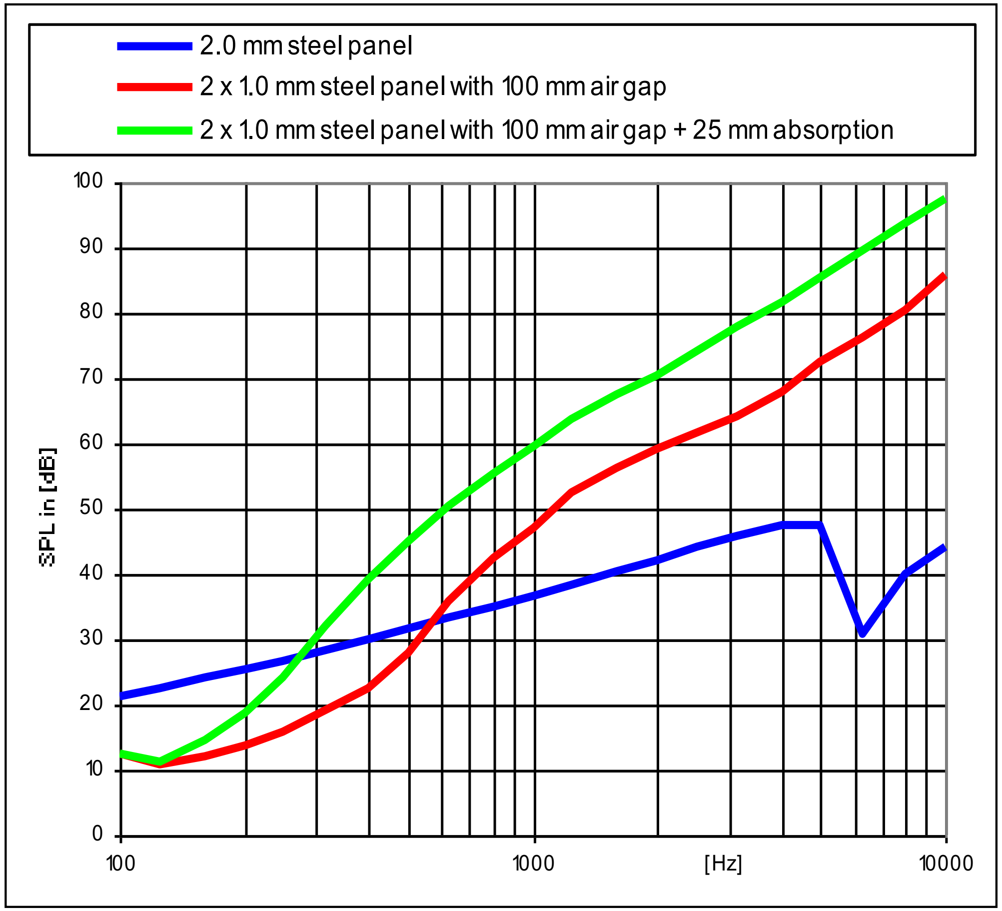
Kd: compression modulus [kp/cm^2]
D: thickness of air gap [cm]
Kd = 0.7 [kp/cm^2] w absorption in gap
Kd = 1.0 [kp/cm^2] air isothermic
Kd = 1.4 [kp/cm^2] air adiabatic
where as the reduced mass (M) equals
M = m1*m2 / (m1+m2)



10.7 STL of Double Wall System with Abs. in Cavity





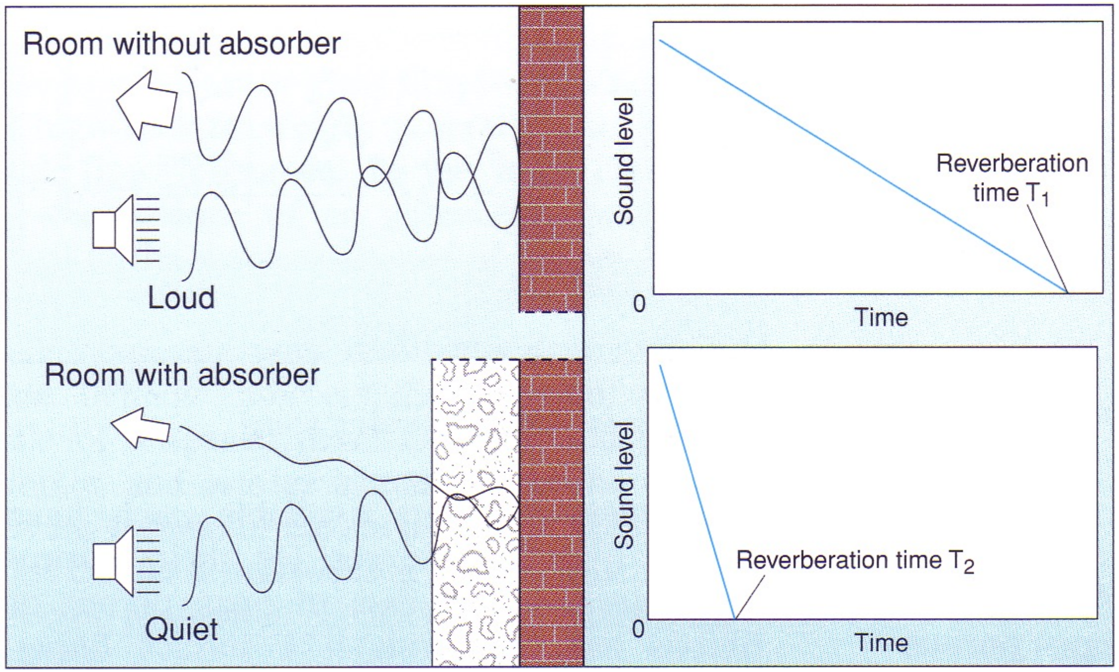


STL improvement due to absorption in double wall cavity

11. Airborne Noise: Sound Absorption

Airborne sound insulation, or absorption, is achieved by using lightweight materials.

All porous, fibrous or non-woven materials and open cell foams have sound absorption properties.



Physically the absorption is achieved due to the conversion of vibrational energy into heat by friction within the material.

11.1 Airborne Noise: Absorption Coefficient

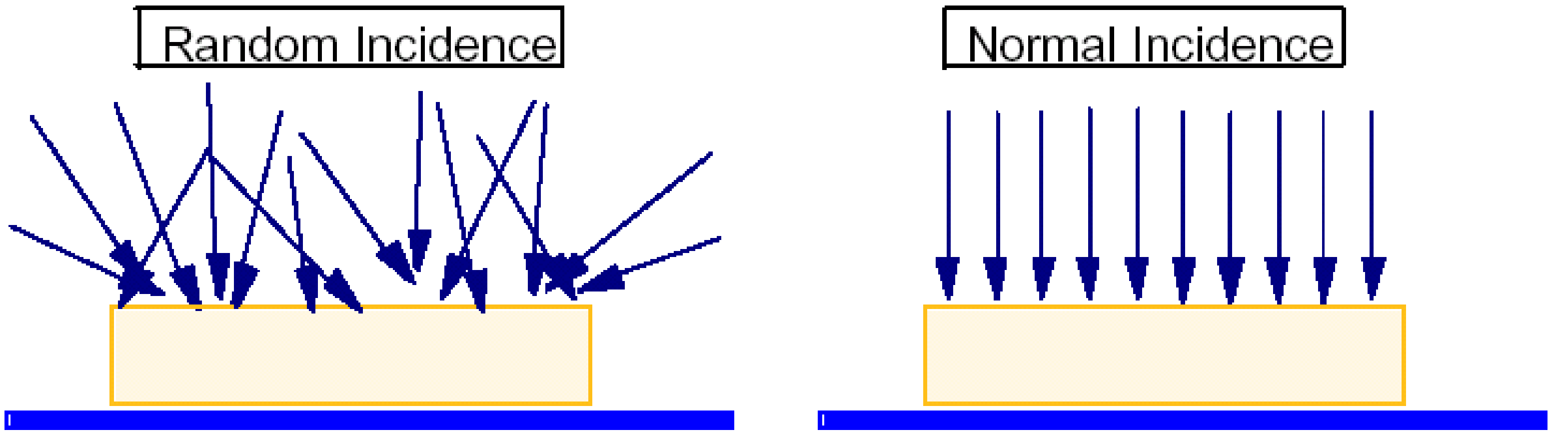
In case of open porous or fibrous materials without a surface lining, e.g. foams, non-wovens and glass wool, the absorption capacity goes up with frequency to a max, followed by a series of minima/maxima.

The law is as follows:

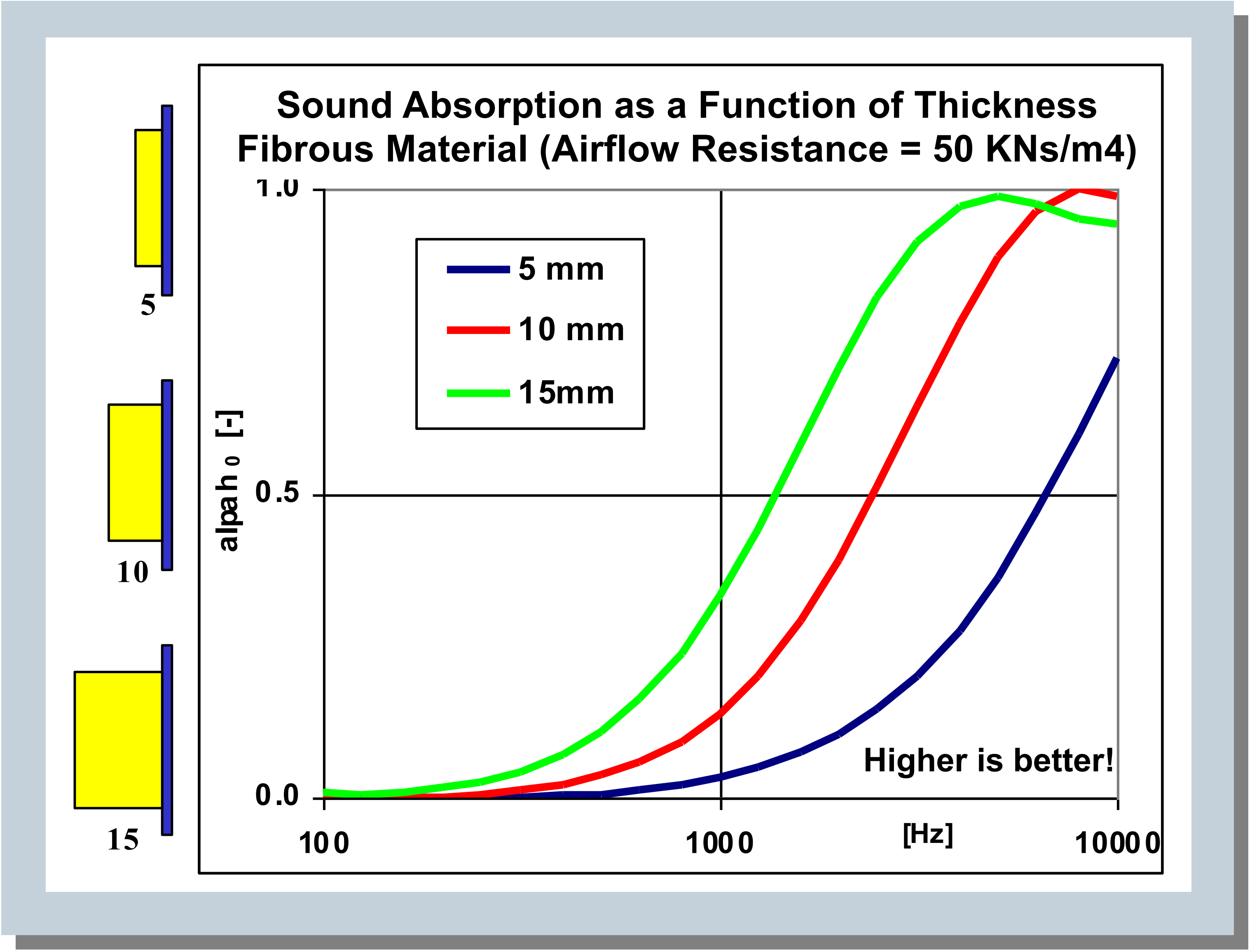
$$f_x(Hz) = t_{ref}(mm)/t_x(mm) * f_{ref}(Hz)$$

By covering the surface of such materials with sheets, liners or non-wovens, an absorption band is created with a lower frequency max for the same material thickness.

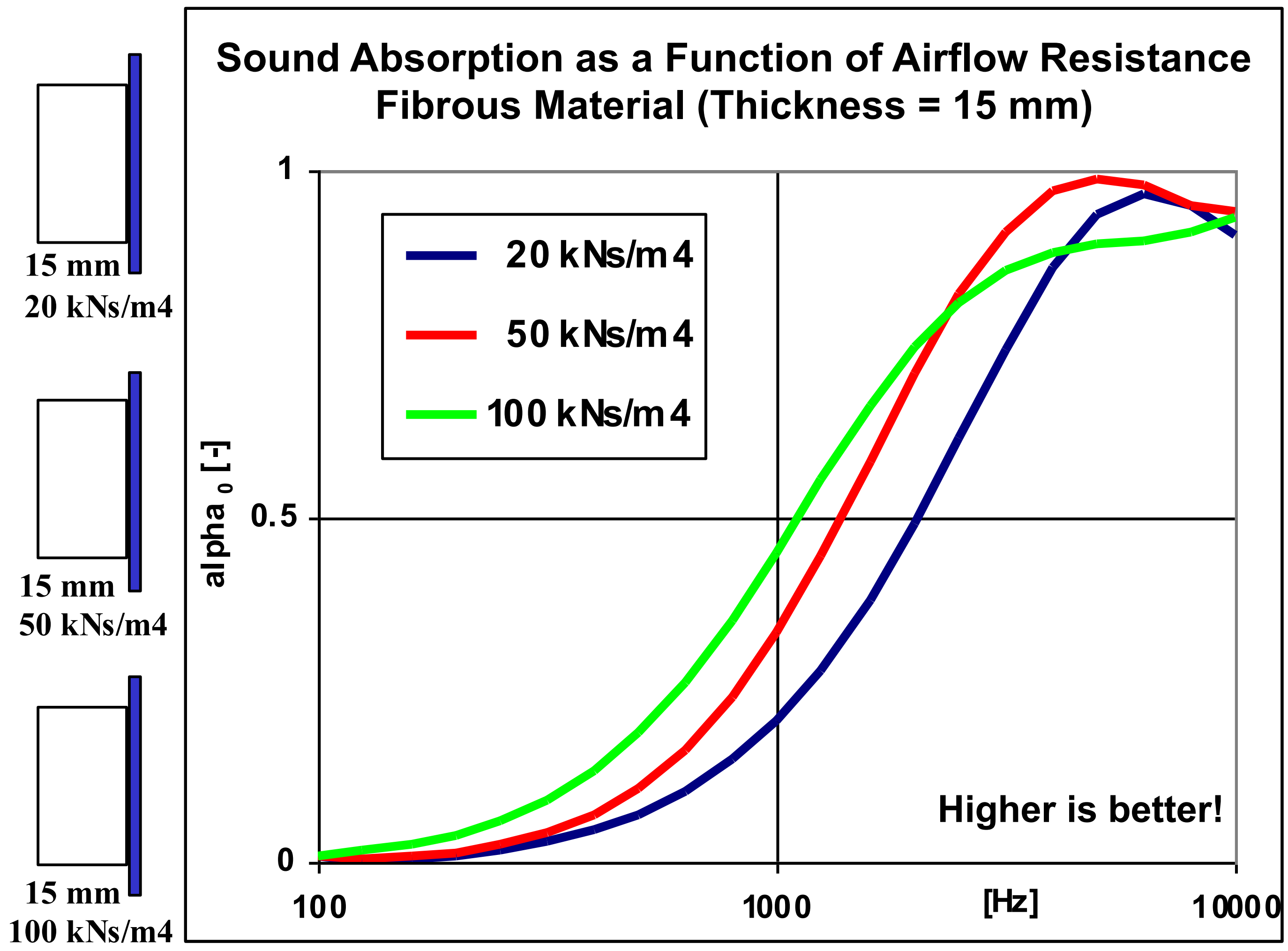
Wherever possible, absorber liners are perforated or other openings are added in order to enlarge the frequency band of the absorber.



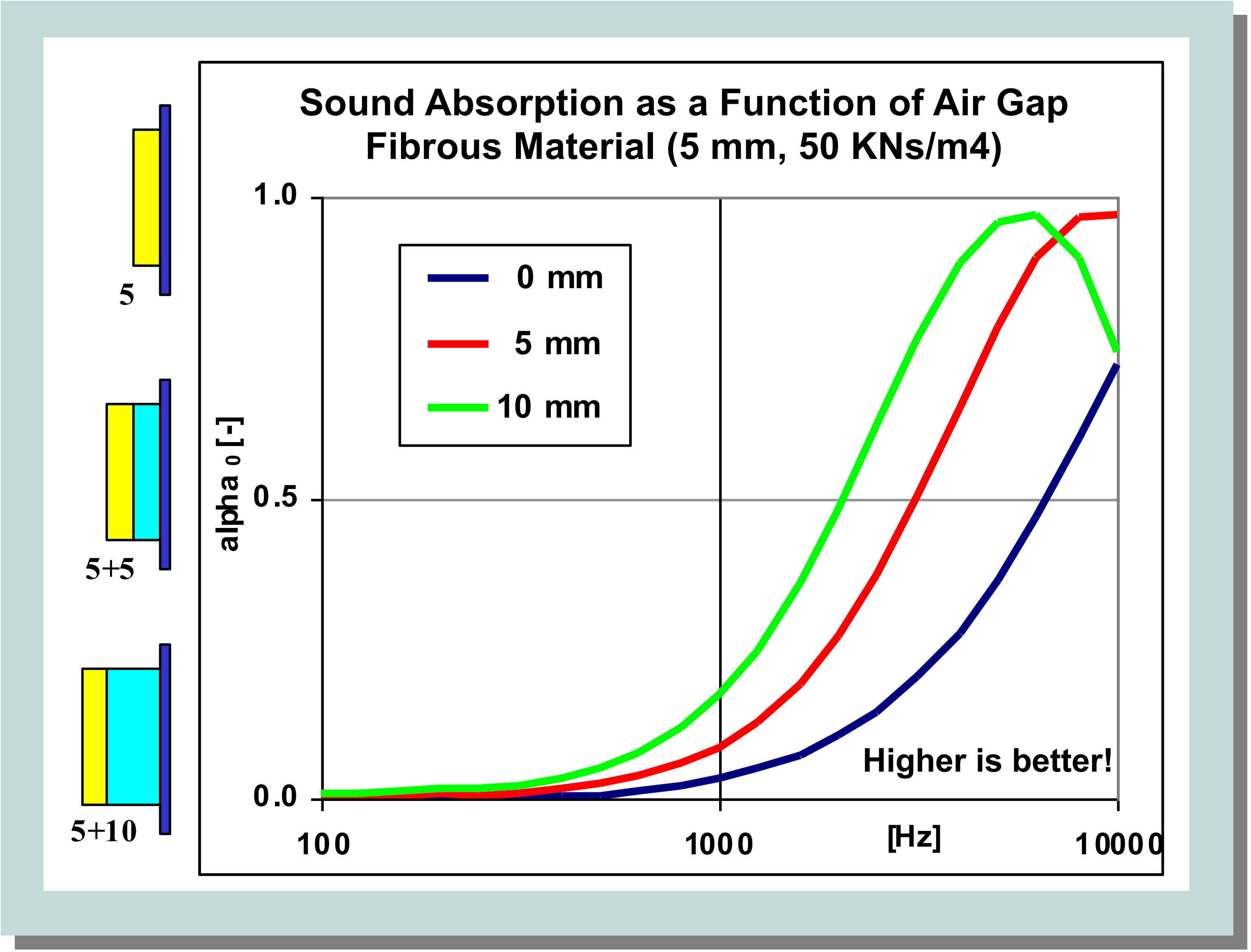
11.2 Absorption as a Function of Thickness



11.3 Absorption as a function of Airflow Resistivity

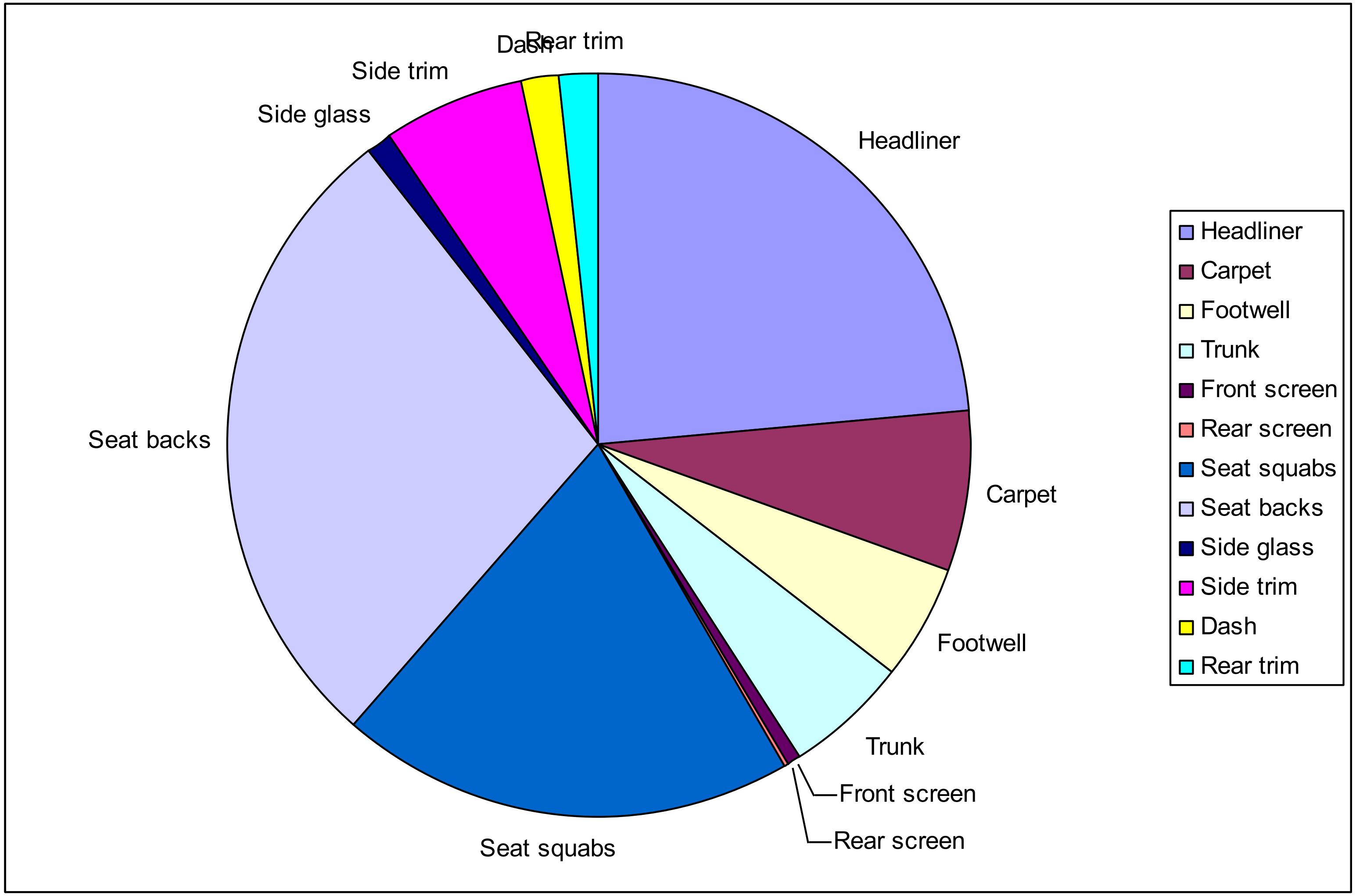


11.4 Absorption as a Function of Airgap



11.5 Absorption in Cars

Sound absorption in a typical European wagon style passenger car with cloth seats:



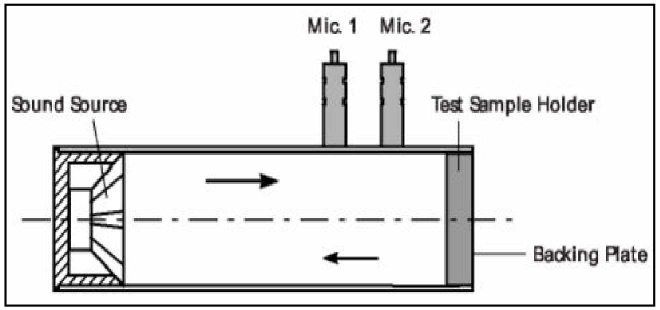
Average SPL difference between fabric and leather seats: **1.1dB** !

11.6 Impedance – Tube: Absorption Alpha

Measurement
according DIN 52215 or ISO 10534

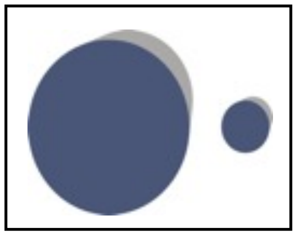
Measurement of the absorption with orthographic sound incidence in the "Impedance Tube"

A loudspeaker is mounted at one end of the tube. At the other end the sample is to be fitted. The loudspeaker generates a wideband noise. Two-dimensional waves disperse inside the tube, impinge on the sample and were reflected. The interference of these two waves generates a standing wave. By measuring the sound pressure level at two fixed points the absorption coefficient of the material can be defined.



Samples:

Diameter of the samples:
30mm und 100mm



11.7 Reverberation Time (RT 60)

<u>Sabine Equation:</u> (W.C. Sabine 1868 – 1919)

Time in Seconds until SPL in a Room is reduced by 60 dB (inaudible).

$T = 0.163 * V / A$

T = Reverberation Time in [sec]

V = Volume of Room in [m³]

A = equivalent Absorption Area of Room [m²]

Absorption Measurement in Reverberation Room:

$\alpha_{Sab} = 0.163\ V / S\ (1 / T2 - 1 / T1)$

V = Volume of Reverberation Room m³]

T1 = Reverberation Time Room empty [sec]

T2 = Reverberation Time Room with Test Sample [sec]

S = Area of Test Sample [m²]

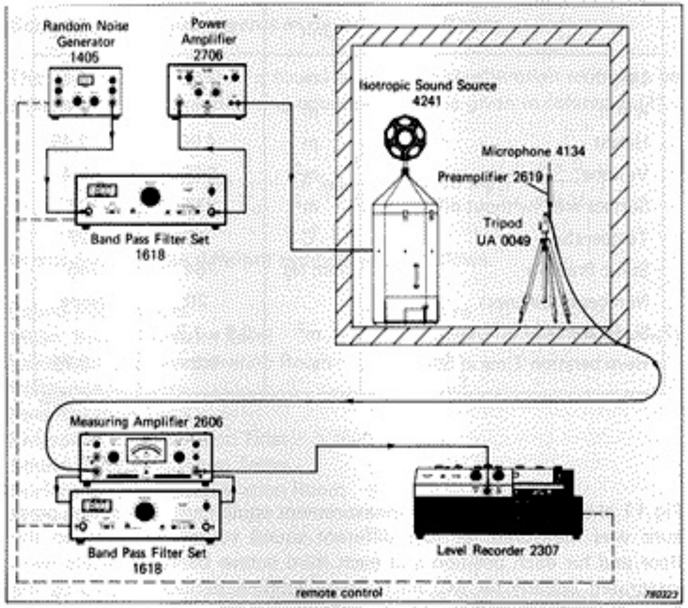


Fig.12. Semi-automatic measurement set-up for reverberation time

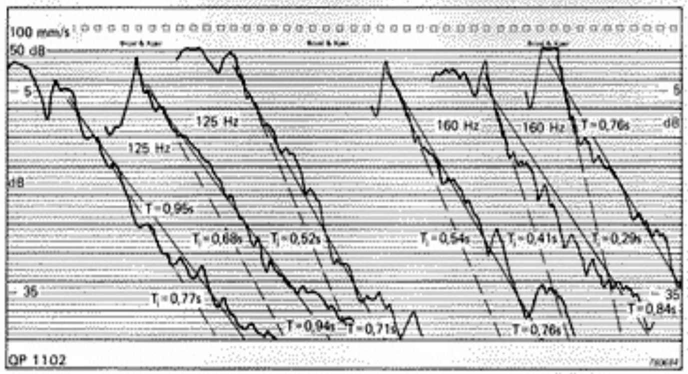
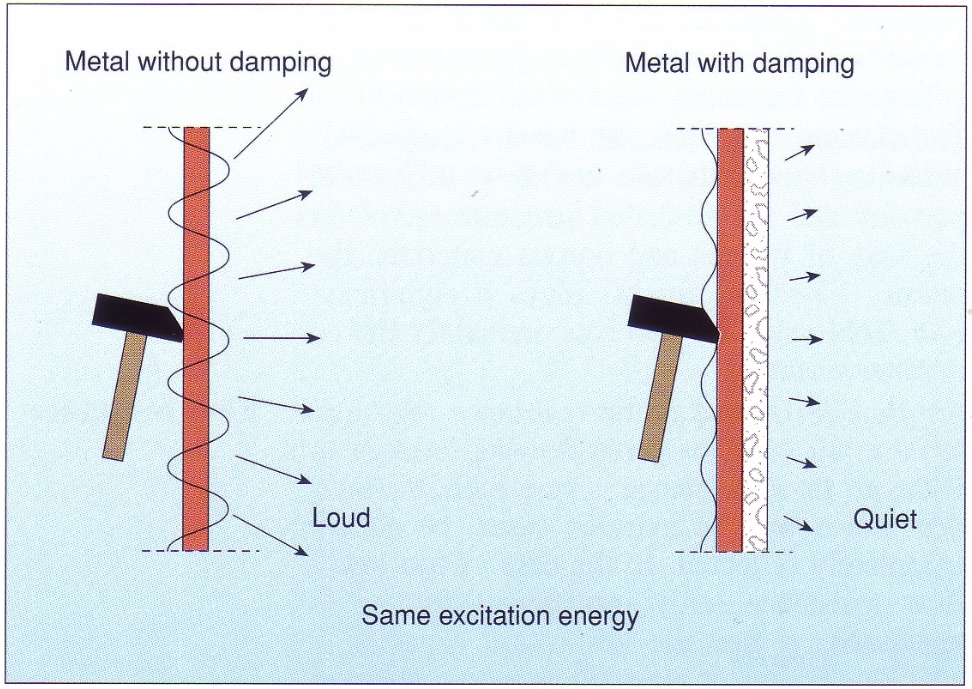


Fig.13. Examples of low frequency decay curves. T is the reverberation time obtained from the slope between −5 and −35 dB levels. T_i is reverberation time from early decay rate

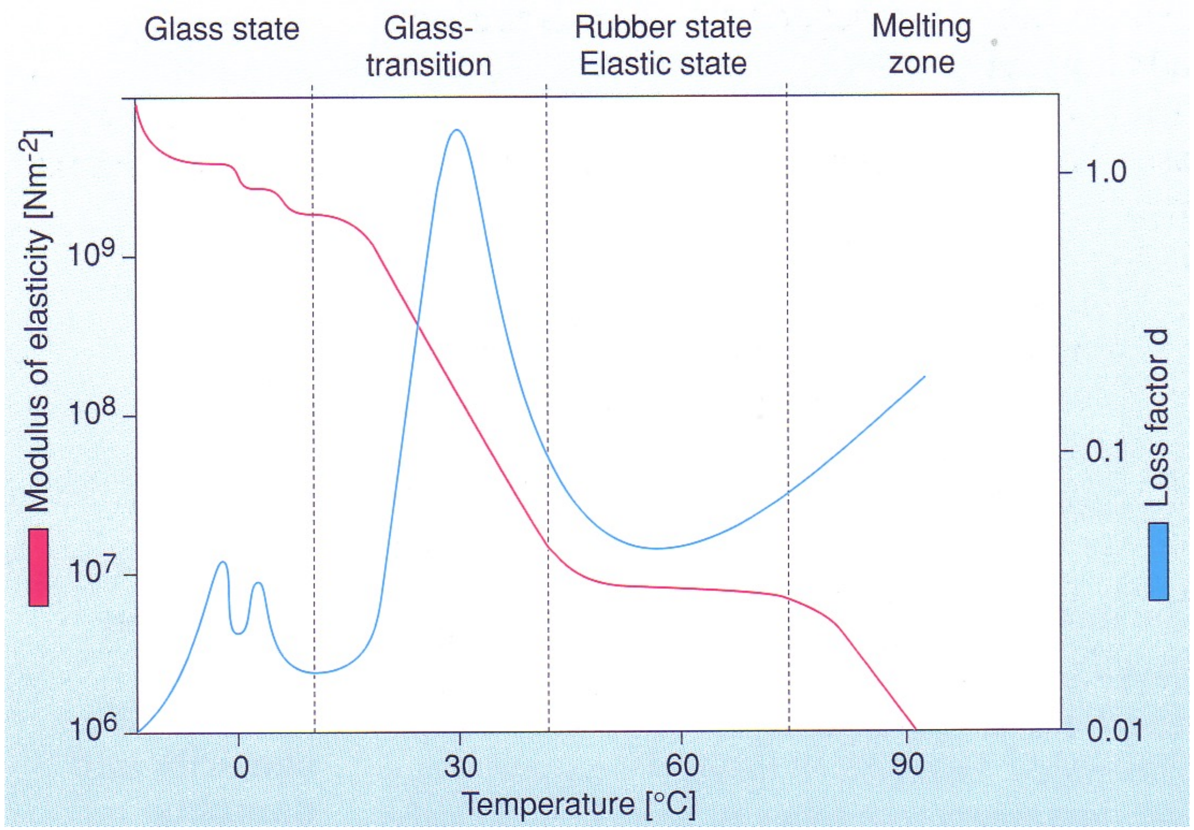
12. Structure Borne Noise: Damping

Structure-borne noise damping, also known as vibration damping, is achieved using single-layer homogeneous coatings and multi-layer coatings, known as sandwich systems.



12.1 Structure Borne Noise: Loss Modulus

The radiation of noise by vibrating metal sheets is mainly dependent upon the bending-vibration modes. Single-layer coatings must have a high modulus of elasticity, depending on frequency and temperature. The imaginary component is called „loss modulus“. The real component is known as „storage modulus“.

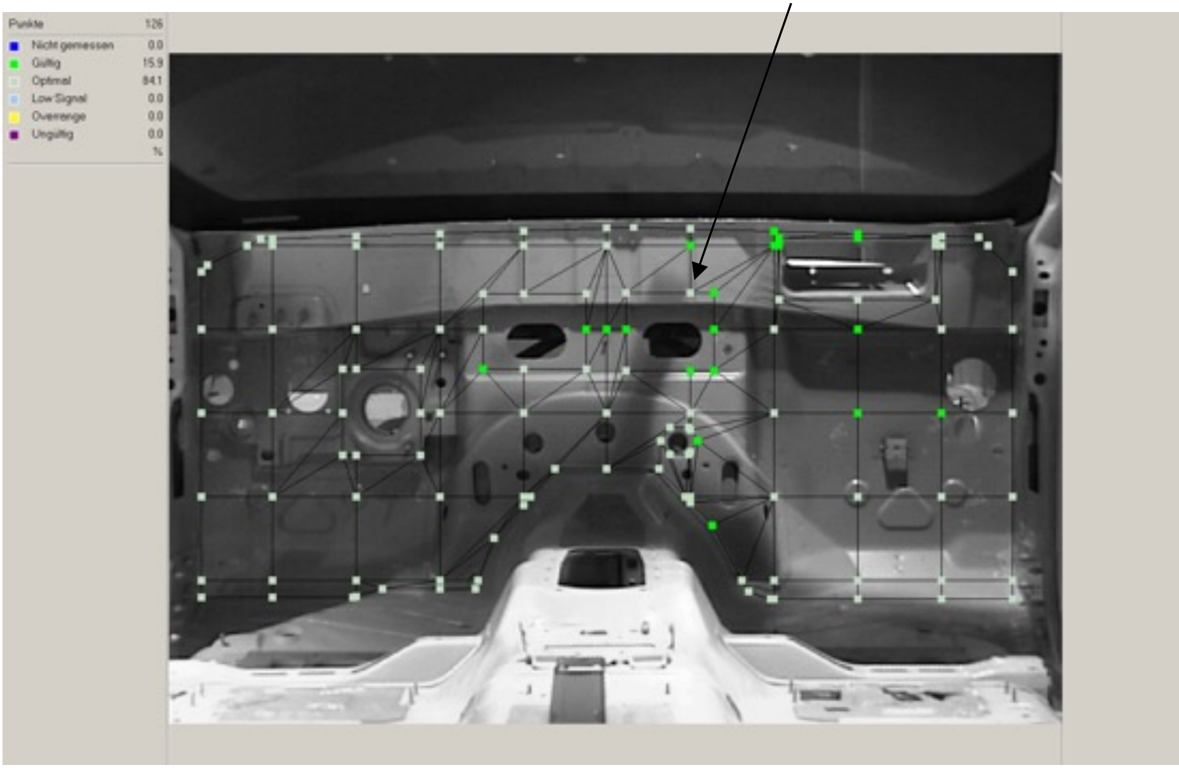


12.2 Panel Vibration

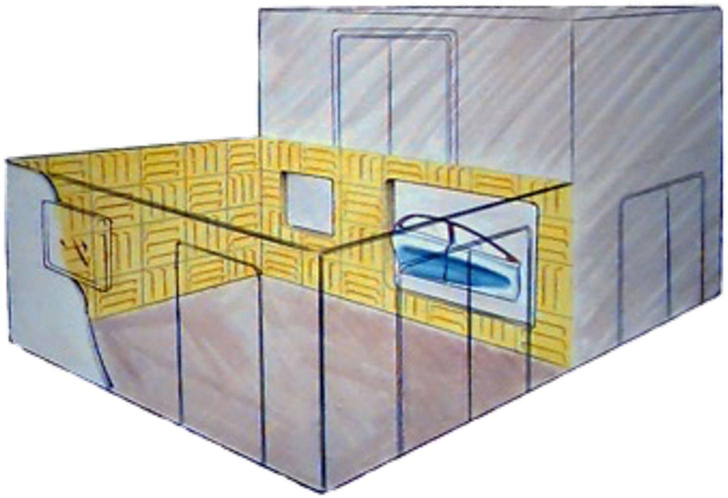
Techniques :

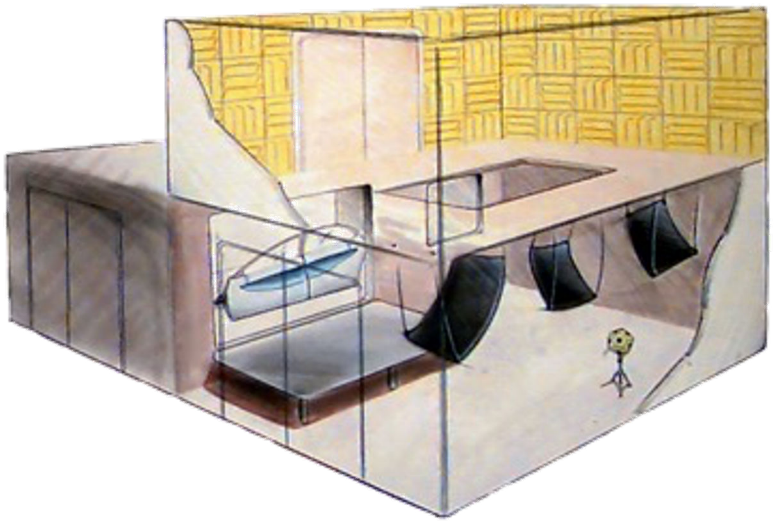
- Acoustic sensitivity (shaker excitation)
- Modal analysis of the body in white with structural modifications
- Damping and/or stiffness modifications
- Laservibrometer

Laser scanning points



13. STL suite and other test riggs

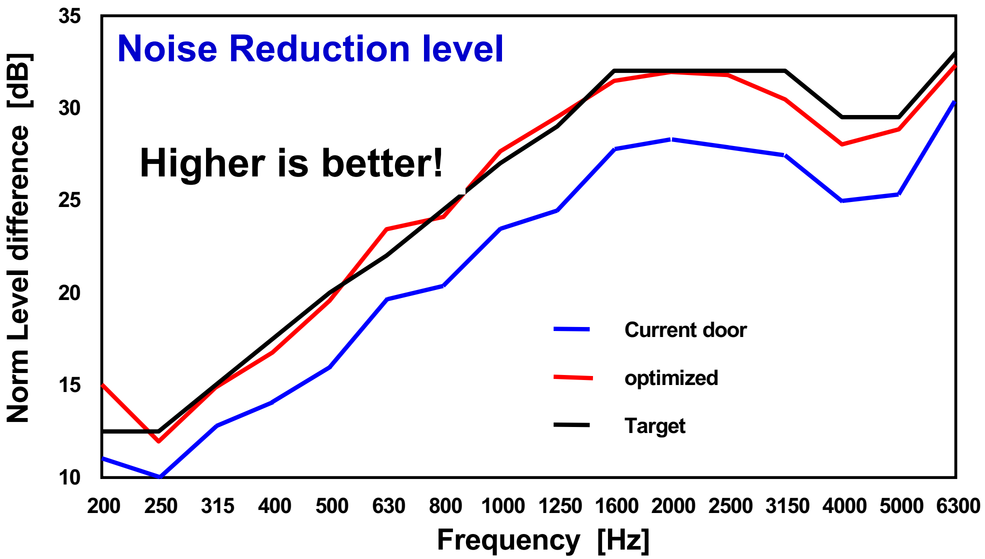


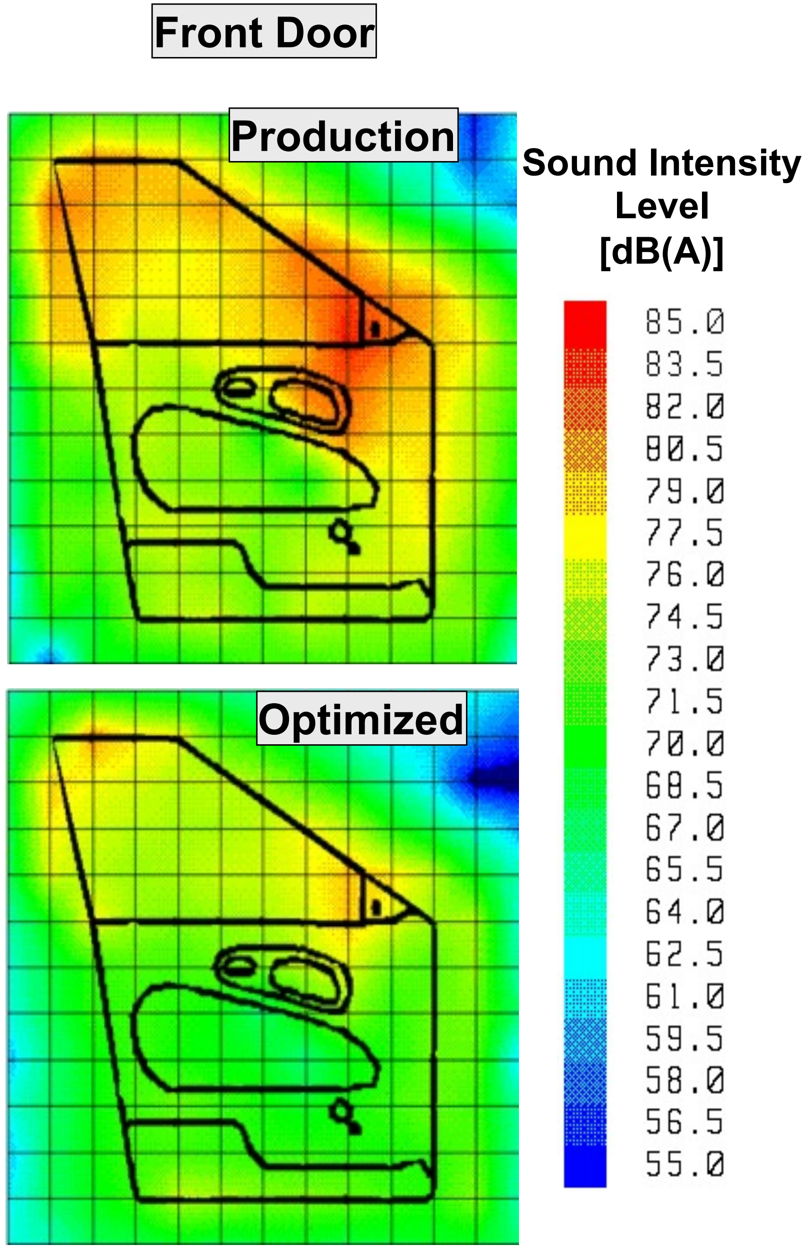


- Real Horizontal Material Testing
- ISO/ASTM Standard Compatible
- also used for absorption measurements
- Sampel size > 10m²

13.1 Body Sub-System Testing: Door

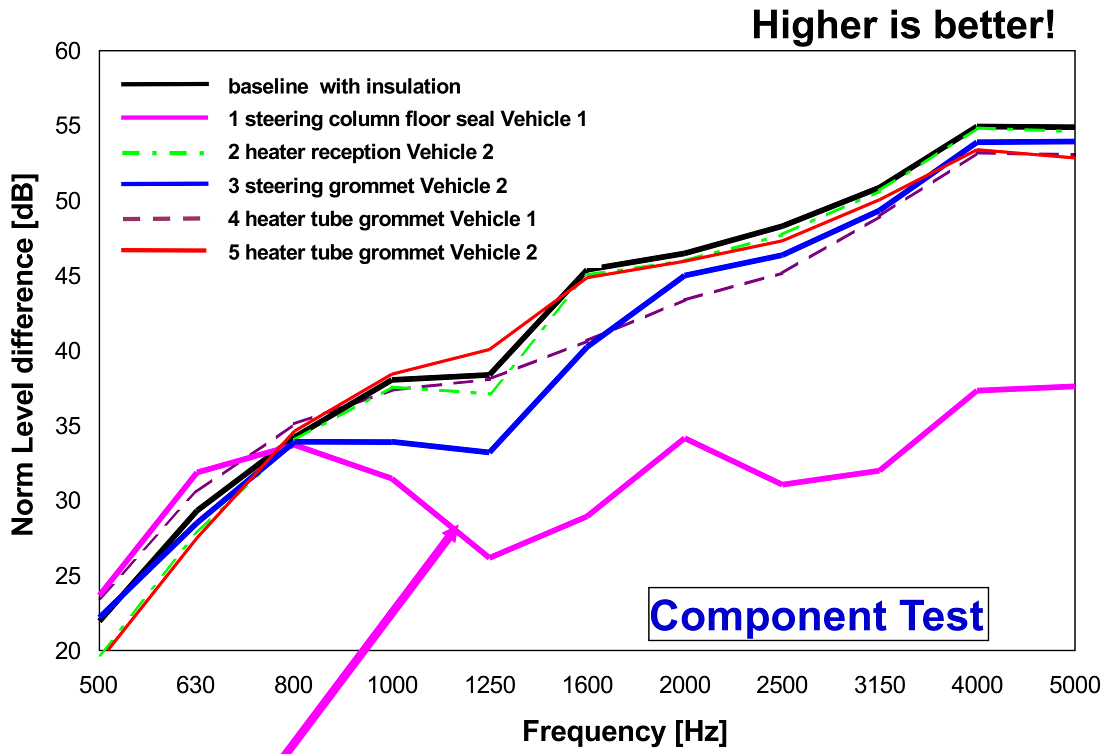
HP Pelzer has several TL test sites at different locations worldwide at its disposal. The size of specimen may vary between a whole minivan side aperture and tiny rubber grommets.
Systematic acoustic studies of Door, Dash and Floor Panel sub-systems have also been carried out during the vehicle development for benchmarking, target setting and sub-system optimization.
The results of those profound body sub-system testsare urgently required for the validation of the analytical SEA models as well.

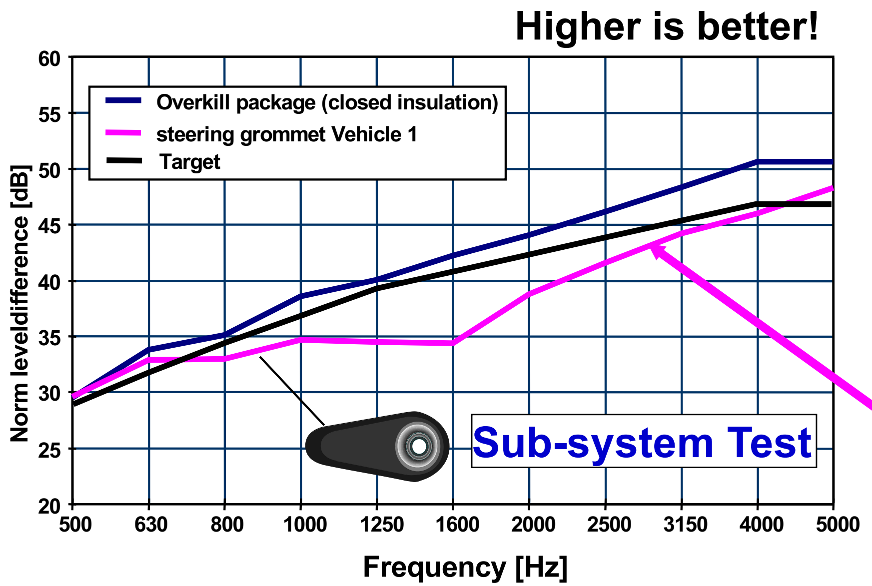


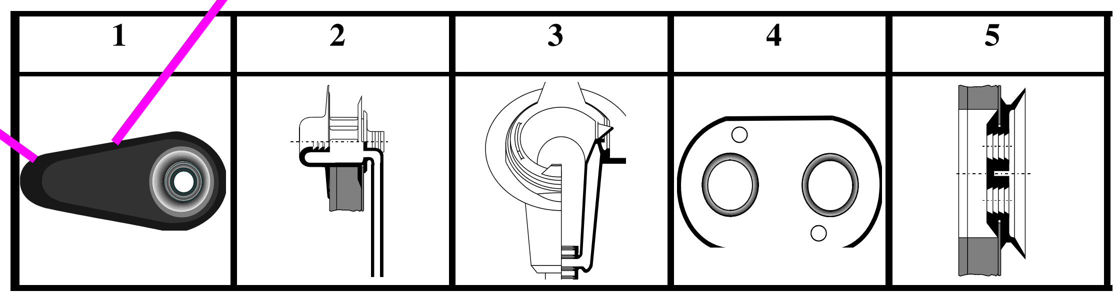


13.2 Component Testing

Body sub-system optimization requires investigations on the included components as well. The results of those component tests are also needed to create libraries with TLs of grommets, seals and insulation parts that later on can be used in analytical SEA models.







13.3 Alpha Cabin

The Alpha Cabin is a reverberation chamber having linear dimensions one-third of those of a standard rev room.

Its volume is **6.44m^3** and no two walls are parallel.

As a result of the size reduction of the room, the sample surface is reduced to **1.2m^2.**

The measurement frequencies are also increased proportionally, so that the useful range lies between 400Hz and 10kHz.

13.4 Semi- or (Hemi) Anechoic Rooms

- Semi-Anechoic (UK) or Hemi-Anechoic (USA) Rooms represent e.g. a road on a free field. The road surface is reflective and the free field is anechoic, that means sound waves will not be reflected.
- In order to bring this very natural and common acoustic condition into a lab, we build a Semi-Anechoic Room with a reflective floor and absorptive walls and ceiling.
- The absorption quality will be defined by the so called “Cut-off Frequency”, the frequency above which more than 99% of sound waves will be absorbed.
- Anechoic rooms have therefore also an absorptive floor. This usually will be realized by a floor made out of a steel net. And the floor underneath is covered with absorptive materials.
- **SPL in free field environment is reduced by 6 dB by doubling of r (r = distance from source)**

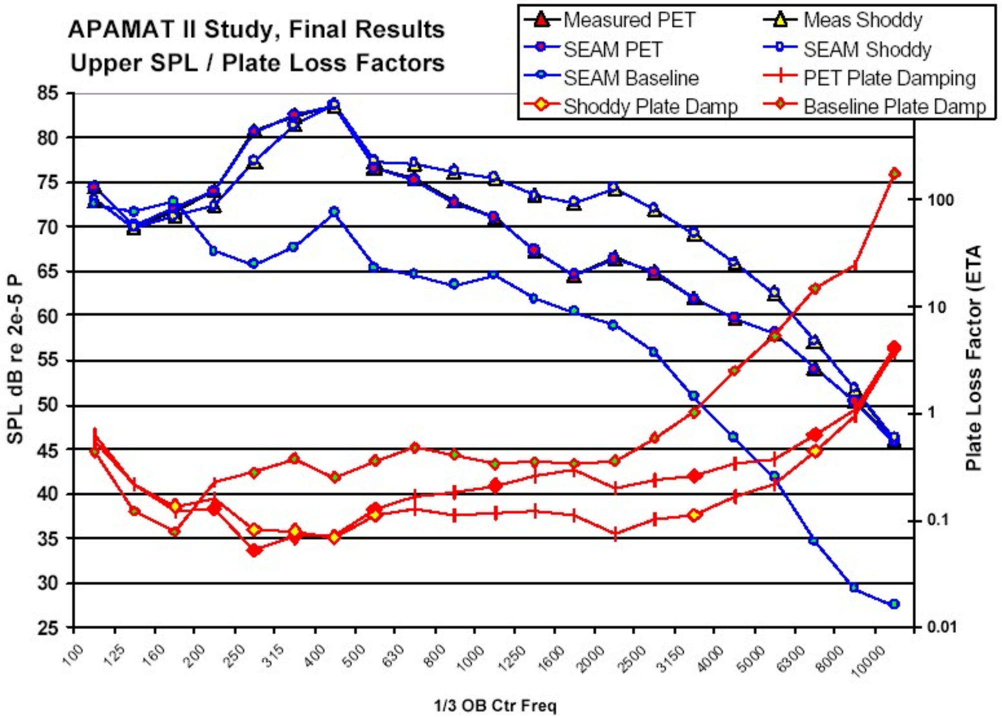
Semi-Anechoic room

STL suite

13.5 Apamat

Apamat II:

Airborne Noise
  Mid to high frequency
Structure borne sound
  Impact of steel balls
Small sample (850mm x 850mm)
  Panel shape, Panel damping/Insulation effect



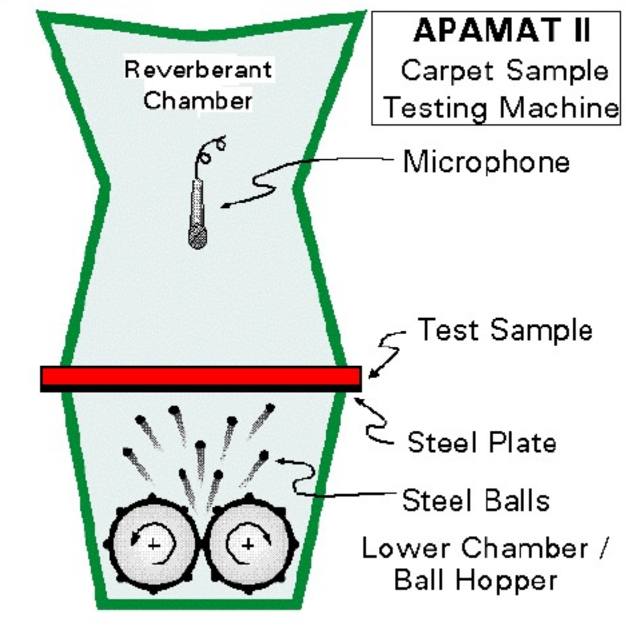


Figure 1: APAMAT II Machine Schematic

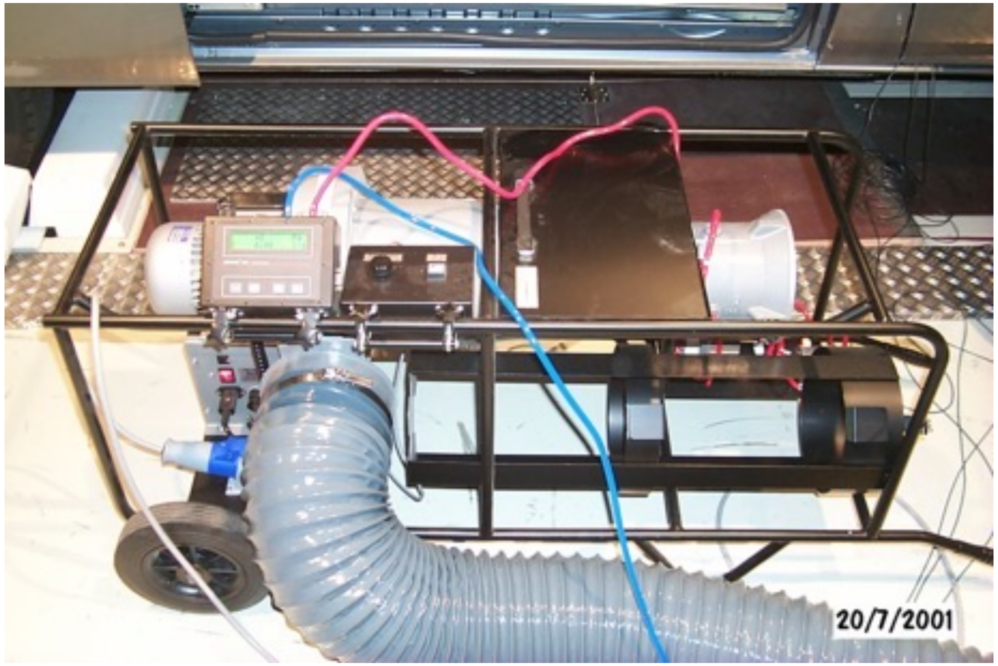
13.6 Cabine Air Leakage Test

Cab Air leakage:

AIRFLOW LM-1 leakage manager:

Max flow rate: 354 l/s

BS848: Part 1: 1997 (ISO 5801)



14. Material Level Testing

Modulus, tensile, tear, density etc

Airflow resistance/ porosity

Compression set, load recovery

Durability -abrasion, flex,

Environmental cycling

Loss factor- complex modulus

15. Modal Analysis

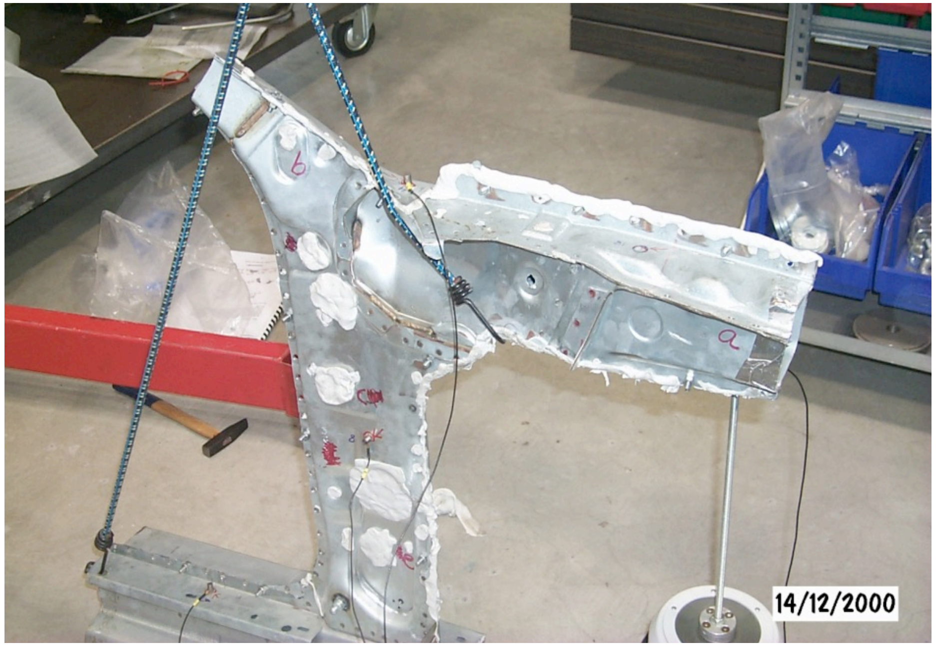
Techniques :

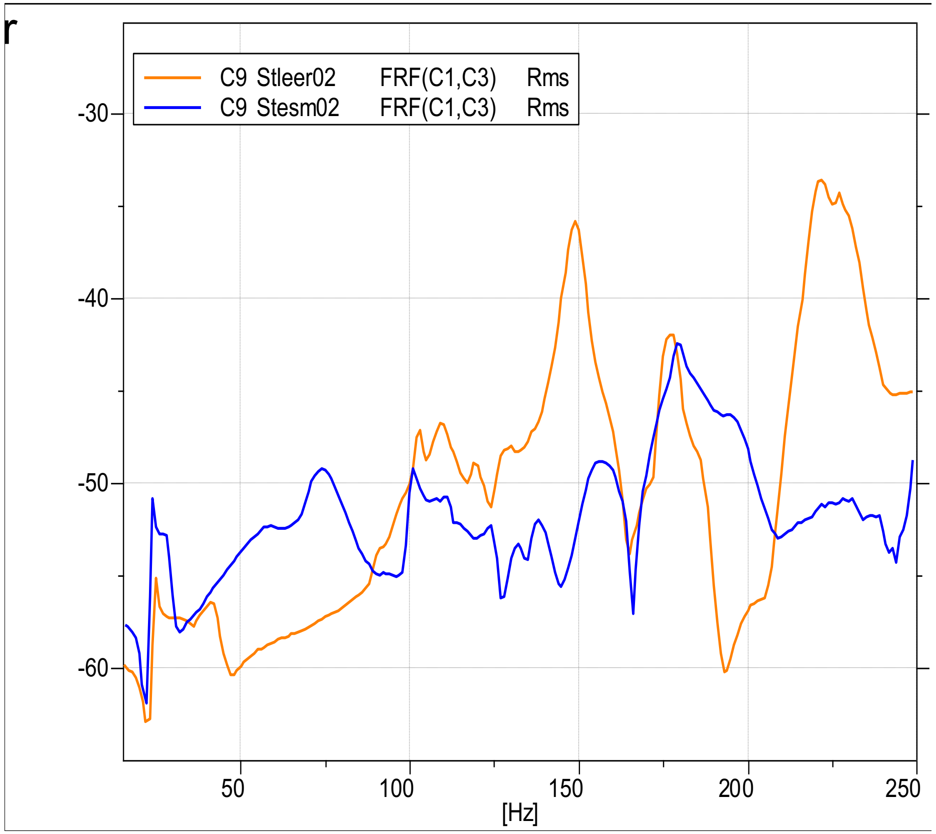
• Acoustic sensitivity (shaker excitation)

• Modal analysis of the body in white with structural modifications

• Damping and/or stiffness modifications

• Laservibrometer/Laser speckle interferometer





Agenda:

- Fundamentals of Acoustics 30 min
- Noise & Vibration Reduction Methods 30 min

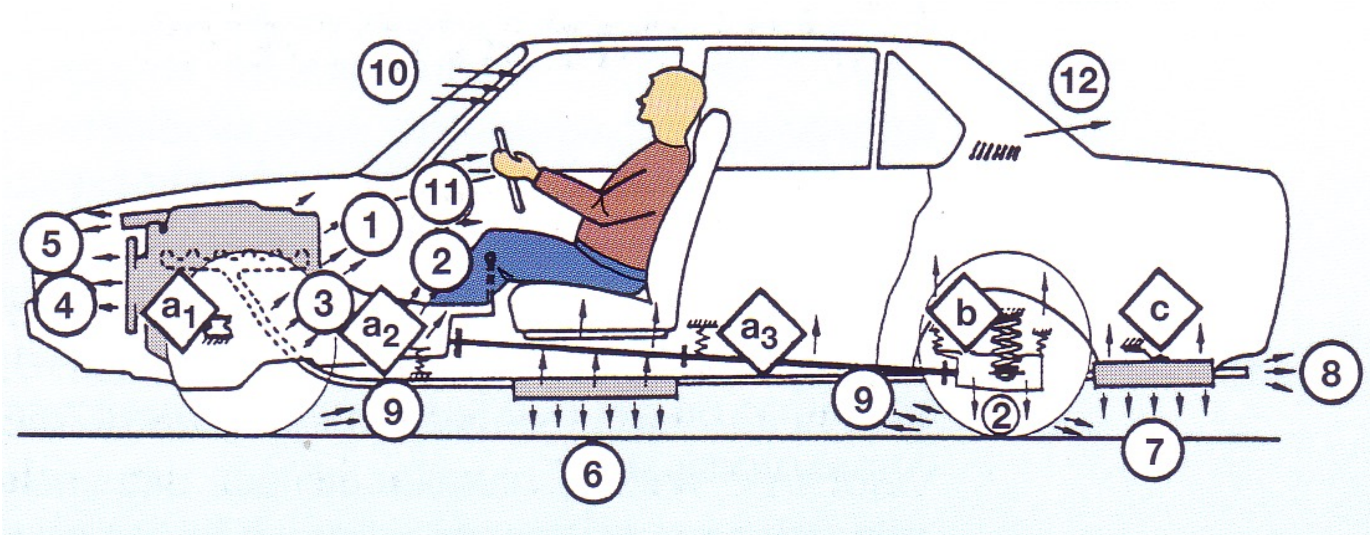
Break

- **NVH in Vehicles 30 min**
- Main Insulation Concepts 30 min

Break

- SEA Prediction 30 min
- Q&A 15min

16. Noise,Vibration & Harshness in Vehicles



Airborne sound

1 Noise radiation from engine block

2 Noise radiation from gearbox/differential

3 Noise radiation from exhaust system

4 Noise from radiator fan

5 Inlet noises

6,7 Noise radiation from silencer

8 Silencer outlet noises

9 Tyre noise

10 Wind noise

11 Ventilation noise

12 Extraction noise

Structure-borne sound

Transmitted via suspension components

a1,a2: engine vibrations

A3 cardan shaft

B Axles: drive structure-borne noise

Road structure-borne noise

C muffler

16.1 NVH Overview

Total Vehicle Targets & Sign Off Criteria (Test & CAE)

Airborne Noise

Structureborne Noise

Airborne Noise Excitation

Windnoise, Tyre & Watersplash Noise
Intake & Exhaust Noise
Powertrain Noise

Airborne Noise Transmission

Structureborne Noise Excitation

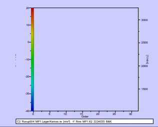
Engine Mount Forces, Exhaust Hangar Forces
Suspension Mount Forces

Airborne Noise Transmission

Transfer path analysis (TPA), FRF
Point Mobilities

System targets

Noise Red. Engine Side to Pass. Ear



Noise Red. Intake Orifice to Pass. Ear

Noise reduction from Tailpipe

Noise Reduction from Tyres to Pass. Ear

Body Transparency

Passenger Compartment Absorption

Subsystem Targets

Transmission Loss: of Dashpanel, Underbody, Doors, Roof and Tailgate
Noise Absorption: Seats

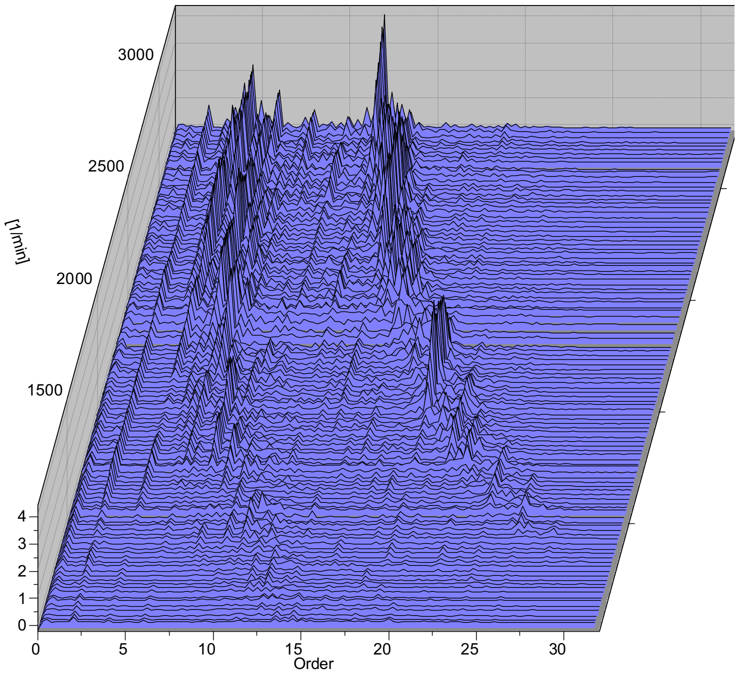
Soundpackage Component targets

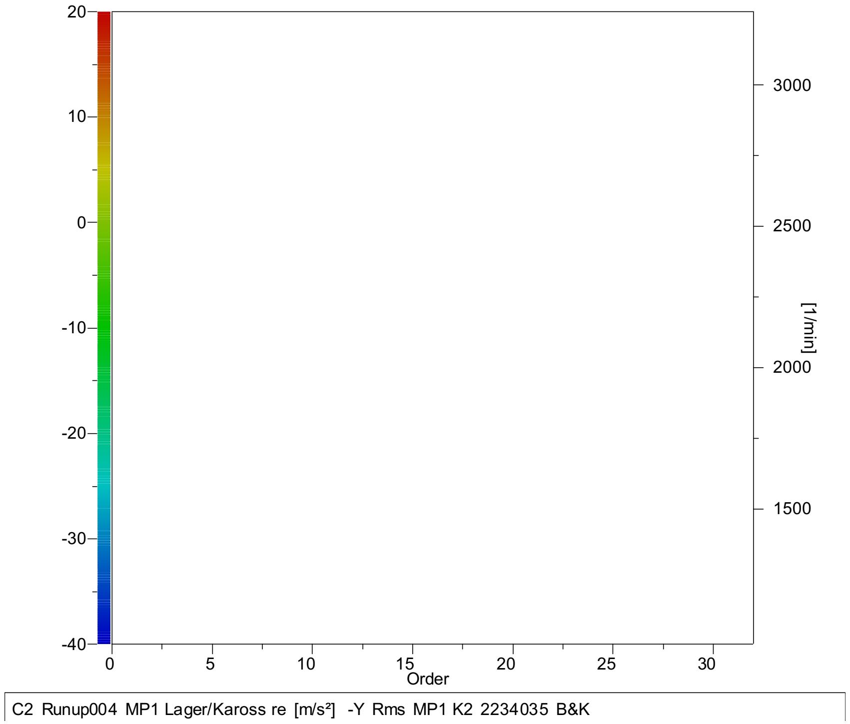
Noise Absorption: Headliner, Side Trim, Carpet, Package Tray, Pillar Trim, Luggage Floor, Wheelhouse Inner/Outer,Air Extraction, Engine Compartment
Transmission Loss: Integrated Carpet, Inner Dash Insulation, Grommets, Heater Box
Loss factor: Deadener Sheets

16.2 Combustion & Powertrain Noise

Complete signature of road, wind and powertrain induced sources

•





16.2 Vehicle Airborne Noise Measurements

Techniques to quantify the degree of barrier control and passthru:

• Full vehicle sound exposure in a reverberation chamber

• Ultrasonic leak detection

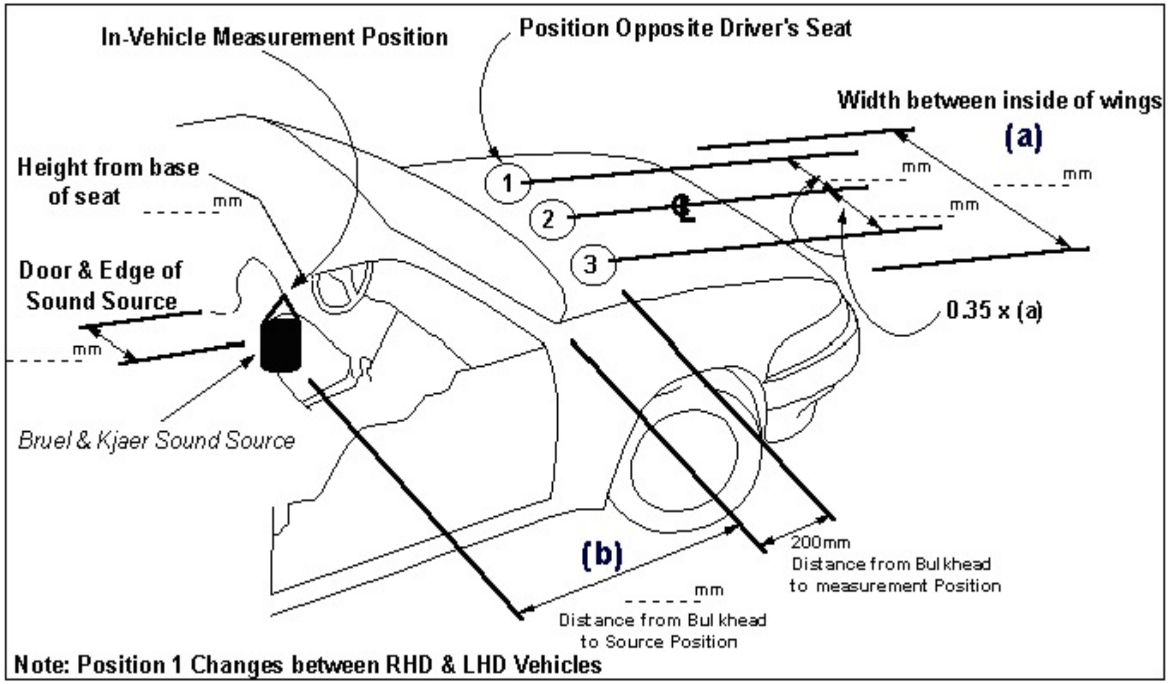
• Interior intensity measurements -> **body transparency**

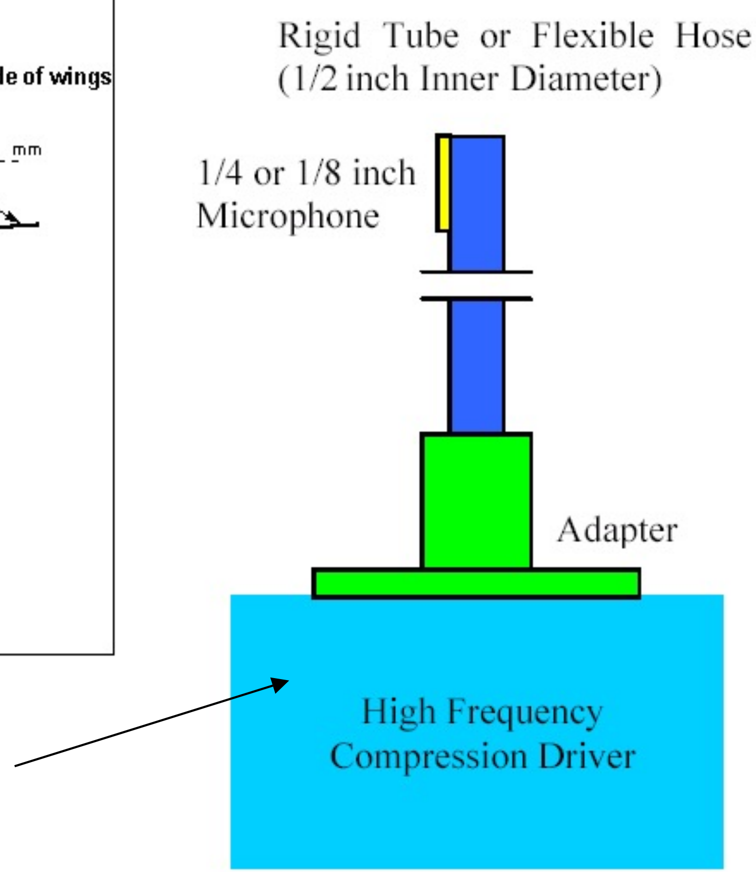
• P/P transfer functions at the tire patch assess the degree of airborne road noise: TPA

• Point sources placed within trunk/engine compartment give clear direction for passthru opportunities

• Interior absorption measurements

16.3 p/p Measurement





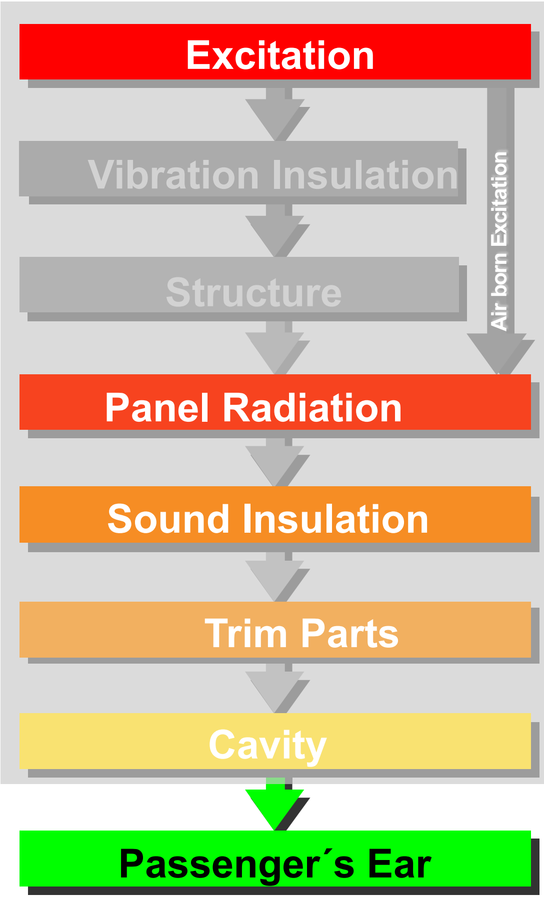
Power Based Noise Reduction technique :

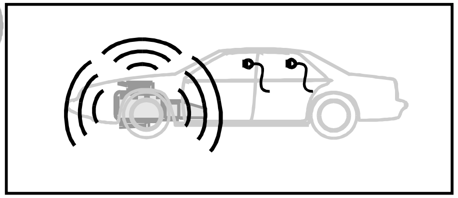
$$PBNR = -20\log_{10}|p/p_s| + Ref$$

$$Ref = 20\log_{10}|p_f/p_s| + 10\log_{10}(4\pi R^2) + 10\log_{10}[400/(\rho c)]$$

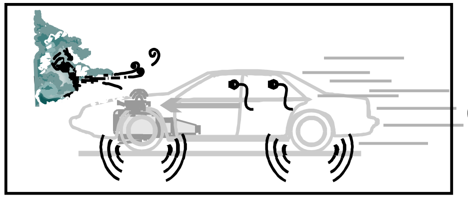
16.4 Combustion/Roadnoise Analysis

SOUND PRESSURE LEVEL in dB(A)
and
(OPEN) ARTICULATION INDEX in %
driver´s and rear right passenger´s outer ear

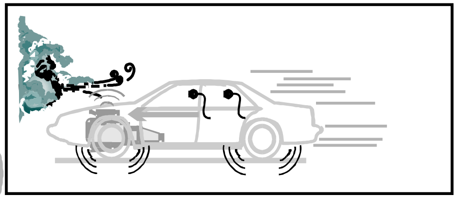




Engine Noise
measured during slow acceleration in gear 2
(average of A-weighted sound pressure level
between 2500 - 5000 rpm;
road measurement)

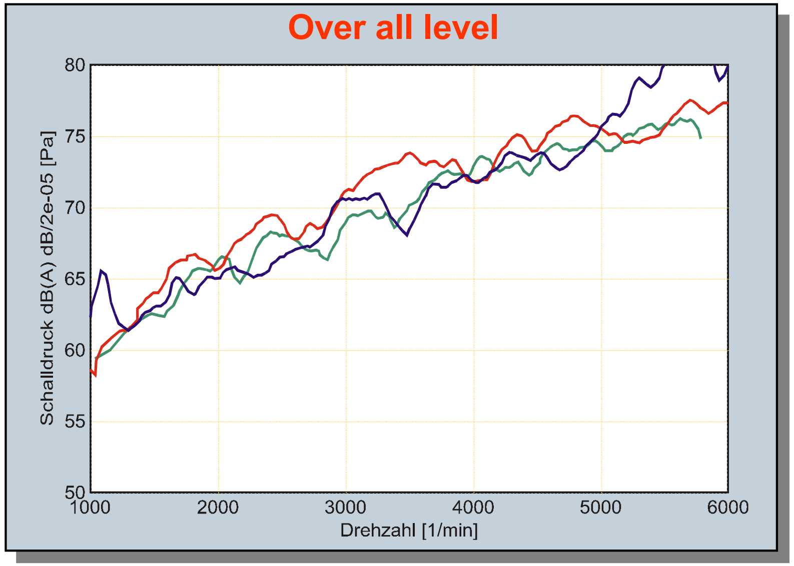


Wind- & Tire Noise
measured during konstant speed and rolling
(average of A-weighted sound pressure level
between 70 - 120 kph;
road measurement



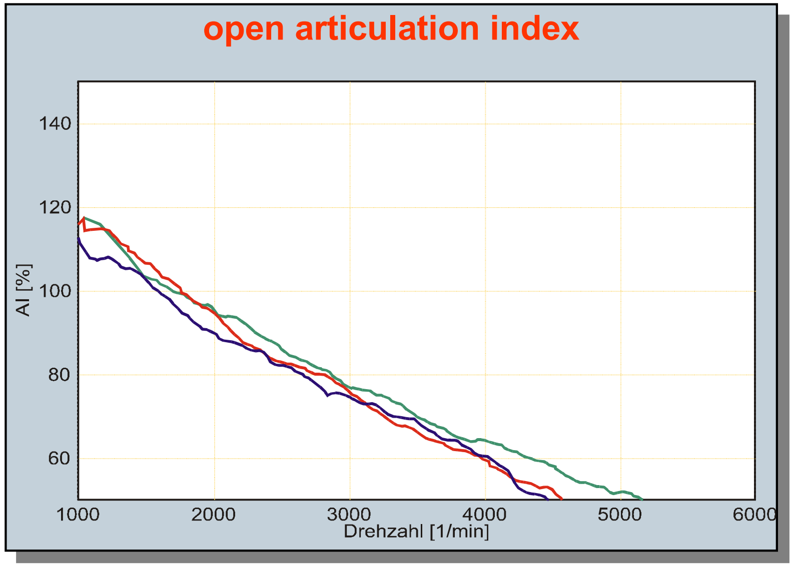
Overall Noise
measured during fast acceleration in gear 3
(average of A-weighted sound pressure level
between 2500 - 5000 rpm;
road measurement)

16.5 Acoustic Results Road Measurements

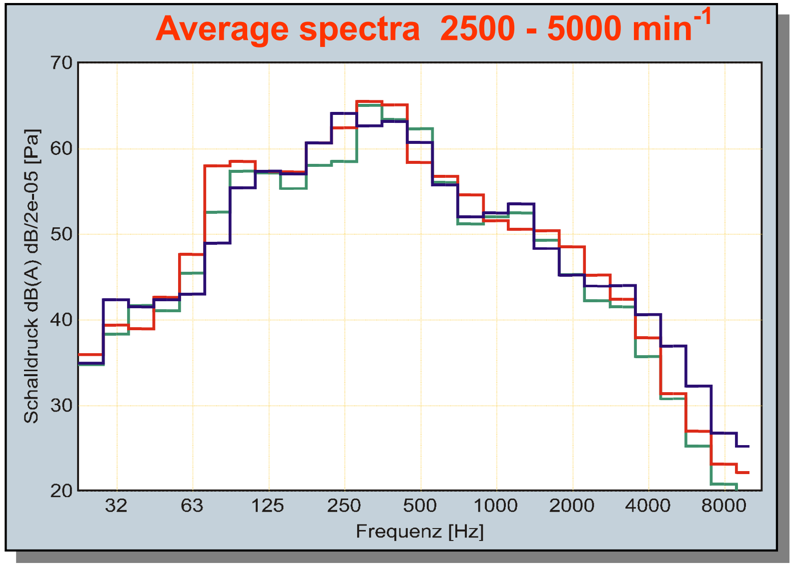


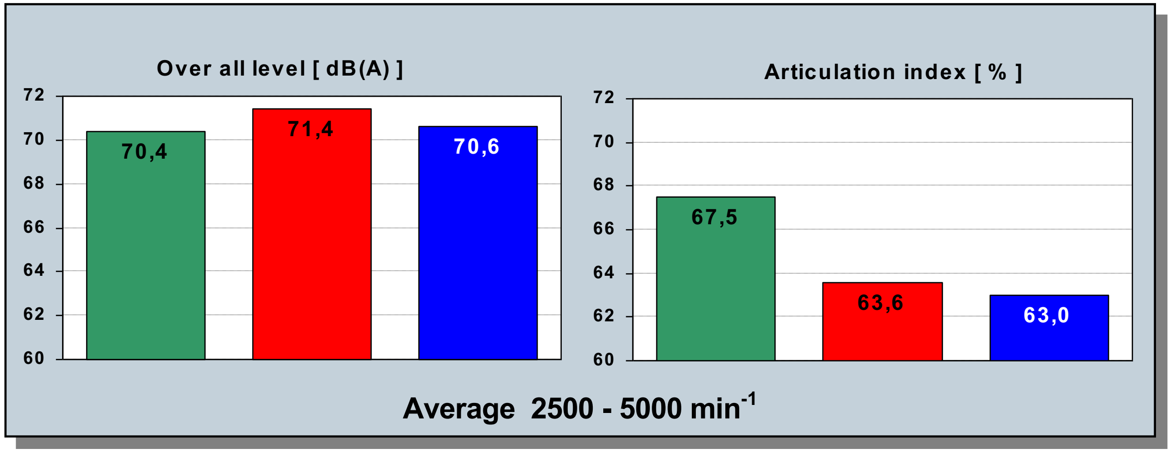
GEAR 3
WOT
acceleration

Driver's ear outer

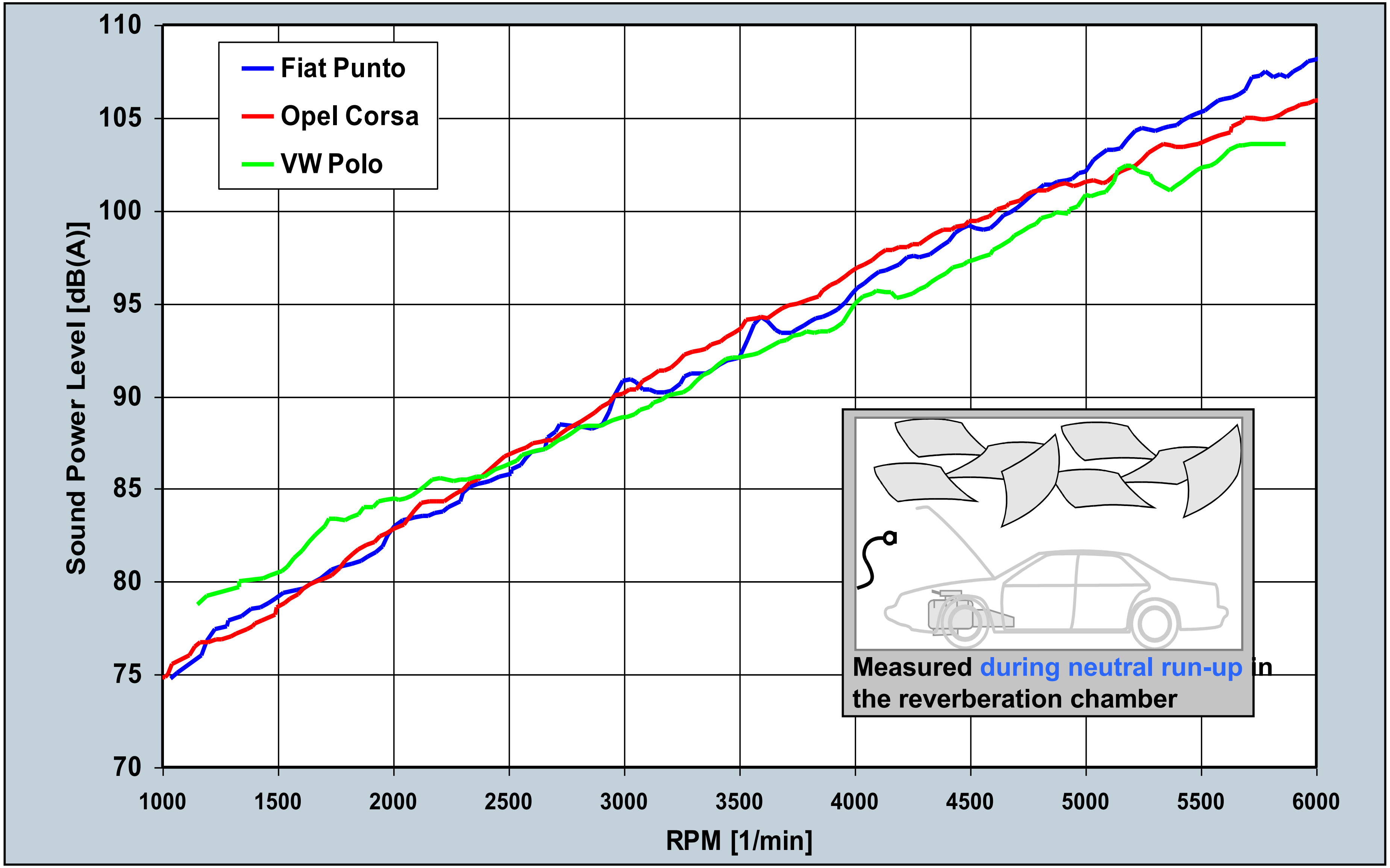


- VW Polo 1,2-12V
- Opel Corsa 1,0-12V
- Fiat Punto 1,2-8V

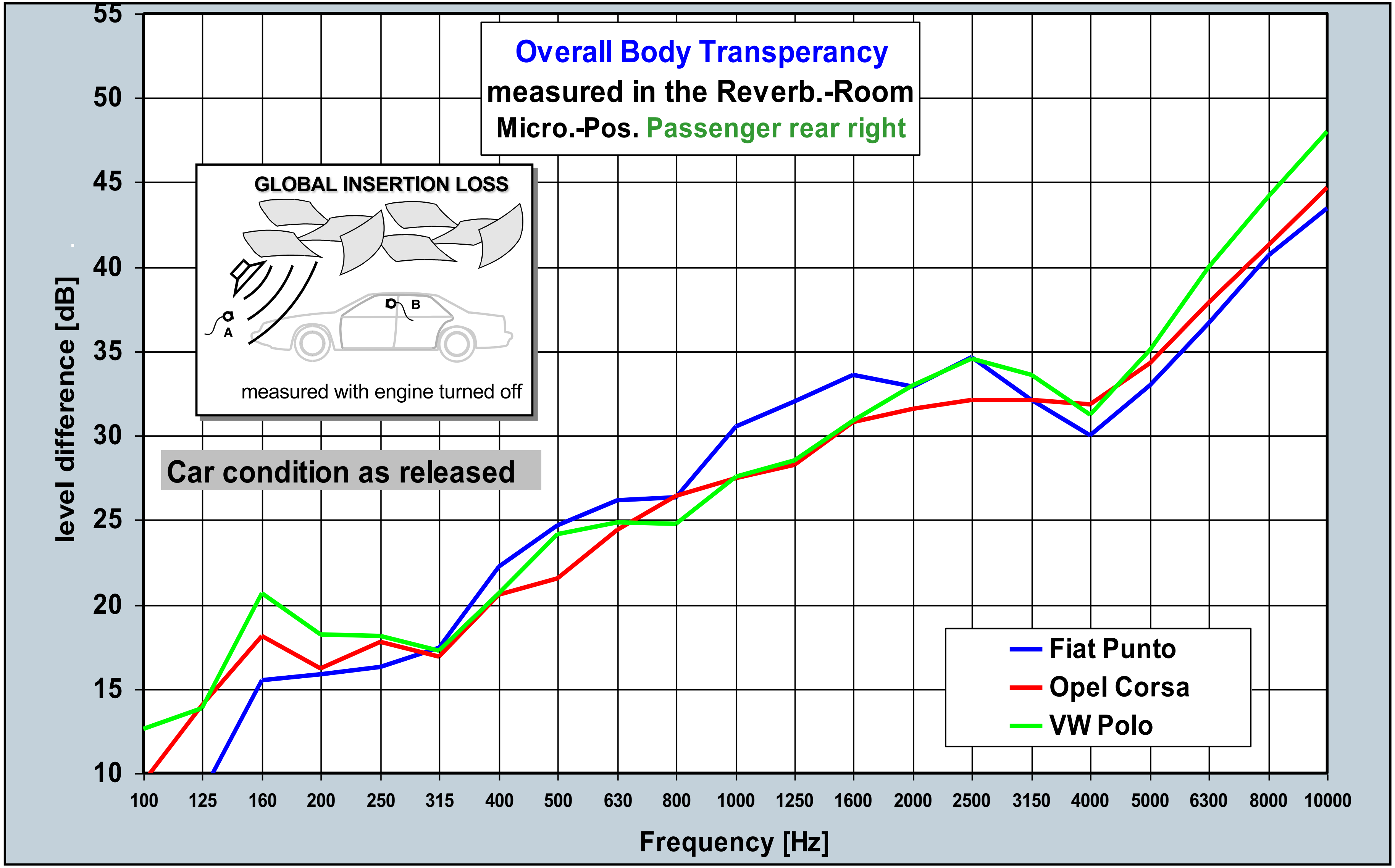




16.6 Acoustic Results Sound Power



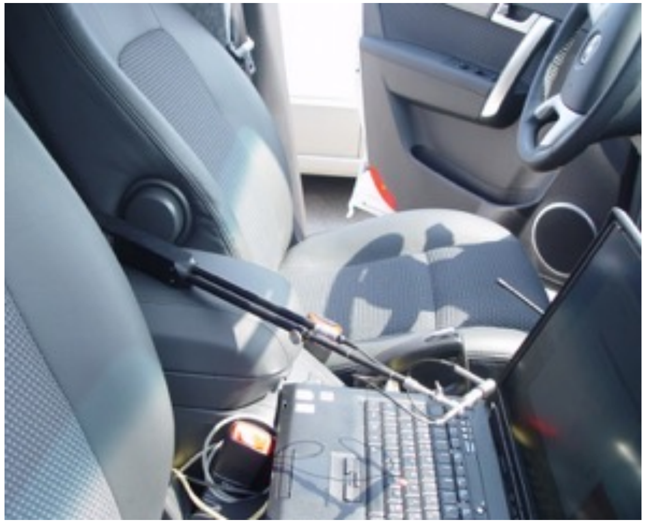
16.7 Acoustic Results Body Transparency



16.8 Noise Source Identification & Ranking

1. - Sound intensity measures according to ISO 9614-1,2,3 for 16 panels at **80kph, 3rd gear**
2. Measurement of the panel to driver´s ear FRF in the hemi-anechoic room
3. Calculation of the weigthed sound power contribution for every panel at the driver´s ear
4. Ranking of the different panels as measured

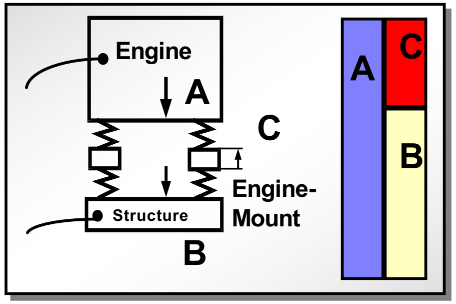
#	Panel
1	A-Pillar
2	B-Pillar
3	Centerconsole
4	Centertunnel
5	Centertunnel rear
6	Fascia right
7	Front door right
8	Front door right glass
9	Front floor right
10	Front tunnel right
11	Headliner front
12	IP right middle
13	IP upper right
14	Left fascia
15	Rear floor panel
16	Right runnel lower

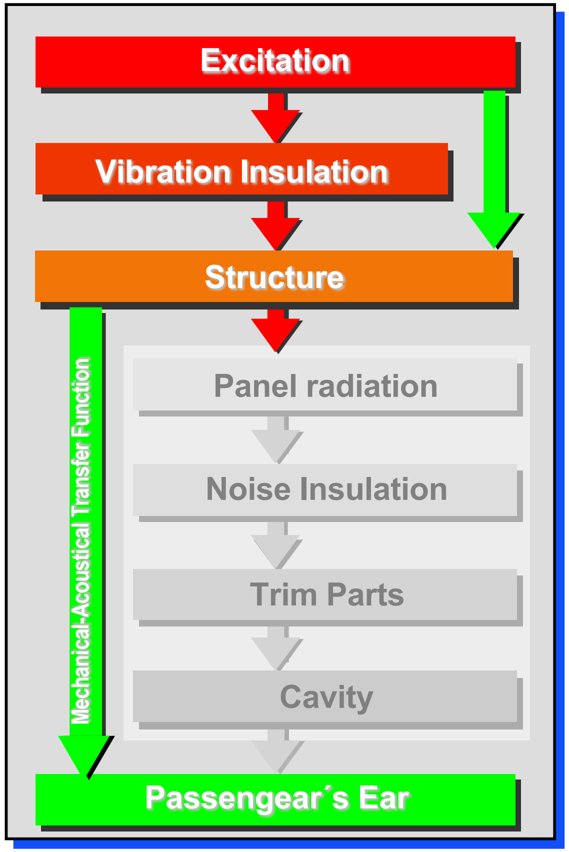


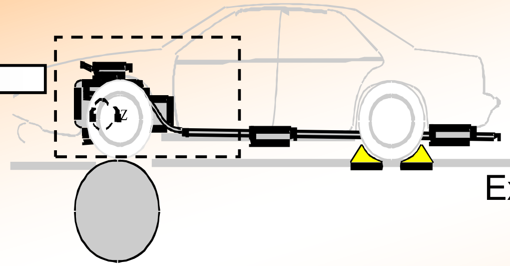
Intensity probe used

Vehicle rear prepared with absorber felt to reduce reactive intensity

16.9 Structure Borne Noise analysis

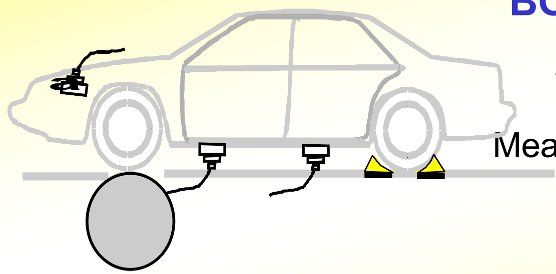






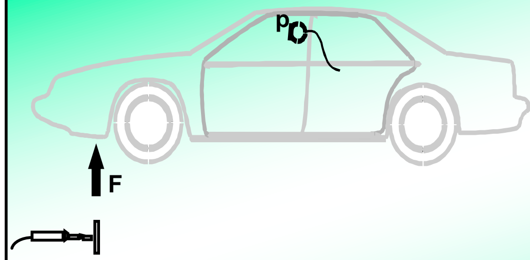
ENGINE MOUNT INSULATION

Acceleration Level in front of and behind the engine mounts.
Excitation with fast acceleration in gear 3 (averaged 2500 - 5000 min^{-1})



BODY ACCELERATION POINTS

Accelerationn Levels at the Body attachment points.
Measured with fast acceleration in gear 3 (averaged 2500 - 5000 min^{-1})



MECHANICAL - ACOUSTICAL TRANSFER FUNCTION

$$MATF = \frac{\text{Sound Pressure}}{\text{Force}}$$

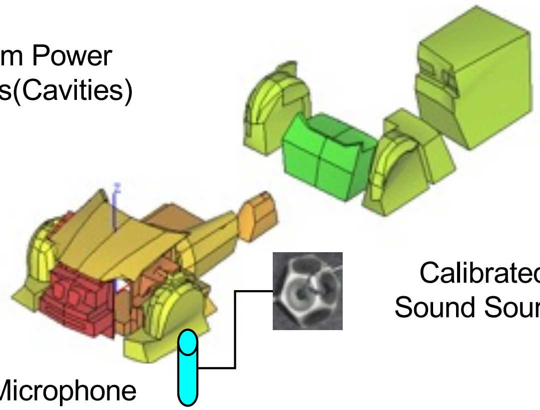
Measured with engine turned off
(Excitation with Impact Hammer)

16.10 Extracting the D.N.A. dataset

1. SPL Measurement with Calibrated Sound Source

System Power Sources(Cavities)

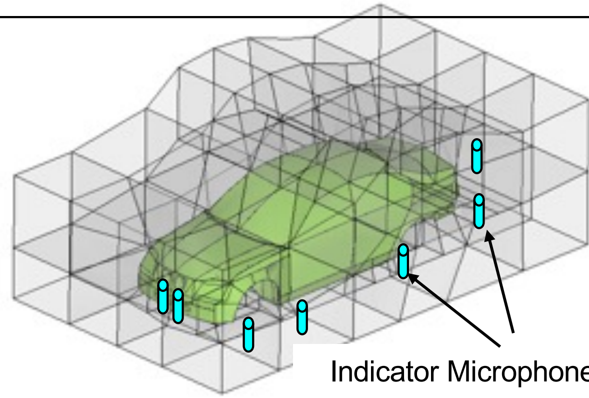
Using Power Substitution techniques a set of indicator microphones are created that will measure sound power directly.



Calibrated Sound Source

Indicator Microphone

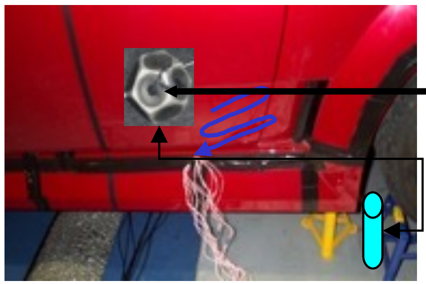
2. Operating SPL close to Noise Generating Sources



1. and 2. Combine to produce the operating sound powers of each component.

Indicator Microphones

3.Measurement of Transfer Functions (serial & parallel)



Calibrated Sound Source

Spatial Transfer Function

4. RT60 Measurements inside Cavities

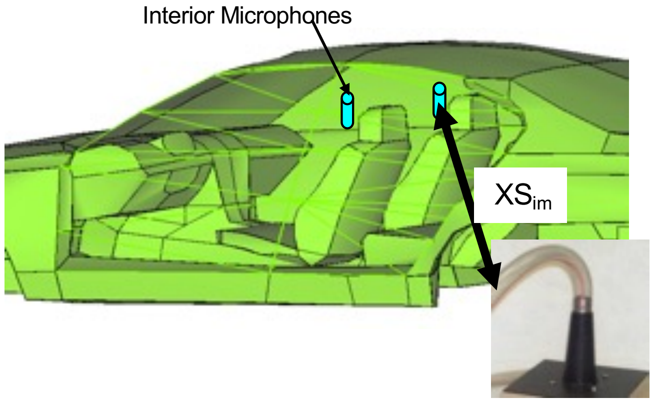
DLF for SEA Cavities

$$\eta = \frac{\ln(10^6)}{\omega T}$$

The complete procedure for extracting Acoustic DNA is shown above. Operating sound powers from each component are established directly in-situ and noise transfer is also split into serial and parallel paths.

16.11 Serial Transfer Functions XS

XS is a Serial Transfer Function



Transfer function from key noise generating sources i.e. tyre contact patch into cabin / boot

High power tube source placed at key noise generating areas, such as tyre contact , exhaust orifice, differential etc.

XS_{im} = FRF from noise generating **Source** (i.e. tyres) directly into cabin

with: i = 1..N; N = Number of sound Sources

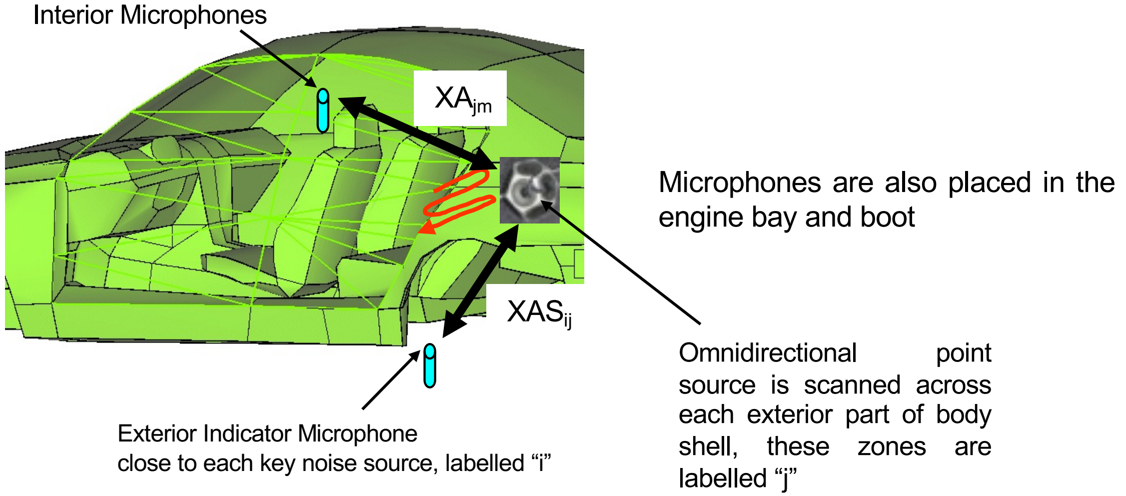
m = 1..4; Number of interior microphones

FRF = Frequency response function

Splitting up the transmission of noise from the key noise generating sources into serial and parallel transfer functions, enables the detailed study of which areas of the body shell are critical to noise insulation. When comparing with the competition it is the serial transfer function that determines whether the target has been achieved, but it is the composite parallel transfer functions that define the necessary acoustic treatment solution.
The resultant levels of noise inside the cabin during operation are, of course, determined by the combination of actual operating noise source generators and the serial transfer functions. However, studying the effect of the body shell does enable the possible cause of interior noise to be highlighted.

16.12 Parallel Transfer Functions XA&XAS

<u>**XA & Xas are Parallel Transfer Functions**</u>



XA_{jm} = FRF from external scan **Area** into **Cabin**

XAS_{ij} = FRF from external scan **Area** to noise generating **Source**

with: i = 1..N; N = Number of sound Sources

j = 1..M; M = Number of scan Areas

m = 1..4; Number of interior microphones

FRF = Frequency response function

16.13 Composite Transfer Functions XC

XS_{im} = FRF from noise generating **Source** (i.e. tyres) directly into cab

XA_{jm} = FRF from external scan **Area** into **Cabin**

XAS_{ij} = FRF from external scan **Area to** noise generating **Source**

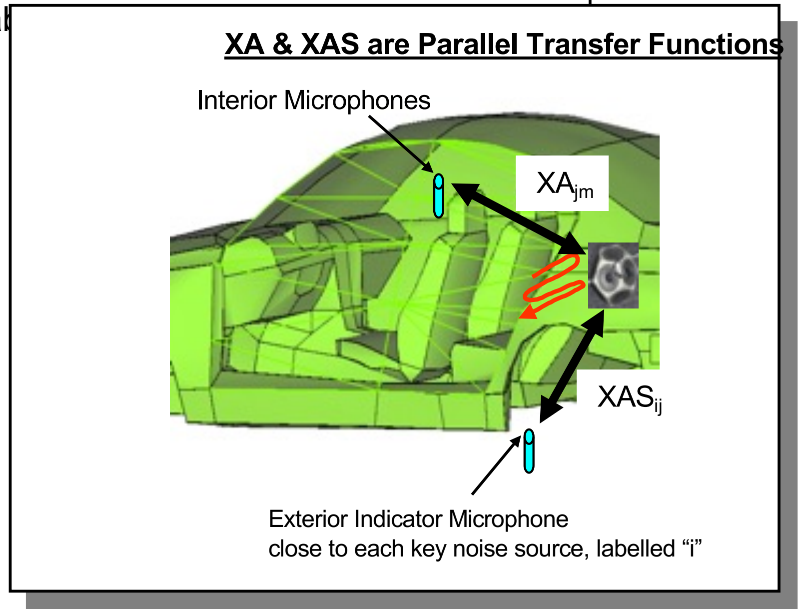
XC_{ijm} = **Composite** FRF = XA_{im}*XAS_{ij}

with: i = 1..N; N = Number of sound Sources

j = 1..M; M = Number of scan Areas

m = 1..4; Number of interior microphones

FRF = Frequency response function



Out of the area scans the composite transfer functions XC_{ijm} will be calculated, leading to a matrix of size NxM FRF´s, e.g.

$$XA_{19;1} * XAS_{21;19} = XC_{25;19;1}$$

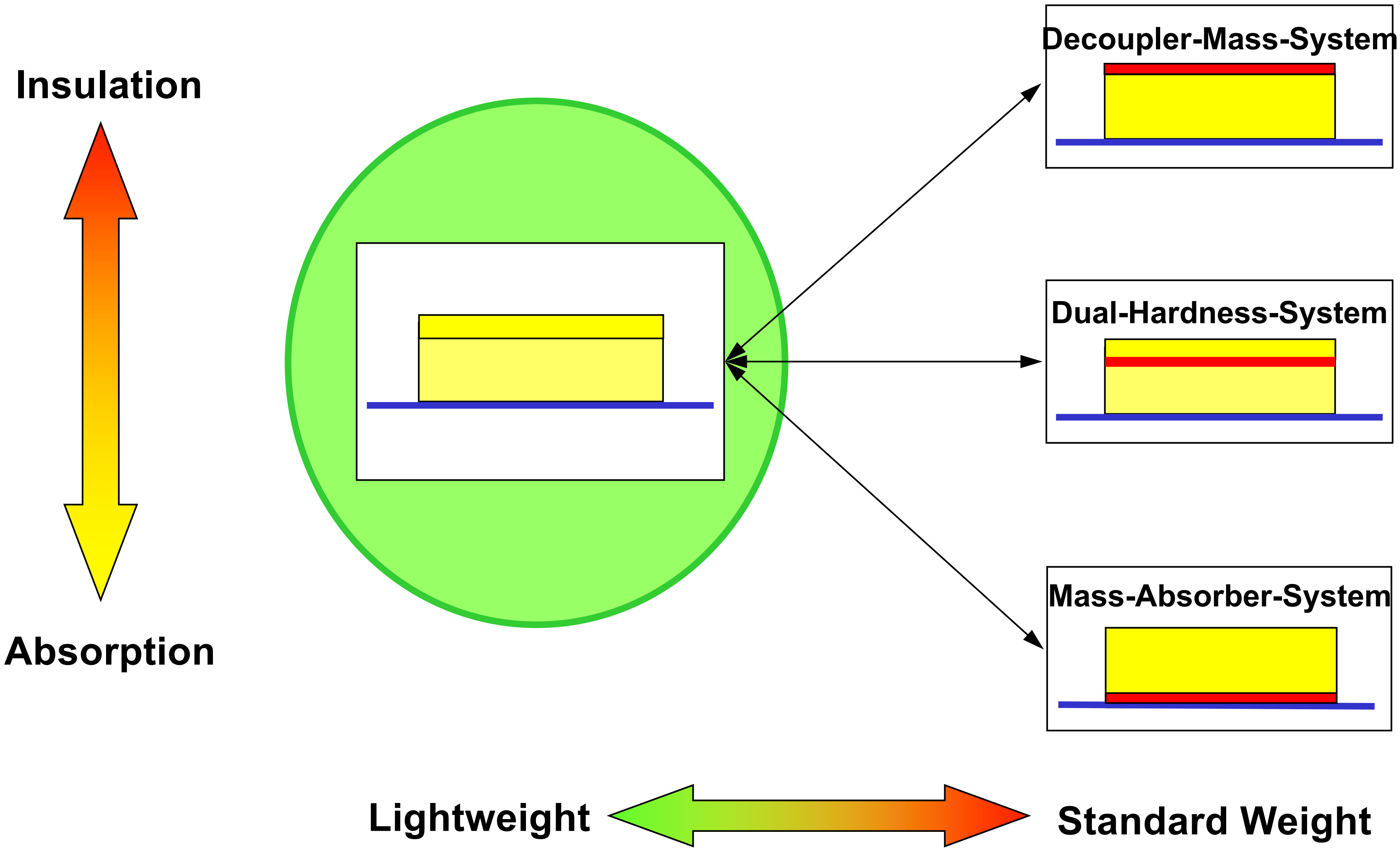
Front wheel tyre patch rear (i=21) -> Floor under front (j=19) -> Driver ear outer (m=1)

Validation: the sum of all composite transfer functions to a certain interior microphone should be equal to the serial transfer function

Agenda:

• Fundamentals of Acoustics	30 min
• Noise & Vibration reduction methods	30 min
Break	
• NVH in vehicles	30 min
• **Main Insulation Concepts**	**30 min**
Break	
• Pexmet & SEA Prediction	30 min
• Q&A	15min

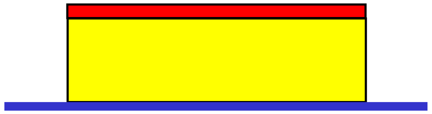
17. The 4 Main Insulation-Concepts



17.1 Comparison of the 4 Main Insulation Systems

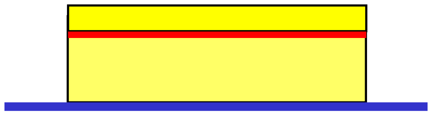
Area weight 3.0 +/- 0.25 kg/m²; total thickness = 20 mm

Decoupler-Mass-System

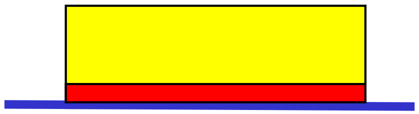


Dual-Impedance-System

Dual-Hardness-System



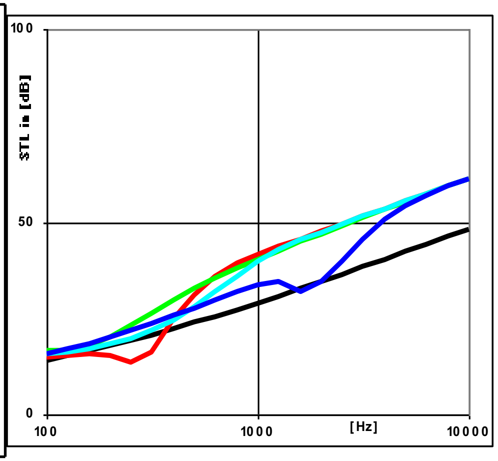
Mass-Absorber-System



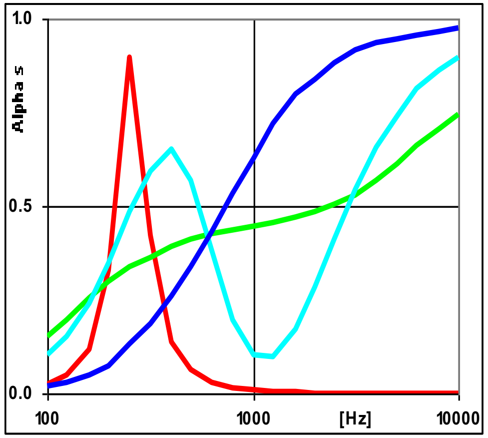
STL, no Leakage

The picture can't be displayed.

STL, 5% Leakage

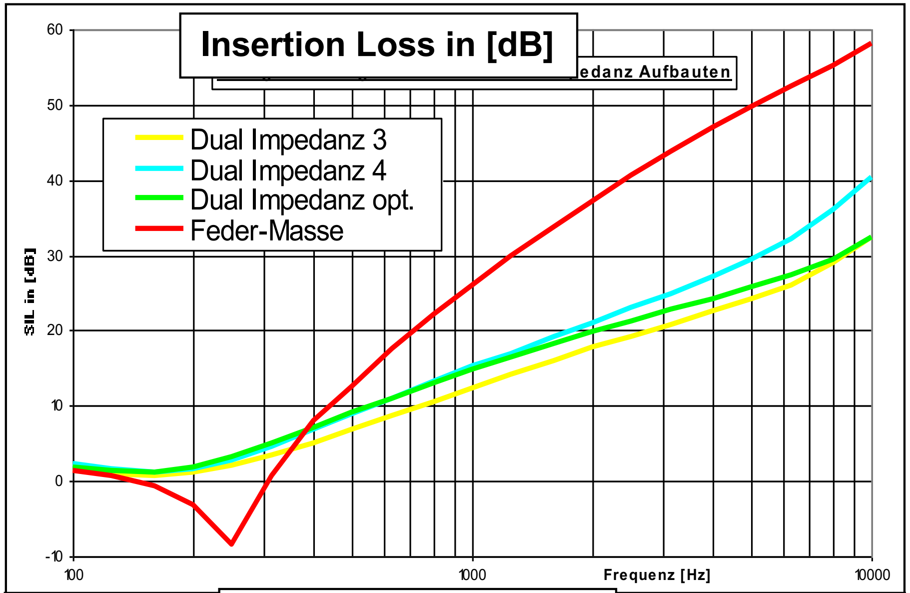


Absorption



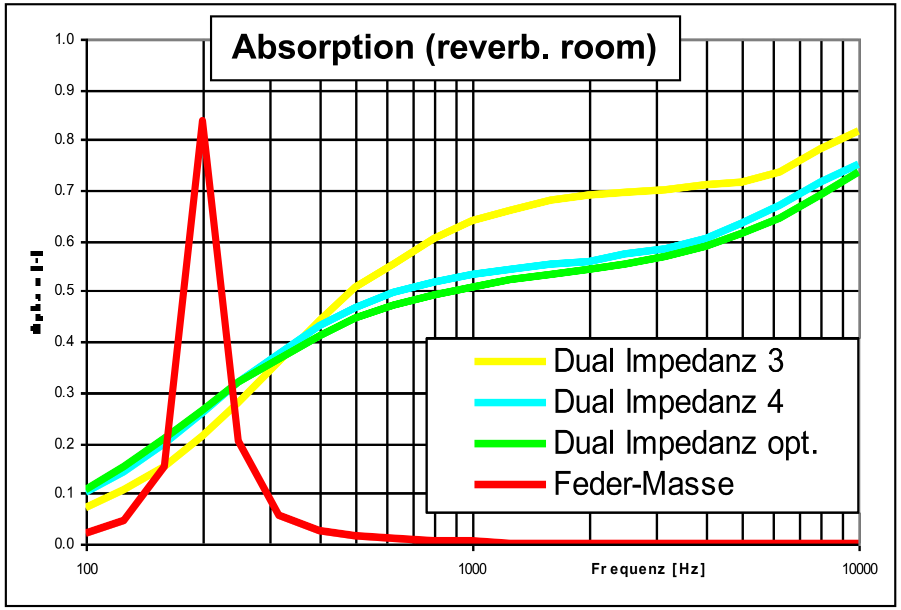
Decoupler-Mass-System	Dual-Impedance-System	Dual-Hardness-System	Mass-Absorber-System

17.2 Insertion Loss & Absorption of Lightweight Ins.



The picture

Insertion Loss in [dB]
with 5 % Leakage

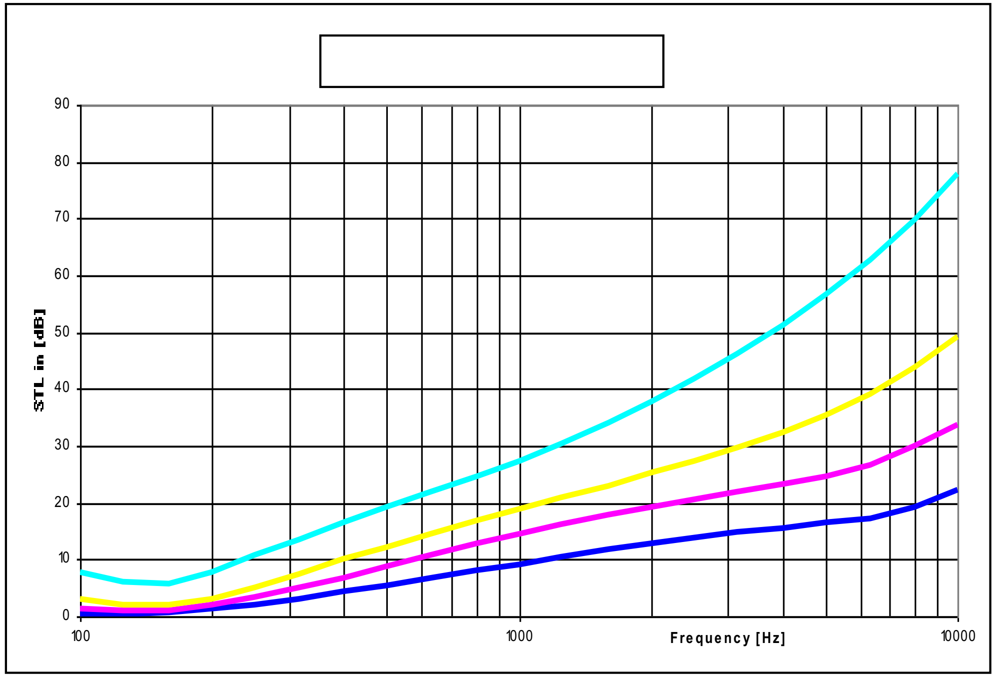


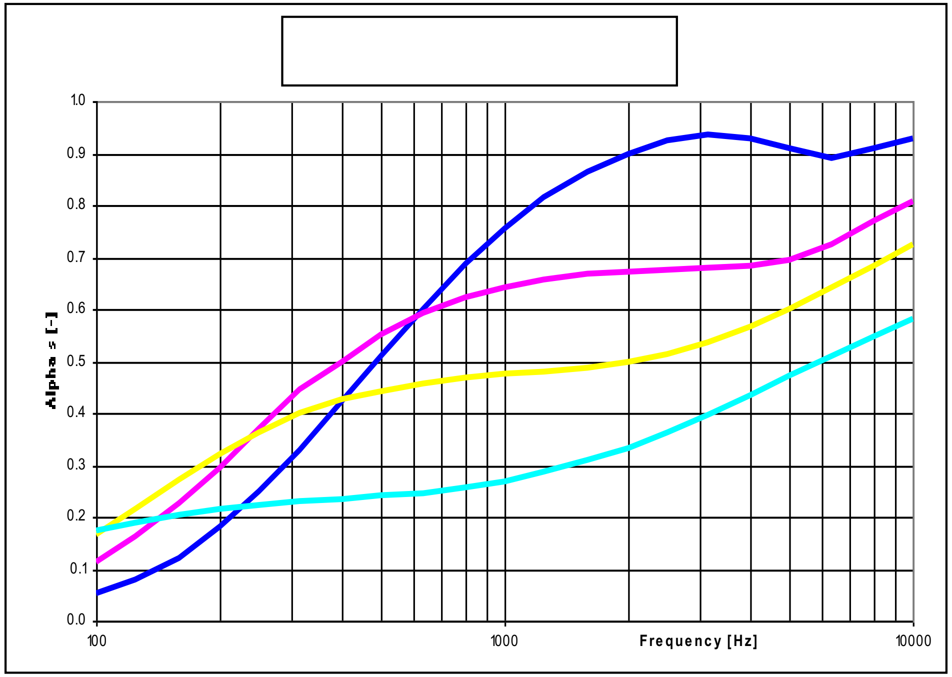
Sample 3	= 2.4 kg/m²
Sample 4	= 3.0 kg/m²
Optimized sample.	**= 2.4 kg/m²**

Increasing area weight and density of the “Hard Layer” and at the same time decreasing area weight and density of the “Soft Layer” leads to further weight reduction.

17.3 Influence of Fiber Diameter

Sound Insertion Loss and Absorption for equal material density



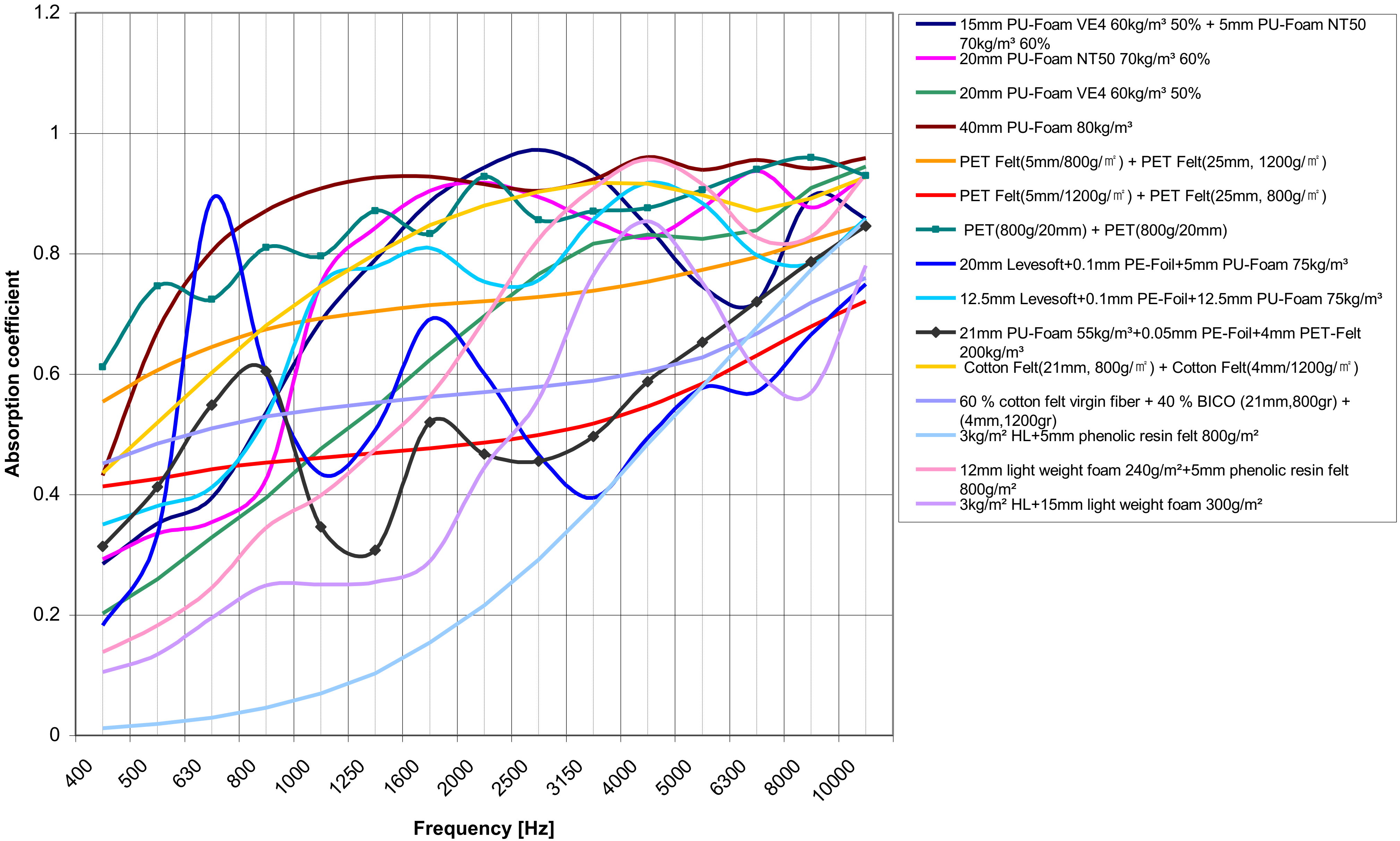


- Micro Fiber
- Fine Fiber
- Cotton Fiber
- Cotton fiber & foam flakes

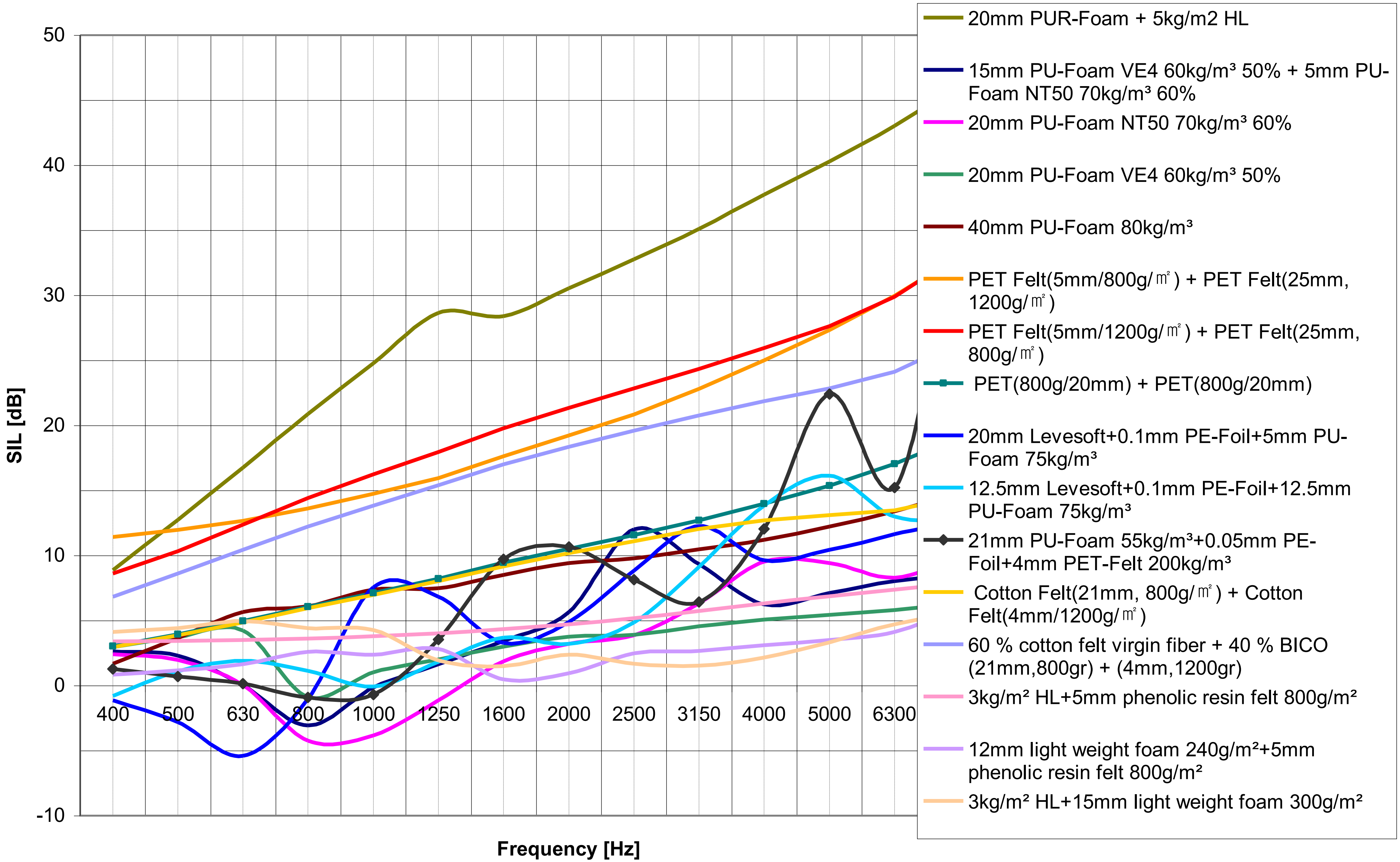
A smaller fiber diameter leads to a higher air flow resistivity and better Sound Insertion Loss (SIL) or lighter part respectively. The absorption properties are at the same time improved for low frequencies and reduced for high frequencies.

➔ Fiber diameter, layer thickness and density are the key

17.4 Absorption for different materials



17.5 SIL for different materials



Agenda:

• Fundamentals of Acoustics	30 min
• Noise & Vibration Reduction Methods	30 min
Break	
• NVH in Vehicles	30 min
• Main Insulation concepts	30 min
Break	
• **SEA Prediction**	**30 min**
• **Q&A**	**15min**

18. Full Vehicle Modeling: High Frequency

High Frequency

Statistical **E**nergy **A**nalysis

18.1 Setup SEA models

The steps leading to an SEA model of a vehicle which is ready for design studies can be described as follows:

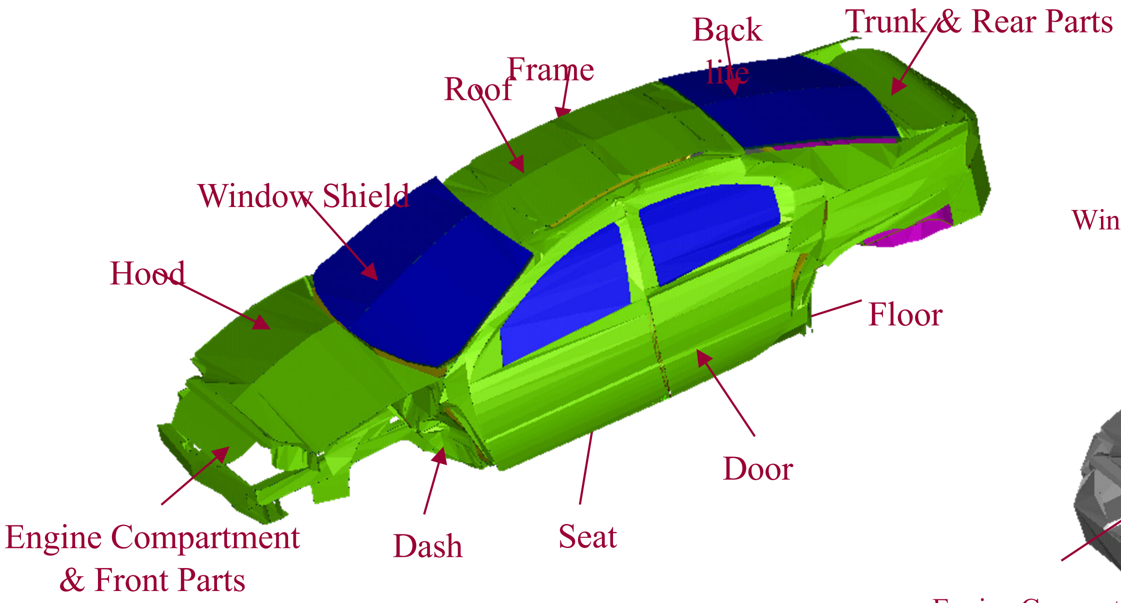
1. Obtain representative geometry (CAD/CAE model or direct measurements) of a surrogate or project vehicle
2. Build the panel and acoustic cavity subsystems
3. Apply trim information
4. Apply load cases
5. Validation data from laboratory (e.g. chassis dyno, rev room..) and on-road tests
6. Validate model (includes corrections or retest)
7. Model declared read for use in design studies

Optionally, if a surrogate model was build:

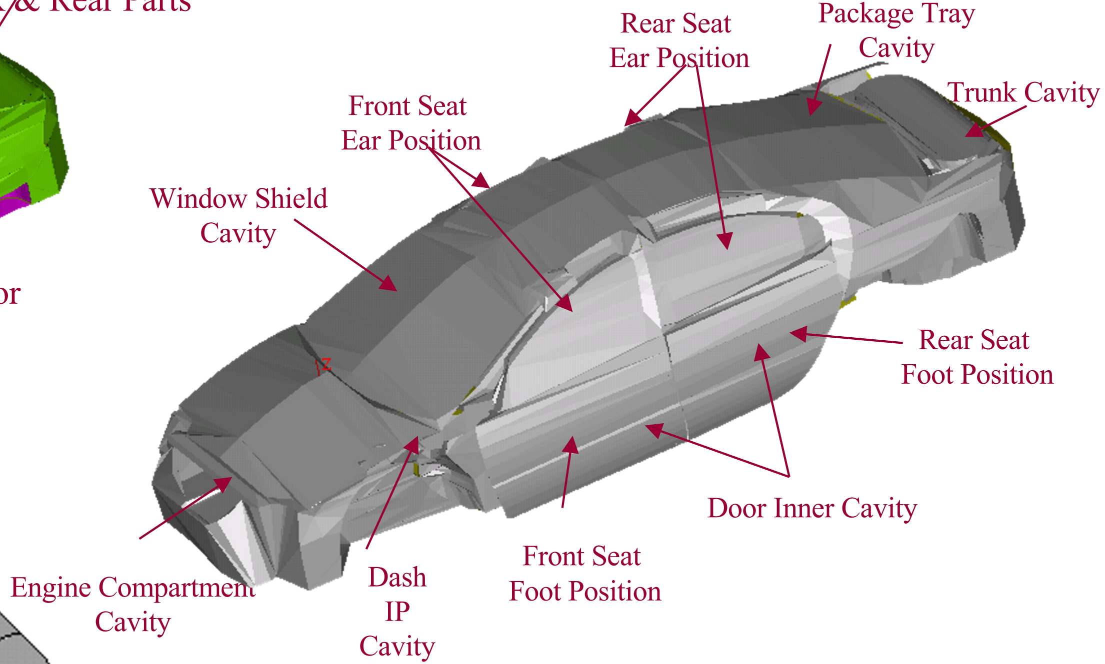
1. Morph the surrogate vehicle geometry to a new geometry

18.2 Build panel/acoustic cavity subsystem

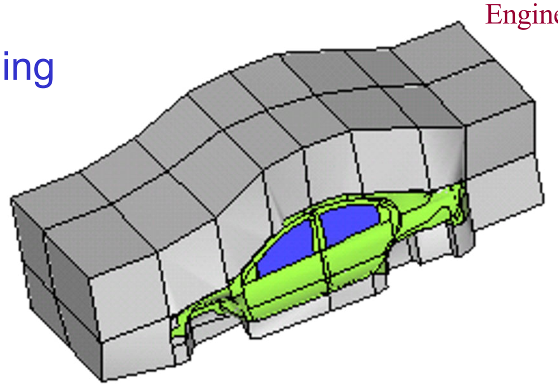
- Subsystems are generated for each component



Structural modeling



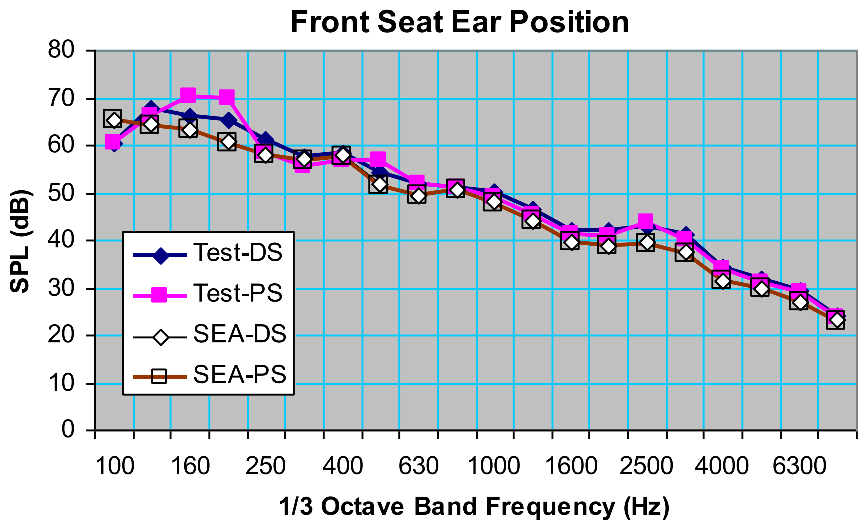
Inner cavity modeling

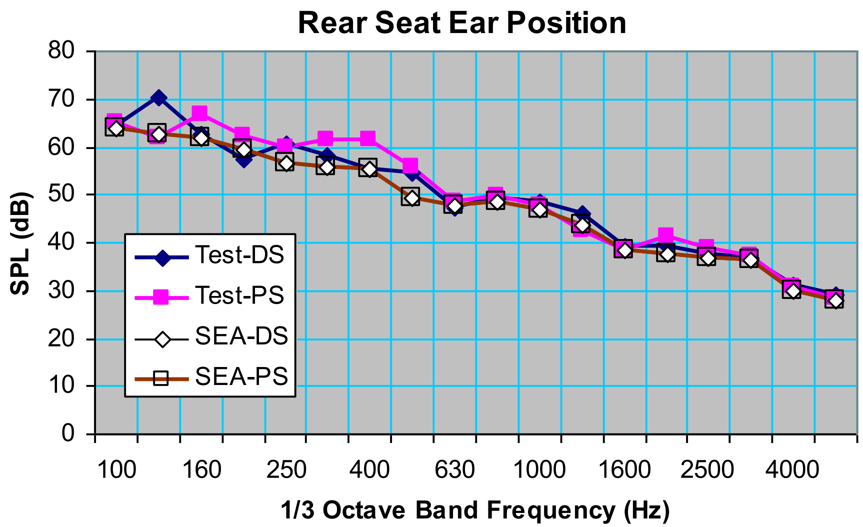


Exterior modeling

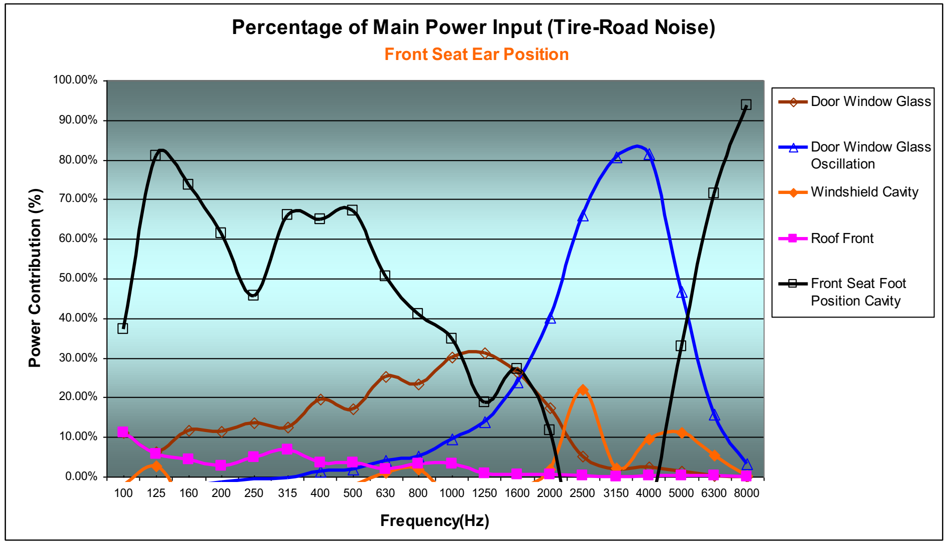
18.3 Validate model

SPL in inner cavity





Power Contribution Analysis



Tire/Road

Engine Noise

19. Full Vehicle Modeling: Low Frequency

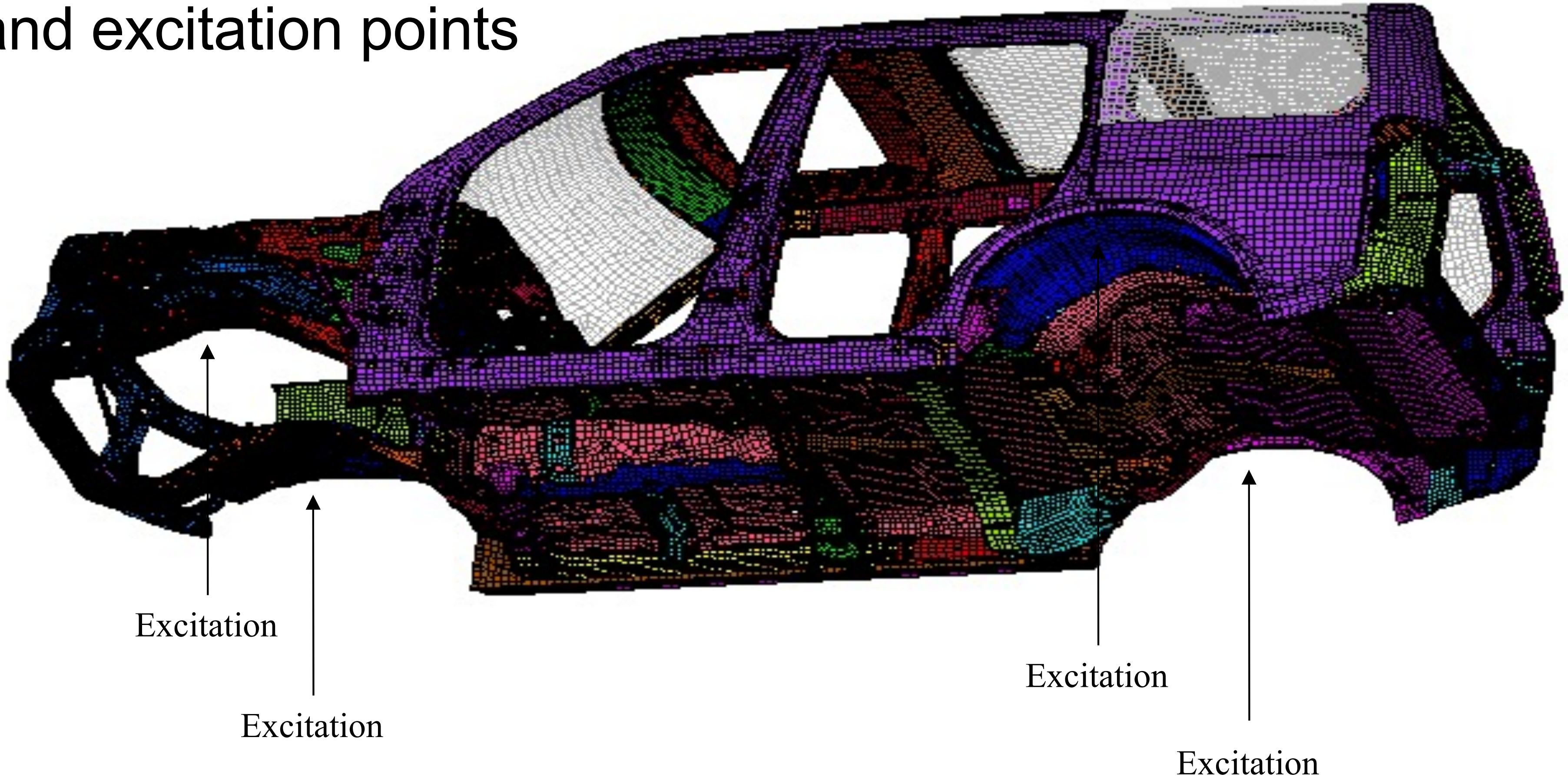
Low Frequency: Boundary **E**nergy **A**nalysis

Panel Acoustic Contibution Analysis : PACA

Near Field Acoustic Holography: NAH

19.1 Boundary Element Model

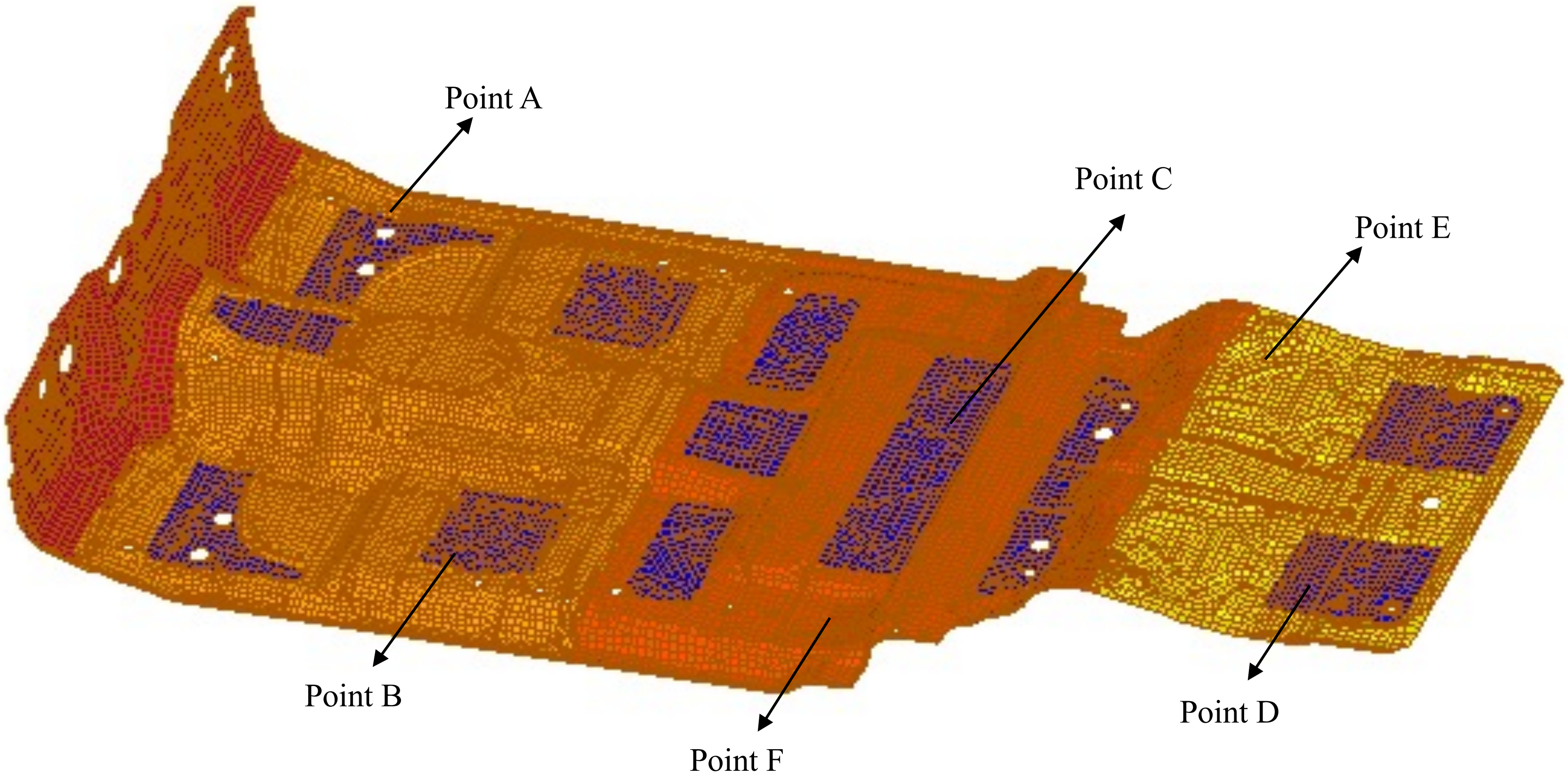
Build BEM model and define load conditions and excitation points



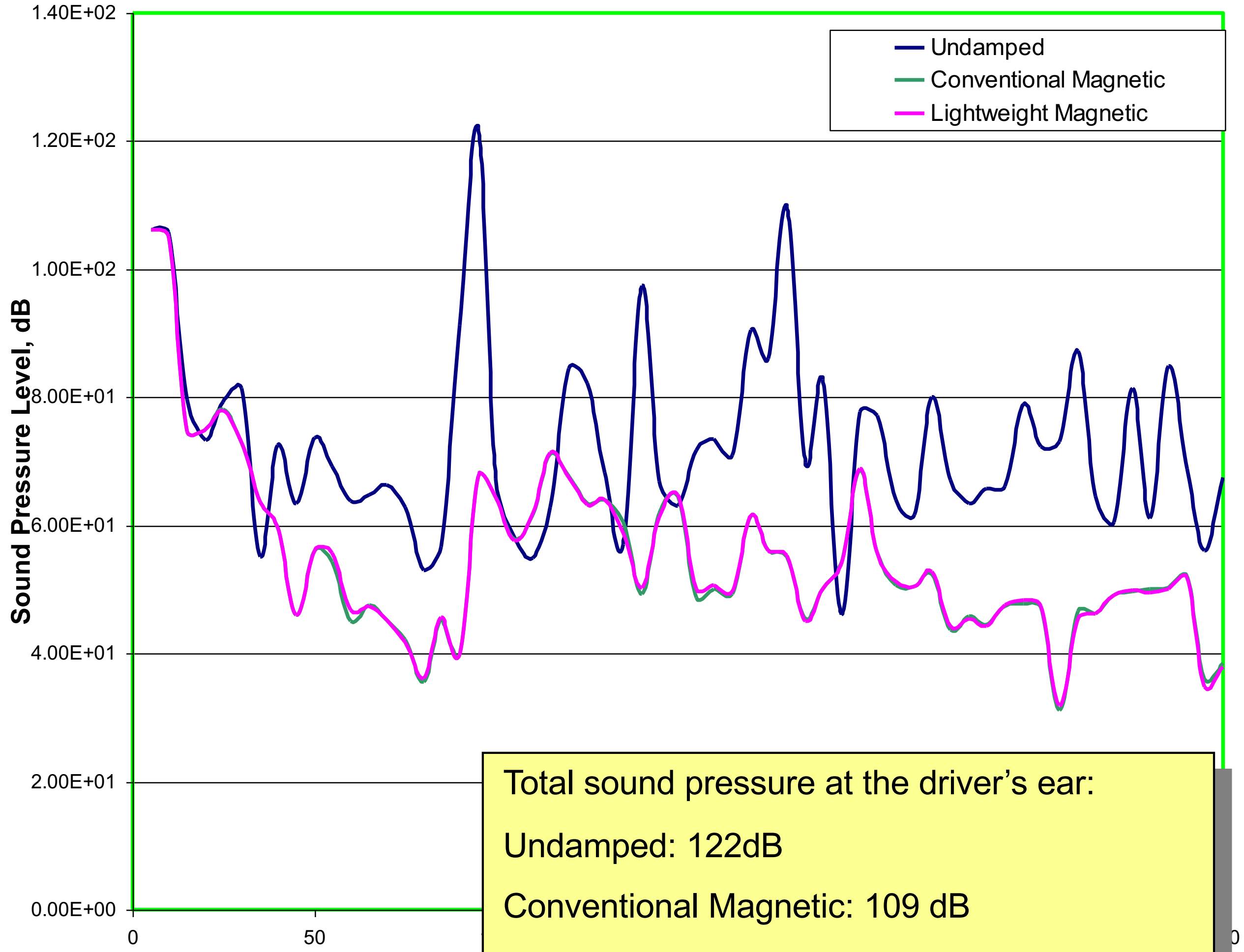
Magnitude of excitations =100 mm/s/s (accelerations)

19.2 Damping Pad Locations

Calculate hot spots and damping pad locations



19.3 SPL Reduction



Total sound pressure at the driver's ear:

Undamped: 122dB

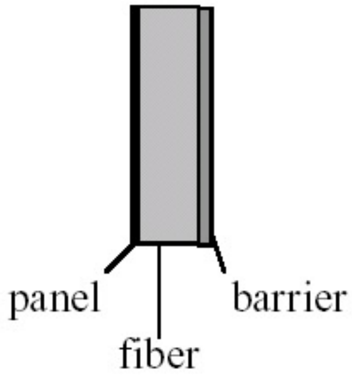
Conventional Magnetic: 109 dB

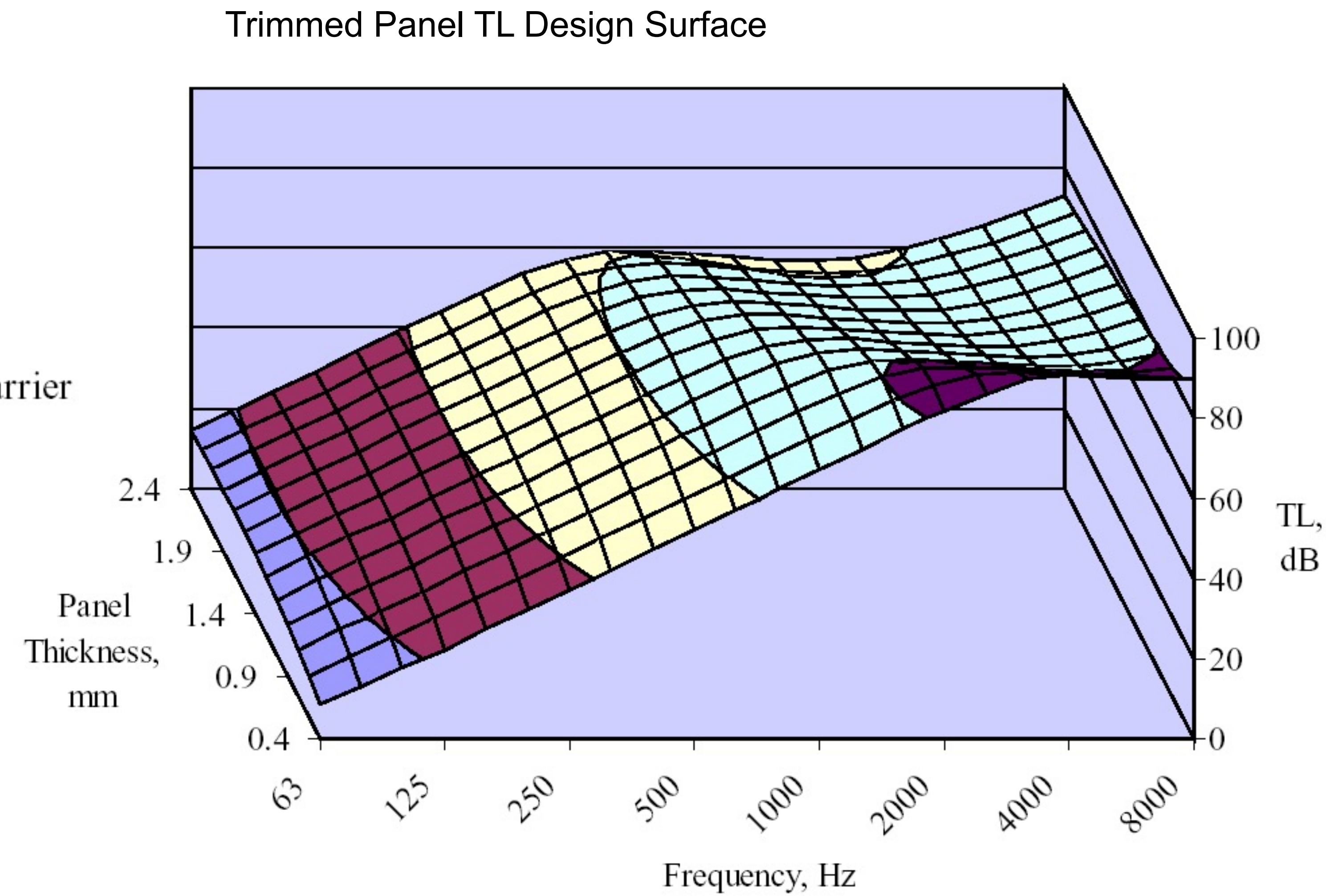
Lightweight Magnetic: 109 dB

20. Sub-System & Component Modeling

High Frequency: **S**tatistical **E**nergy **A**nalysis
Low Frequency: **B**oundary **E**nergy **A**nalysis
Finite **E**lement **A**nalysis

20.1 SEA: Trimmed Dash Panel

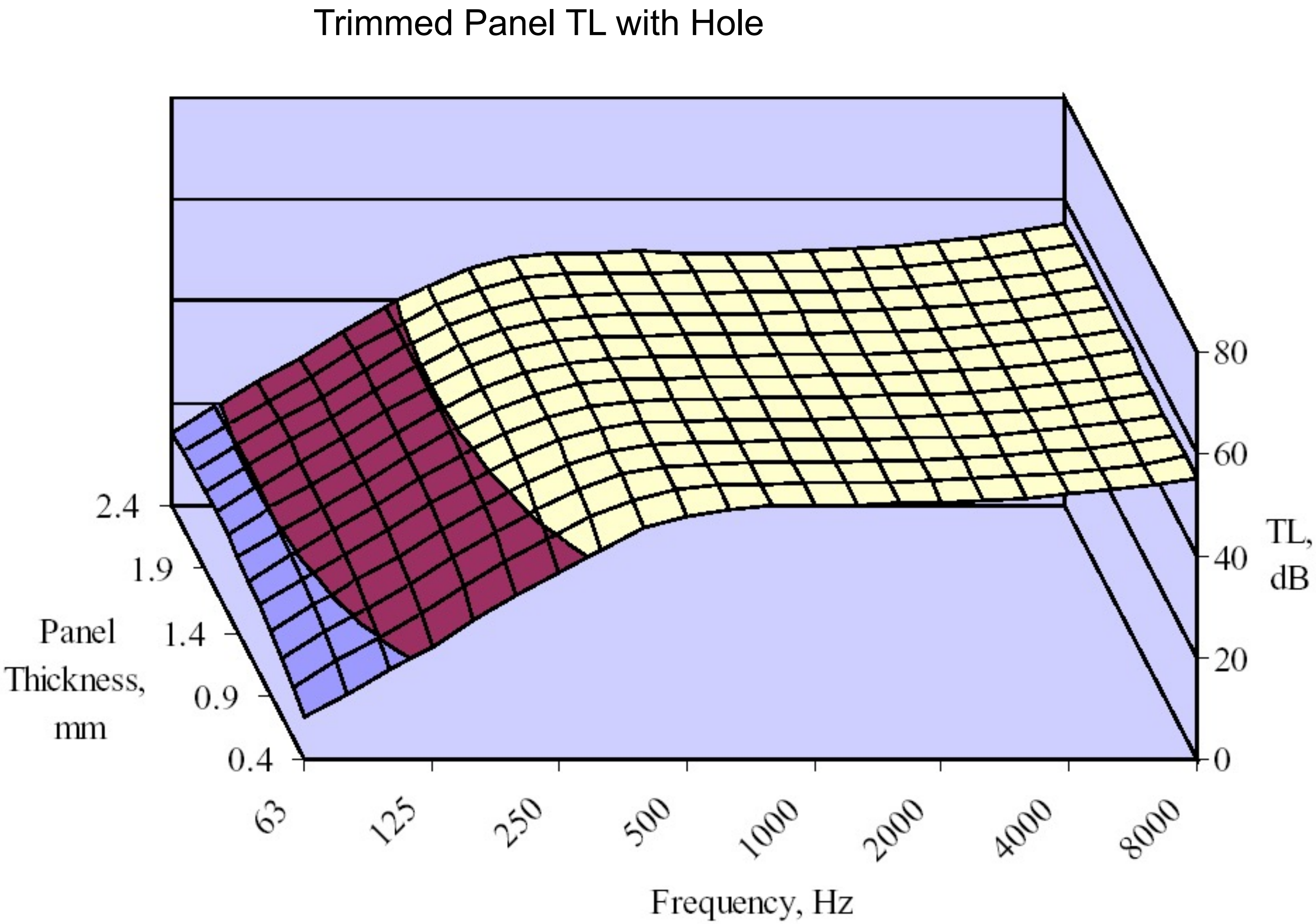




Question 1: Is there a better design with the same weigth which max. the TL ?

Question 2: Is there a difference in the optimum design if low or high frequency sounds are preferred from a sound quality pount of view ?

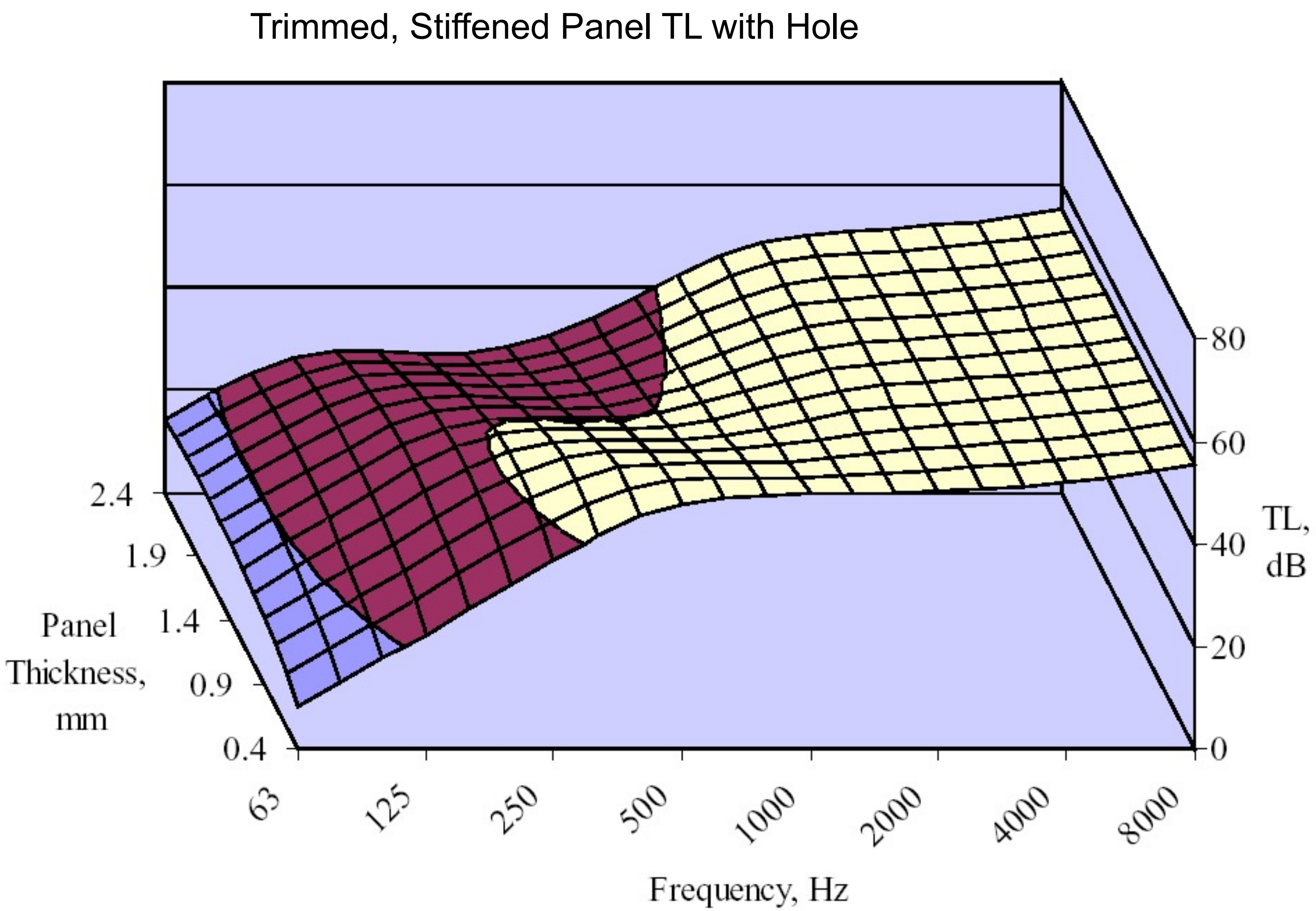
20.2 SEA: Trimmed Dash Panel



The hole flattens of the TL at high frequencys,

However, the benefit of the thicker panel remains at lower frequencys !

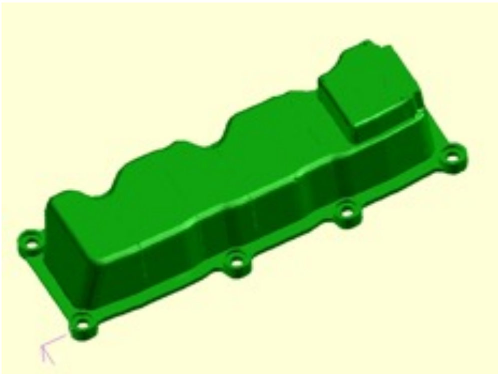
20.3 SEA: Trimmed Dash Panel

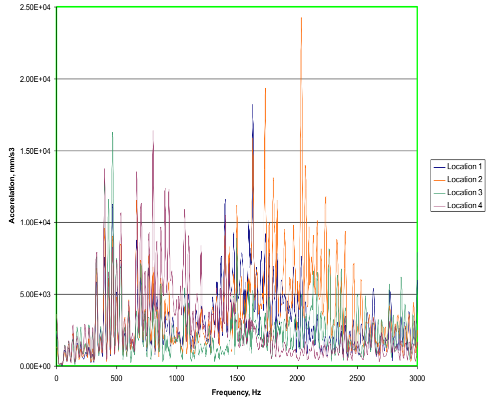


With the stiffened panel the coincidence dip in the TL moves down into the 500Hz range for the larger panel thickness.

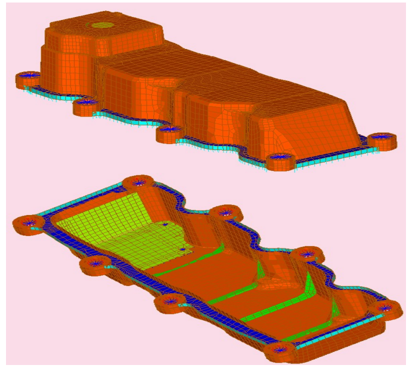
20.4 Component Analysis

- **Comparison of acoustic performance of magnesium, plastic engine cover**



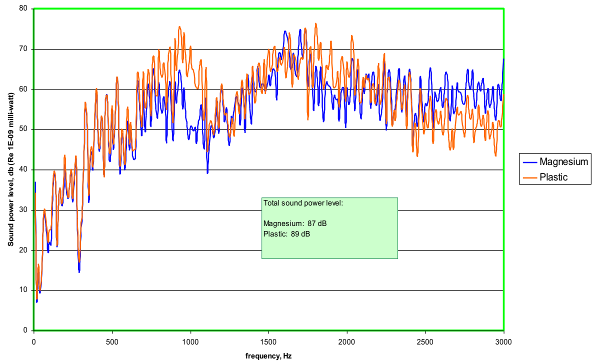


Measured engine excitation

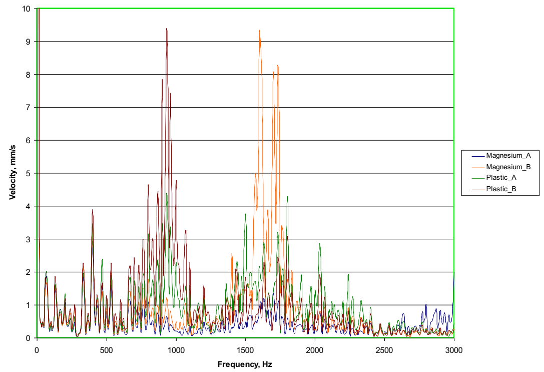


Finite element mesh

Total radiated sound power
- Plastic, 89 dB
- Magnesium, 87 dB



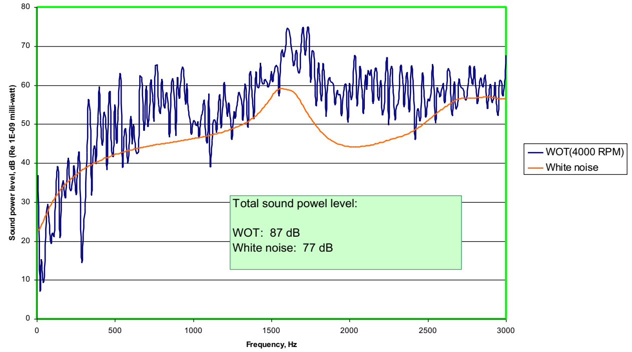
Acoustic Response



Structural Response

20.5 Design Iteration Process

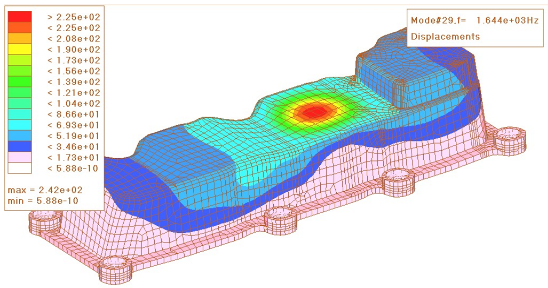
- Reduce radiated power at peak frequencies
  - Identify peak frequencies
  - Compute sound power based on white noise
- Associate the structural modes and peak frequencies

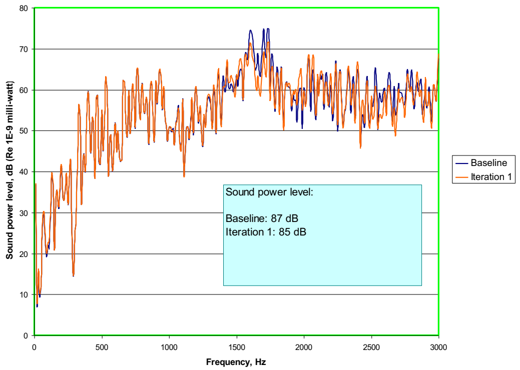


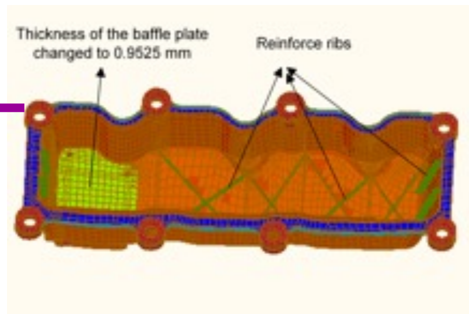
Noise peaks at 1600Hz, 2800Hz

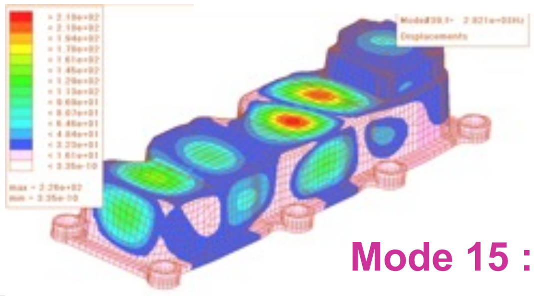
Baseline : 87dB
New design: 85dB

Mode 5: 1640 Hz









Mode 15 : 2820 Hz

Table of contents: