DELIVERABLE 4.3.1

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ACRONYM	**CITYHUSH**		
TITLE	**Acoustically Green Road Vehicles and City Areas**		
Work Package 4	**Propagation attenuation of road traffic noise**		
4.3	**Reduction of low frequency airborne noise**		
	Concept for high low frequency absorption at building facades		
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PU	Public	✓
PP	Restricted to other programme participants (including the Commission Services)	
RE	Restricted to a group specified by the consortium (including the Commission Services)	
CO	Confidential, only for the members of the consortium (including the Commission Services)	

Nature of Deliverable

R	Report	✓
P	Prototype	
D	Demonstrator	
O	Other	

SEVENTH FRAMEWORK PROGRAMME

This deliverable has been quality checked and approved by CityHush Coordinator
Nils-Åke Nilsson

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0 EXECUTIVE SUMMARY

0.1 OBJECTIVE OF THE DELIVERABLE

Designing solutions to reduce low frequency airborne noise (such as e.g. noise from exhaust system of buses or from idling diesel engines) by measures at the façade walls (increased absorption).

0.2 DESCRIPTION OF THE WORK PERFORMED SINCE THE BEGINNING OF THE PROJECT

- Determination of the source spectra of buses and trucks through a measurement campaign near a bus stop in a city centre;
- Definition of criteria for low frequency indoor noise;
- Calculation of the indoor noise levels that occur when a bus is idling at a bus stop or when a bus is departing from a bus stop by means of a noise model;
- Determination of the characteristics and dimensions of two low frequency absorbers: a solid panel absorber and a perforated panel absorber;
- Defining applications for the two low frequency absorbers at façade walls.

0.3 MAIN RESULTS ACHIEVED SO FAR

The determination of the characteristics and dimensions of two low frequency absorbers: a solid panel absorber and a perforated panel absorber.

0.4 EXPECTED FINAL RESULTS

In the next stage of the project, the designed low frequency absorbers will be tested thoroughly in the lab and in practice and will be fine-tuned. The final characteristics and dimensions of the low frequency noise absorbers will be determined and the absorption properties will be ascertained.

0.5 POTENTIAL IMPACT AND USE[1]

The potential impact of the use of the low frequency noise absorbers is to reduce the low frequency airborne noise inside the buildings.

0.6 PARTNERS INVOLVED AND THEIR CONTRIBUTION

APT is involved in designing and testing the solutions to reduce low frequency noise by measures at the façade walls.

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1 *including the socio-economic impact and the wider societal implications of the project so far*

0.7 CONCLUSIONS

Trucks and buses are major contributors to traffic noise. At low speeds, the engine and exhaust typically produce low frequency noise (LFN) with dominant frequencies between 31,5 Hz en 63 Hz.

Commonly used window types do not perform well when it comes to low frequency sound insulation. Trucks and buses passing by at low speeds and at close proximity to building façades therefore generate noise inside the building with a high low frequency content.

In the CityHush project, two LFN absorbers for installation on the façade are developed to reduce the LFN around the exposed windows, so they will transmit less to the inside of the building: the solid panel absorber and the perforated panel absorber.

Both LFN absorbers are designed to have a resonance frequency that matches the frequencies of the traffic noise. The resonance frequency of the solid panel absorber depends on the mass of the panel and on the depth of the cavity behind. For the perforated panel absorber it is the combination of the perforations and the cavity behind that determines the resonance frequency. Both systems are known to be efficient absorbers for LFN.

Tables 5.1, 5.2 and 5.3 give possible dimensions for respectively an aluminium solid panel absorber and two perforated panel absorbers designed to absorb sound at frequencies of 31,5 Hz and 63 Hz.

f_r [Hz]	d [m]	Material	ρ [kg/m³]	t [m]	m'' [kg/m²]
31,5	0,90	Aluminium	2800	0,0014	4
63	0,30	Aluminium	2800	0,0011	3

Table 5.1 Solid panel absorber design

f_r [Hz]	l [m]	b [m]	d [m]	r [m]	perforation rate [%]
31,5	0,03	0,04	1,0	0,0025	1,1
63	0,03	0,04	1,0	0,0050	5,1

Table 5.2 Perforated panel absorber design 1

f_r [Hz]	l [m]	b [m]	d [m]	r [m]	perforation rate [%]
31,5	0,04	0,14	0,6	0,0080	1,1
63	0,04	0,14	0,6	0,0180	5,5

Table 5.3 Perforated panel absorber design 2

Both LFN absorbers, the solid panel absorber and the perforated panel absorber, can be used for two applications. The first application consists of panels next to the drive-line of buses and trucks. The second application consists of panels placed on the balconies of dwellings.

In the months M12 to M24, the prototypes of both LFN absorbers will be built and tested in the lab. The characteristics and dimensions of the absorbers will be verified and fine-tuned so that the LFN absorbers will resonate at the desired frequencies and to maximise the absorption ratio.

1 SOURCE SPECTRA BUSES AND TRUCKS

Trucks and buses are major contributors to traffic noise. At speeds lower than ±50 km/h the noise emission of the drive-line (exhaust, engine) of buses and trucks exceeds the tyre-road interaction (rolling noise), see figure 1.1 and 1.2.

The noise produced by the exhaust and by the engine at low rotational speeds, is typically low frequent.

Therefore, for buses and trucks driving at low speeds, the noise emitted has a low frequency content.

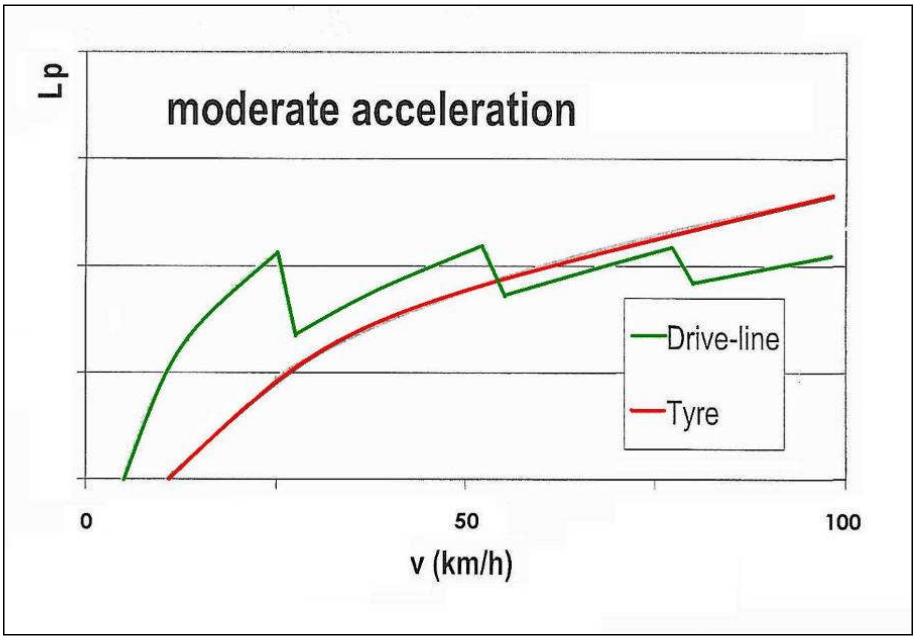


Figure 1.1
Noise level versus speed at moderate acceleration for buses and trucks

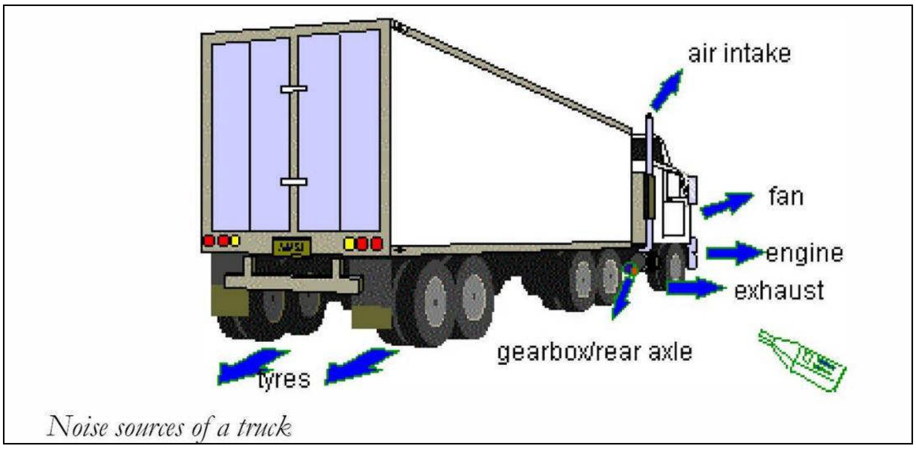


Figure 1.2
Noise sources of a truck

1.1 MEASUREMENT CAMPAIGN

Measurements have been performed near a bus station in a city centre. The city holds a speed restriction of 30 km/h.

Table 1.1 summarises the results from the measurement campaign. In the figures A.1 to A.12 in appendix A, following information is given:

- Figures A.1, A.4, A.7, A.10:
  - Overall level [dB(A)] versus time [s];
  - Maximum level: level [dB(A)] versus third-octave band frequency [Hz];
  - Equivalent level: level [dB(A)] versus third-octave band frequency [Hz];
- Figures A.2, A.5, A.8, A.11:
  - Overall level [dB] versus time [s];
  - Maximum level: level [dB] versus third-octave band frequency [Hz];
  - Equivalent level: level [dB] versus third-octave band frequency [Hz];
- Figures A.3, A.6, A.9, A.12:
  - Third-octave band frequency [Hz] versus time [s] versus level [dB].

Event	Distance from source [m]	Bus speed [km/h]	Equivalent level L_{Aeq} [dB(A)]	Equivalent level L_{eq} [dB]	Maximum level L_{max} [dB]	Dominant frequency [Hz]	Figures
Bus pass-by	14	30	69,9	89,5	96,3	50	A.1 – A.3
Bus idling	14	0	68,4	89,1	90,4	31,5	A.4 – A.6
Bus departure	14	0-30	76,2	98,4	104,3	50-63	A.7 – A.9
Bus arrival + idling + departure	14	0-30	69,7	90,5	102,6	31,5-63	A.10 – A.12

Table 1.1 Results measurement campaign

In the figures with the dB spectra, the dominant frequencies can easily be detected. It can be seen that the dominant frequencies lie between 31,5 and 63 Hz for buses passing by, arriving, idling and departing at bus stops.

2 LOW FREQUENCY NOISE (LFN): INDOOR CRITERIA

Low frequency noise (LFN) can cause more annoyance than predicted from A-weighted noise levels. To evaluate the LFN exposure, two guidelines based on non-weighted noise levels will be used:

- Guideline from the Dutch Association for Noise Annoyance (Nederlandse Stichting Geluidhinder NSG, reference [1]);
- Guideline from the Swedish National Board of Health and Welfare (Socialstyrelsen, reference [2]).

2.1 NSG-GUIDELINE

In the NSG-guideline, audibility is taken as the criterion for the assessment of LF noise: as soon as LFN is audible, it can be annoying.

The reference curve for audibility in this guideline is based on the 90 % hearing threshold for an average group of people between 50 and 60 years old. In this group of people, 90 % will not hear noise below the reference curve; 10 % will be able to hear noise that is (just) below this curve.

The frequency range considered lies between the third-octave bands of 20 and 100 Hz. From measurements in situations with LFN complaints, no audible noise levels at frequencies lower than 20 Hz where found. Frequencies higher than 100 Hz can be assessed with the usual A-weighting.

Frequency [Hz]	20	25	31,5	40	50	63	80	100
Reference curve [dB]	74	62	55	46	39	33	27	22

Table 2.1 NSG-guideline: reference curve

2.2 SWEDISH GUIDELINE

The criteria are given for third-octave bands between 31 and 200 Hz.

Frequency [Hz]	31,5	40	50	63	80	100	125	160	200
Reference curve [dB]	56	49	43	41,5	40	38	36	34	32

Table 2.2 Swedish guideline: reference curve

2.3 COMPARISON GUIDELINES

In figure 2.1, the NSG-guideline and the Swedish guideline are shown (dB-level versus frequency).

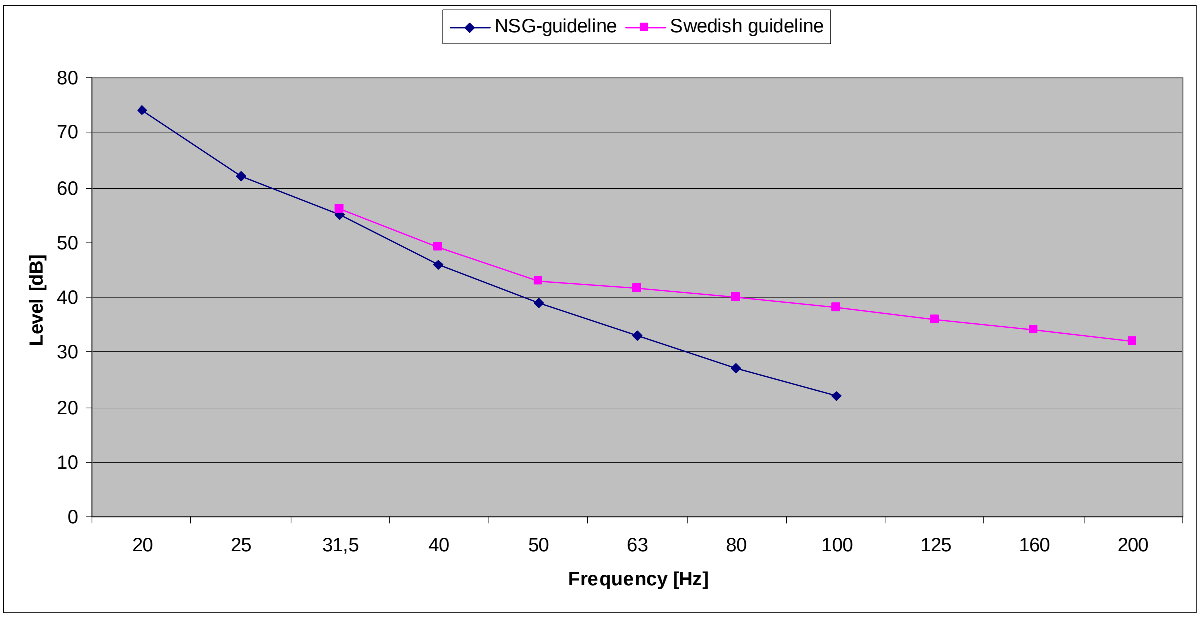


Figure 2.1

Comparison NSG-guideline and Swedish guideline

From the comparison between both guidelines, it can be concluded that the Swedish guideline is less severe than the NSG-guideline.

Both guidelines will be used further on to evaluate indoor noise levels caused by buses driving outdoor.

3 NOISE MODEL BUS STOP

In this paragraph, the results from a noise model from a bus stop in a city centre are presented. The model has been made using the modelling software *IMMI 2010*. The goal of the model is to calculate the indoor noise levels that occur when a bus is idling at a bus stop or when a bus is departing from a bus stop.

3.1 MODEL

The model includes following elements:

- Point source at a height of 0,75 m (ISO 9613);
- Receiver at a distance of 2 m in front of a façade and at a distance of 20 m from the source.

The model uses two source spectra:

- Source spectrum 1: sound power spectrum in dB from maximum level during idling from bus (see §1.1, figure A.5 and figure 3.1);
- Source spectrum 2: sound power spectrum in dB from maximum level during bus departure (see §1.1, figure A.8 and figure 3.1).

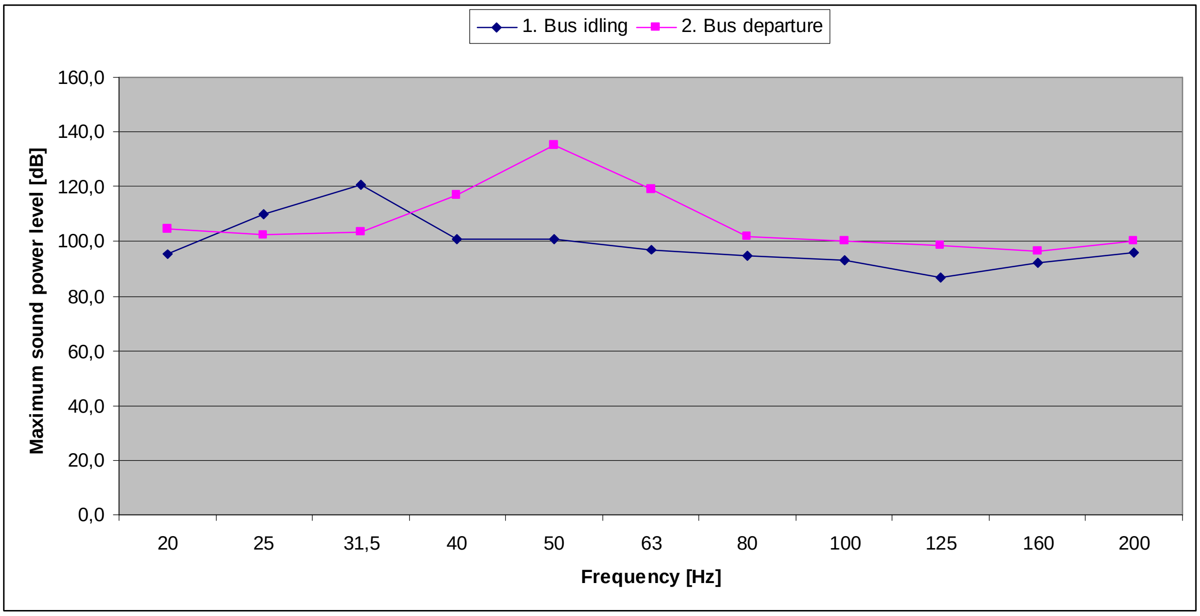


Figure 3.1 Source spectra: bus idling and bus departure

3.2 OUTDOOR NOISE LEVELS

The outdoor noise levels in the receiver position are calculated using the model. The results for the two different source spectra, and for frequencies between 20 and 200 Hz, are given in table 3.1.

	Outdoor noise level [dB]											Overall level	
Frequency [Hz]	20	25	31,5	40	50	63	80	100	125	160	200	[dB]	[dB(A)]
1. Bus idling	66	81	92	71	72	68	66	64	58	63	66	92	70
2. Bus departure	75	73	74	88	106	90	73	71	69	67	71	106	78

Table 3.1 Noise model bus stop: outdoor noise levels

The levels in the table 3.1 are the levels at a distance of 2 m in front of a façade, at a distance of 20 m from the source. The levels include incident and reflected sound against the façade.

3.3 FAÇADE INSULATION

To calculate the indoor noise levels, the insulation values from the façade have to be known. Table 3.2 and figure 3.2 show the insulation values (R_W) from commonly used windows in a city centre:

- Single pane glazing: 6 mm glass;
- Double pane glazing: 6 mm glass – 12 mm air void – 6 mm glass;
- Double pane glazing: 6 mm glass – 20 mm air void – 10 mm glass.

	Insulation values R [dB]											Overall level R_w
Frequency [Hz]	20	25	31,5	40	50	63	80	100	125	160	200	[dB]
6	9	10	11	13	15	16	17	18	20	21	23	31
6-12-6	12	13	15	16	17	18	20	21	21	21	17	35
6-20-10	14	16	18	19	20	21	21	22	20	23	29	38

Table 3.2 Window insulation values: R

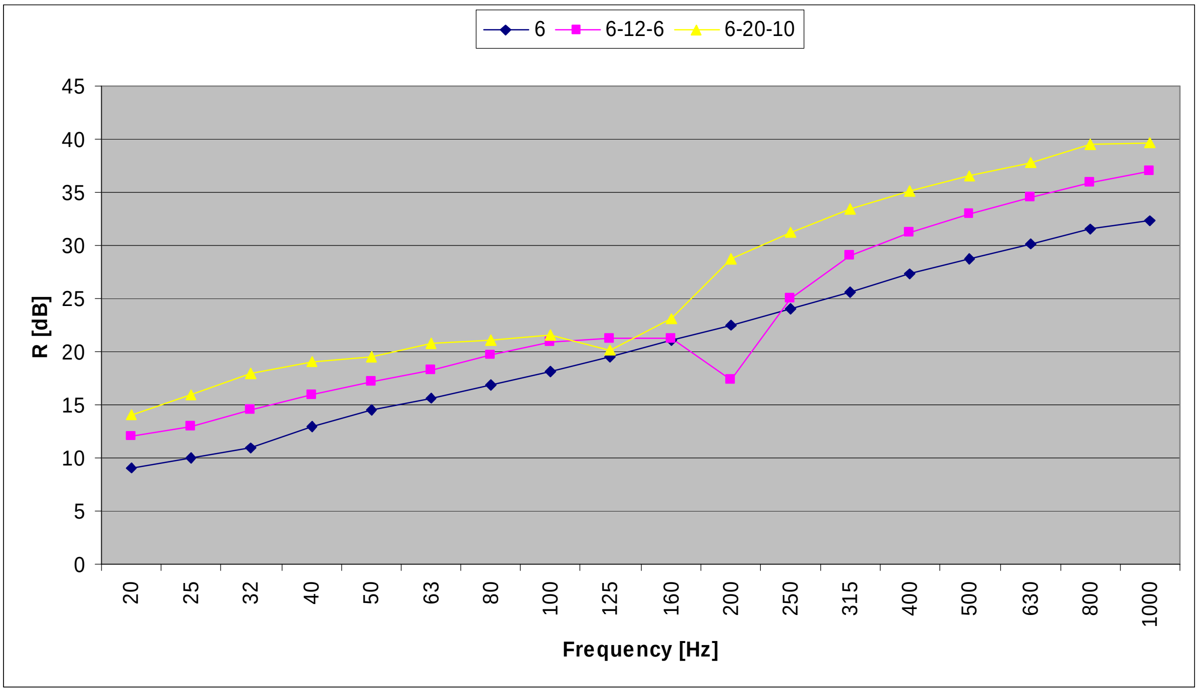


Figure 3.2

Window insulation values: R [dB]

3.4 INDOOR NOISE LEVELS

With the model, the outdoor noise levels in front of a façade are calculated. Now, the indoor noise levels can be calculated from the outdoor noise levels and the insulation values.

Following formula is used to calculate the indoor noise levels:

$$R = L_{out} - L_{in} + 10\log\left(\frac{4 \cdot S}{A}\right) \quad (3.1)$$

Assume $4S/A = 1{,}1$:

$$L_{in} = L_{out} - R + 0{,}3 \quad (3.2)$$

The indoor noise levels for 3 different types of window insulations are given in table 3.3 for buses idling and in table 3.4 for buses departing:

	1. Bus idling – indoor noise level [dB]											Overall level	
Frequency [Hz]	20	25	31,5	40	50	63	80	100	125	160	200	[dB]	[dB(A)]
6	57	71	81	59	57	53	49	46	38	42	44	81	44
6-12-6	54	68	77	56	55	50	46	43	37	42	49	78	43
6-20-10	52	65	74	53	52	47	45	42	38	40	38	74	38

Table 3.3 Noise model bus stop: 1.Bus idling – indoor noise levels

	2. Bus departure – indoor noise level [dB]											Overall level	
Frequency [Hz]	20	25	31,5	40	50	63	80	100	125	160	200	[dB]	[dB(A)]
6	67	63	64	75	91	75	56	53	50	47	48	92	62
6-12-6	64	60	60	72	89	72	53	50	49	47	54	89	59
6-20-10	62	57	57	69	86	70	52	49	50	45	42	87	57

Table 3.4 Noise model bus stop: 2.Bus departure – indoor noise levels

The indoor noise levels in dB are compared to the NSG-guideline and the Swedish guideline in following figures 3.3 (bus idling) and 3.4 (bus departing).

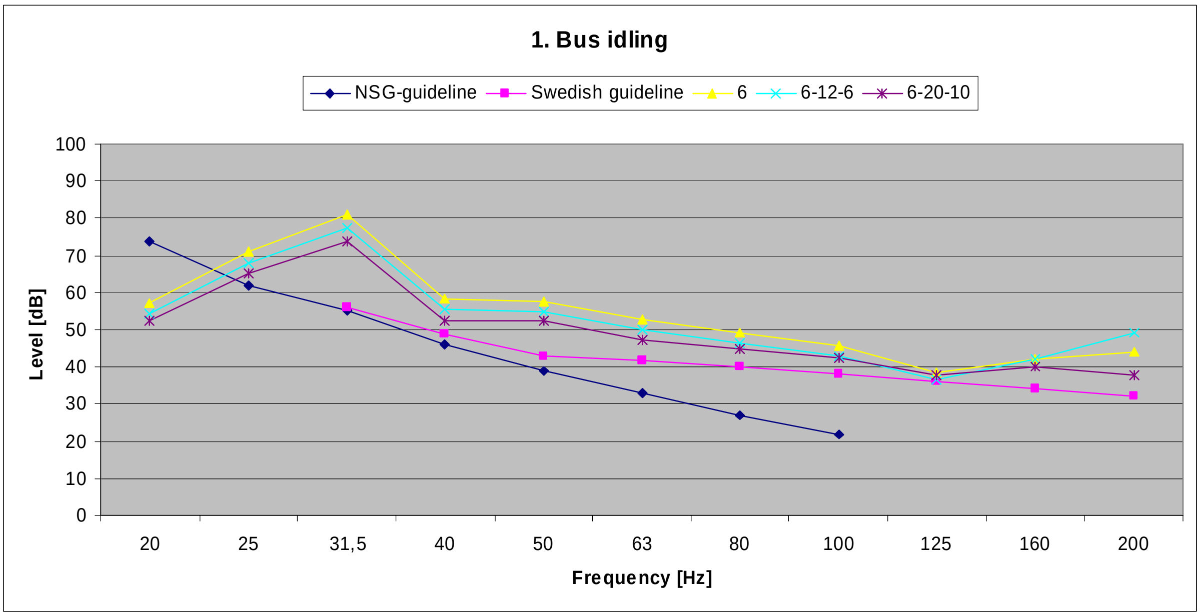


Figure 3.3

Noise model bus stop: 1.Bus idling – indoor noise levels compared to NSG-guideline and Swedish guideline

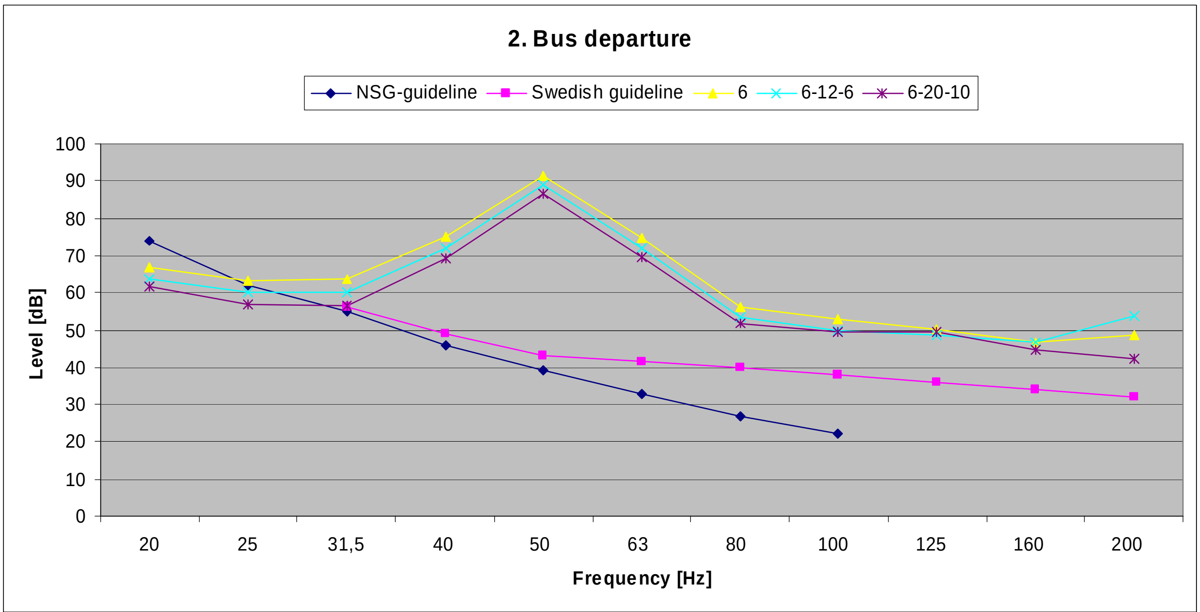


Figure 3.4

Noise model bus stop: 2.Bus departure – indoor noise levels compared to NSG-guideline and Swedish guideline

From both figures it can clearly be seen that, at the dominant frequencies of 31,5 Hz and 50 Hz, both guidelines for low frequency noise (LFN) are largely exceeded.

In the following chapter, solutions will be developed to lower the low frequency noise content indoors: concepts will be designed for high low frequency absorption at façade walls.

4 CONCEPT FOR HIGH LOW FREQUENCY ABSORPTION AT BUILDING FAÇADES

4.1 SOLID PANEL ABSORBER

4.1.1 Theory

If a plate or panel is mounted in front of a rigid wall, the arrangement behaves in the same way as a spring-mass system, the plate being the mass and the enclosed air being the spring.

When a sound wave impinges on the system, the system will tend to be set into vibration. The maximum transfer of energy occurs when the frequency of the incident sound is the same as the resonant frequency of the system.

Since the panel possesses inertia and damping, some of the sound energy is converted into mechanical energy and dissipated as heat, therefore sound absorption occurs. However since the panel itself vibrates it will act as a sound radiator so that it is rare to find such a system with an effective absorption coefficient greater than 0,5.

The resonant frequency of such a system can be calculated from:

$$f_r = \frac{60}{\sqrt{m'' \cdot d}} \tag{4.1}$$

where f_R resonance frequency [Hz]
d distance between the plate and the wall [m]
m'' mass of the plate = $\rho \cdot t$ [kg/m²]
ρ density of plate material [kg/m³]
t thickness of the plate [m]

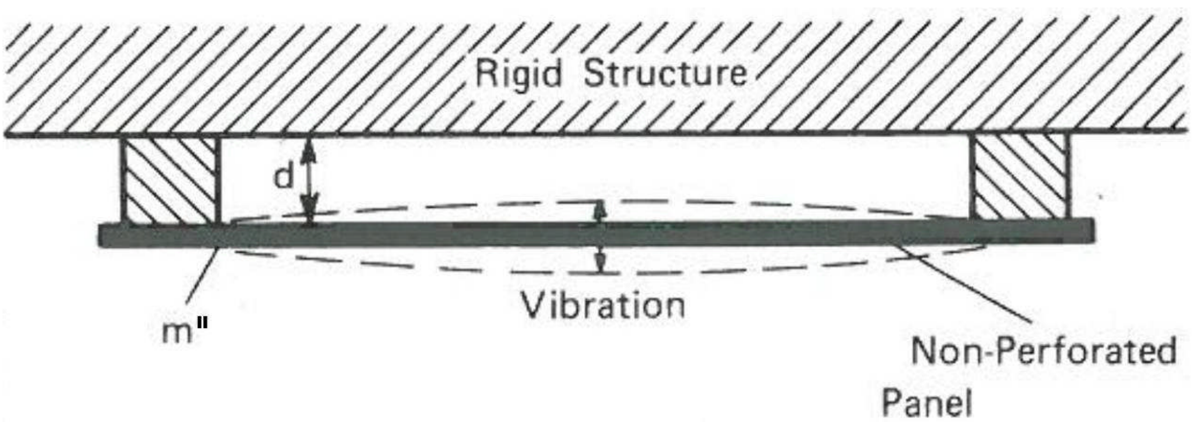


Figure 4.1

Solid panel absorber (reference [3])

Such solid panel absorbers are known to be useful at mid- and low-frequencies.

At the resonance frequency, absorption values between 0,3 and 0,5 are obtained. The absorption of sound energy falls off rapidly at frequencies above the resonant frequency. Further damping may be obtained by introducing damping material into the air volume between the wall and the panel. This broadens the range of frequencies for which the resonator is active and an absorption value between 0,5 and 0,8 can be obtained.

The enclosed air only works as a spring as long as the distance to the wall d is small compared to the wavelength of the incident sound:

$$d_{max} = \frac{\lambda_r}{12} = \frac{28}{f_r} \qquad (4.2)$$

For the octave bands of 31,5 and 63 Hz, the maximum distance d between the panel and the wall is given in table 4.1:

Frequency [Hz]	31,5	63
d_{max} [m]	0,90	0,45

Table 4.1 Solid panel absorber: maximum distance between panel and wall

To lower the resonance frequency f_r of a solid panel absorber, either the mass of the plate m'' can be increased or the distance between the plate and the wall d can be made larger. When the mass is increased, the peak in the frequency range of the absorption value will be smaller, whereas with a larger distance, the peak will be wider. Since it is more interesting to have a high absorption value for a wider frequency range, the distance to the wall d should be made as big as possible, considering the restriction of formula (4.2).

To obtain a high efficiency when designing a solid panel absorber, the following rules must be applied:

- The distance between the panel and the wall in cm must be at least twice the mass of the plate in kg/m²: $d_{min} = 0{,}02 \cdot m''$;
- The plate must be able to move freely:
  - The absorbing material has to be placed near the plate, and must not be placed tightly between the plate and the wall;
  - The distance between the supports of the plate must be larger than 0,5 m;
  - The "free" surface of the plate must be at least 0,4 m².

4.1.2 Description of the solid panel absorber design

In this section the dimensions of two solid panel absorbers will be determined using the theory from the previous chapter so that the resonance frequencies of the two solid panel absorbers are 31,5 and 63 Hz respectively. These are the dominant frequencies for buses passing by, arriving, idling and departing at bus stops (see chapter 1).

The previous section mentions that in order to have a wide frequency range for the absorption value, the distance between the plate and the wall should be as high as possible, but the distance should not exceed the values stated in table 4.1.

The thickness of the plate t can be calculated as follows (see formula (4.1)):

$$t = \frac{60^2}{f_r^2 \cdot \rho \cdot d} \tag{4.3}$$

When an aluminium sheet with a density ρ of 2800 kg/m³ is used as the plate material, the thickness of the plate can be determined with formula (4.3) for the frequencies of 31,5 and 63 Hz:

f_r [Hz]	d [m]	Material	ρ [kg/m²]	t [m]	m'' [kg/m²]
31,5	0,90	Aluminium	2800	0,0014	4
63	0,30	Aluminium	2800	0,0011	3

Table 4.2 Solid panel absorber: possible dimensions

4.2 HELMHOLTZ RESONATOR AND PERFORATED PANEL ABSORBER

4.2.1 Theory

A Helmholtz resonator consists of a volume of air contained within a cavity connected to the air outside the cavity by a small opening known as the neck (figure 4.2). When a sound wave impinges on the opening of the neck, the air in the neck will be set into vibration and the air in the cavity will undergo periodic compression and rarefaction. The friction between the resulting amplified motion of the air particles in the neck itself causes sound energy to be absorbed. The absorption of the undamped resonator falls off very rapidly at frequencies above and below the resonant frequency.

The resonance frequency of an undamped resonator may be evaluated from:

$$f_r = \frac{c}{2 \cdot \pi} \sqrt{\frac{S}{V \cdot (l + 2 \cdot \Delta l)}} \quad (4.4)$$

where f_R resonance frequency [Hz]

c velocity of sound [m/s]

S cross sectional area of the neck [m²]

V volume of cavity [m³]

$l + 2 \cdot \Delta l$ effective neck length [m]

l length of neck [m]

$2 \cdot \Delta l = 1{,}6 \cdot r$ [m] for round opening (r = radius of opening [m])

$2 \cdot \Delta l = 0{,}9 \cdot a$ [m] for square opening (a = side of square [m])

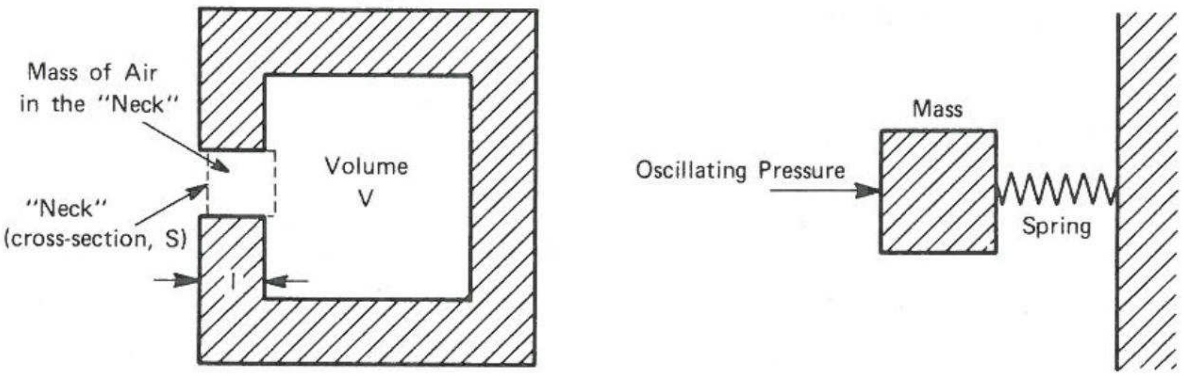


Figure 4.2
Helmholtz resonator and the equivalent mechanical analogy (reference [3])

The limitations to the volume V and the neck length l are given by following formulas:

$$V_{max} = 1{,}3 \cdot 10^{5} \cdot \frac{1}{f_r^3} \quad (4.5)$$

$$l_{max} = 43 \cdot \frac{1}{f_r} \quad (4.6)$$

For the octave bands of 31,5 and 63 Hz, the maximum volume V and the maximum neck length l are given in table 4.3:

Frequency [Hz]	31,5	63
V_{max} [m^3]	4,2	0,5
l_{max} [m]	1,4	0,7

Table 4.3 Helmholtz resonator: maximum volume and maximum neck length

Formula (4.6) is also valid for the diameter for round openings, for the side for square openings or for the width for slits.

If the resonator is damped by lining the cavity and the neck with a porous sound absorbing material then the resonator will be effective over a wider bandwidth although it maximum absorption at resonance will be reduced (figure 4.3).

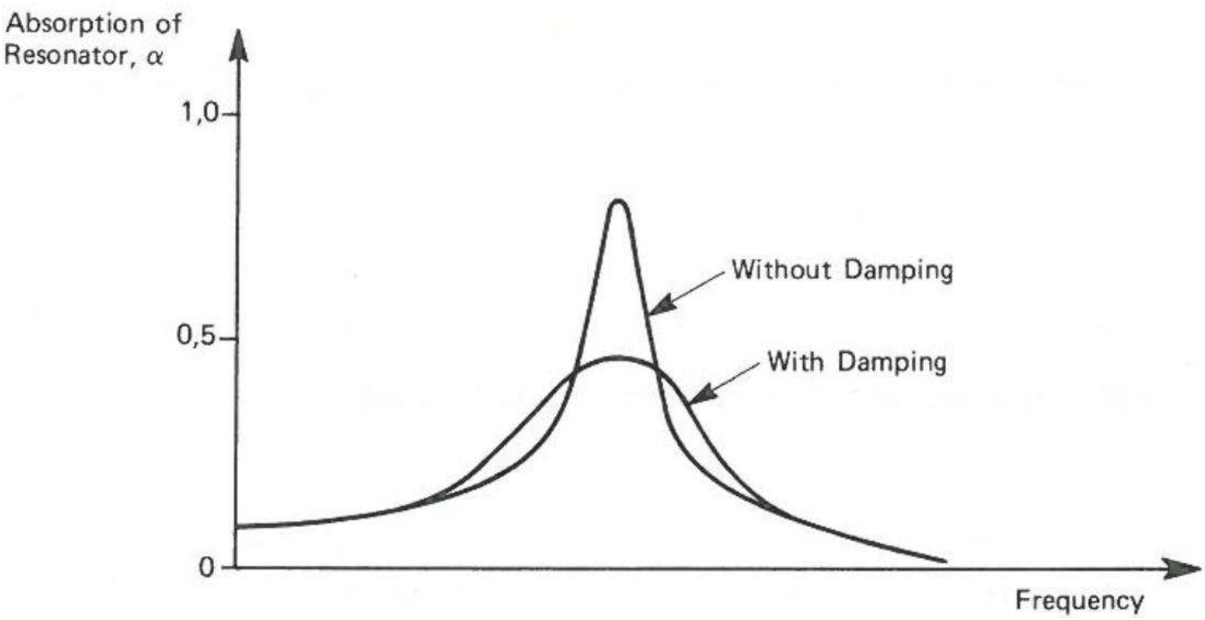


Figure 4.3

Typical sound absorption characteristics of a Helmholtz resonator with and without damping (reference [3])

Helmholtz resonators are known to be most efficient at lower frequencies.

A much more common application of the Helmholtz resonator principle for sound absorption is to be found in the perforated panel absorber. This consists of a panel or plate with a drilled of punched pattern of holes, mounted in such a way so as to enclose an air space between itself and the wall (figure 4.4). The slots or holes form the neck of the resonators and the portion of the air space behind each hole forms the cavity of the resonator.

Usually there is no need to divide the separate resonator cavities from one another by partitions. As for the single resonator, the resonant frequency of the multiple resonator is determined by the dimensions of the neck and the cavity but the multiple resonator is not so selective in its absorption. A damping material such as mineral wool or fibre glass is inserted into the air space of perforated panel absorbers, thus increasing the effectiveness of the absorption above and below the resonant frequency (figure 4.5).

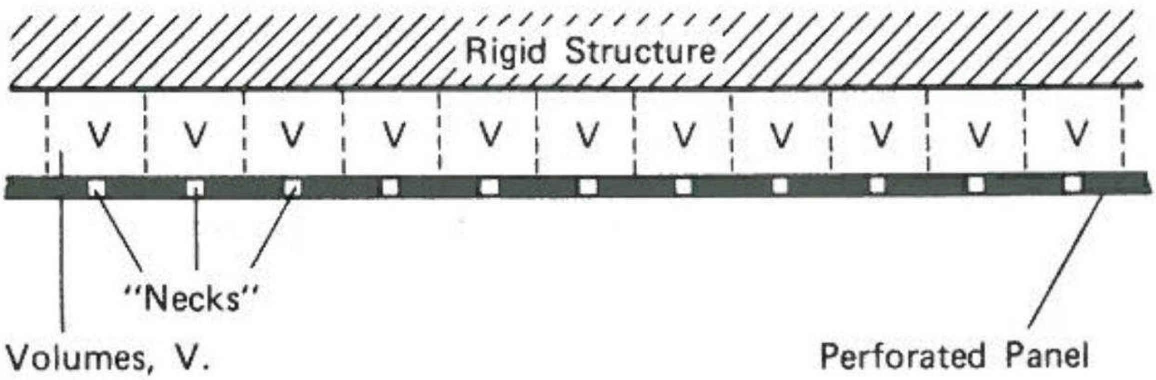


Figure 4.4
Perforated panel absorber seen as an assembly of Helmholtz resonators (reference [3])

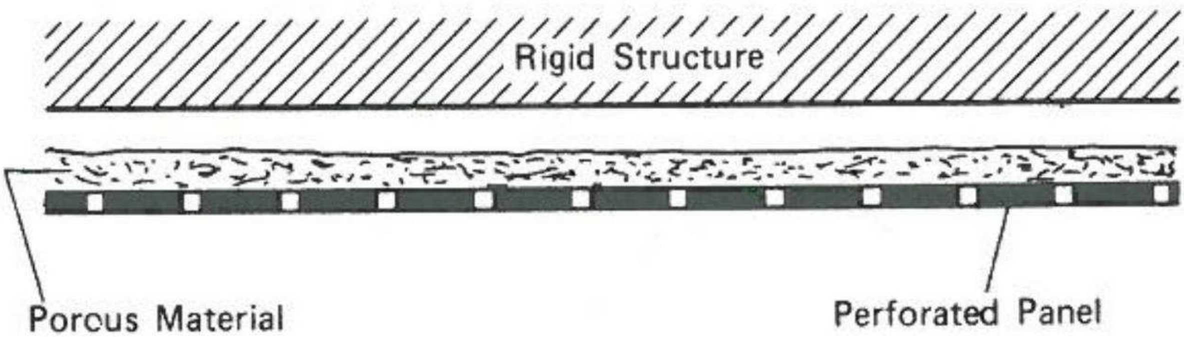


Figure 4.5
Perforated panel absorber lined with damping material (reference [3])

For panels with regularly spaced holes of the same dimensions such resonator has the same resonant frequency. It is possible to alter the spacing and dimensions of the holes to produce a panel with the desired absorption characteristics.

The perforation rate of the panel may not be too small: a low percentage of perforation remarkably diminishes the absorption of the acoustic material behind the panel. As a guideline it can be stated that the perforation rate should be more than 1 %.

The size of the perforations cannot be too large since the acoustic resistance for large perforations will not be sufficient.

4.2.1.1 Sound absorption at the resonance frequency

In a tuned perforated panel absorber, the sound absorption reaches a maximum value, α_{max}, at the resonance frequency, f_r, falling off to lower values at higher and lower frequencies.

This maximum value of absorption can be controlled by the choice of the sound absorptive material with which the airspace is filled. The maximum value of absorption depends only on the flow resistance of that material, and not on any of the physical dimensions of the sound absorptive treatment (such as the depth of airspace, perforation diameter, percent open area, etc.).

The flow resistance of a piece of material tells how easy it is for air to move through the material. The flow resistance depends upon the density of the absorptive material and the fibre diameter: generally, the heavier the blanket and the finer the fibres, the higher the flow resistance. Thicker layers also have more flow resistance than thin ones.

The flow resistance, R, of a layer of material is always related to the characteristic impedance of the air given by the product of the density of the air ρ, and the propagation velocity of sound c, by forming the resistance ratio $R/\rho c$.

If a layer of material has a flow resistance such that $R/\rho c = 1$, then a sound wave will not recognize the existence of that material, because it can't tell the difference between this material and air.

If the value of $R/\rho c$ is either substantially greater or less than unity, then the sound wave will notice the layer, and tends to be reflected from it rather than entering and passing through it.

The maximum amount of absorption achieved at the resonance frequency is calculated as follows:

$$\alpha_{max} = \frac{1}{\frac{1}{2} + \frac{1}{4}\left(R/\rho c + \rho c/R\right)} \tag{4.7}$$

This expression is illustrated in figure 4.6.

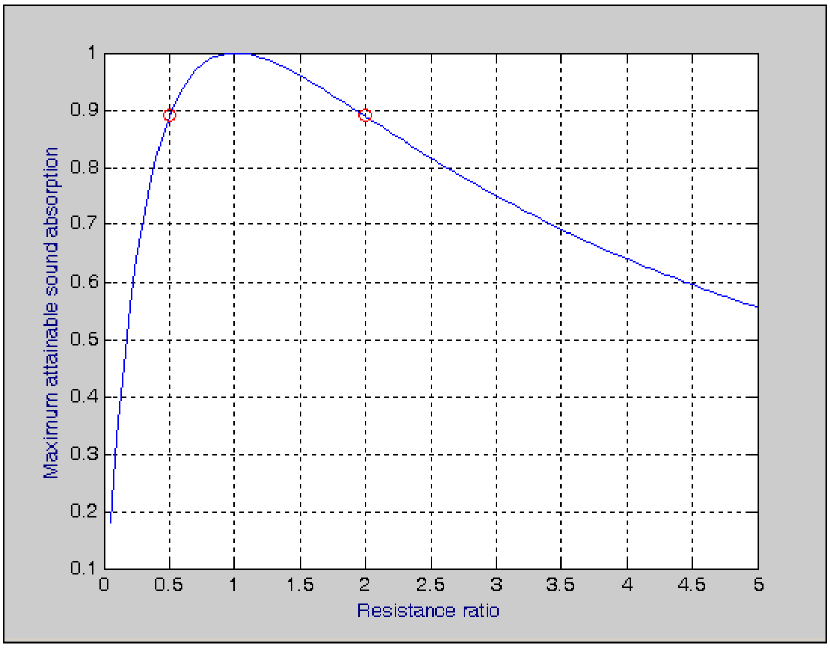


Figure 4.6

Maximum attainable sound absorption at the resonance frequency as a function of the flow resistance ratio of the filling material

As we see, the maximum absorption coefficient at resonance in a tuned absorber is not very sensitive to the filling material. Any value of the resistance ratio from 0.5 to 2.0 will yield a value of α_{max} of 0.89 or greater.

4.2.1.2 Position of the absorptive material in the air space

In figure 4.7, the sound absorption is plotted for three different mountings:

I. Absorptive material against the wall;
II. Absorptive material free between wall and perforated plate;
III. Absorptive material against the perforated plate.

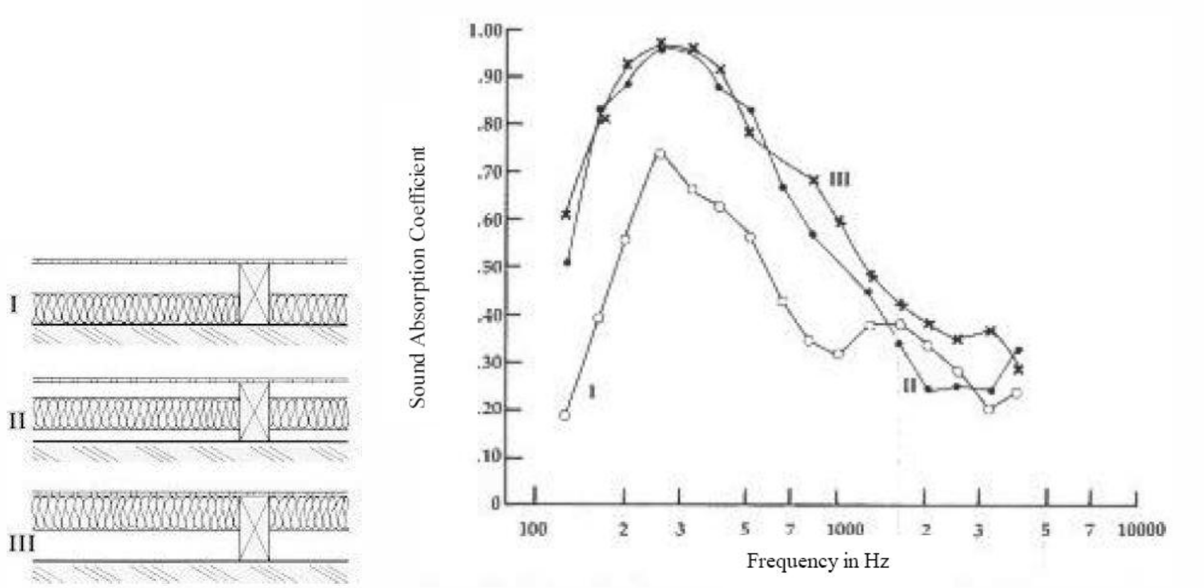


Figure 4.7

Sound absorption versus frequency for three different mountings (reference [5])

It can be seen that the most effective arrangement is that with the absorptive material located near the perforated plate. The worst condition is with the absorptive material close to the wall.

The explanation is that in order for the absorptive layer to work well, turning the sound energy into heat by the friction of the vibrating air particles within the fine pores of the material, there must be freedom for the air particles to move. If anything impedes this motion, then the energy conversion is less efficient and less sound energy is absorbed.

That is what happens at locations near a hard wall: the wall itself, being rigid, cannot move with the sound wave, and this means that the nearby air particles also cannot move. Thus, any sound absorptive material placed against a hard wall is virtually useless, because there can be no air motion within the material to dissipate the sound energy.

4.2.1.3 The absorption bandwidth

Another important characteristic of a tuned absorber is the width of the resonance peak.

The absorption bandwidth of a resonant sound absorber can be characterized by determining the two frequencies, f_2 and f_1 (above and below the resonance frequency, respectively) at which the absorption has dropped to half its value at resonance. For frequencies below f_1 and above f_2, the absorption of the tuned absorber is relatively insignificant.

The difference between f_2 and f_1 is called the "Half-Power Bandwidth" because at all frequencies within this band the sound absorption exceeds half the (maximum) value at resonance:

$$\Delta f_h = f_2 - f_1 = 2 \cdot \pi \cdot [1 + (R/\rho c)] \cdot (d/c) \cdot f_r^2 \qquad (4.8)$$

and

$$f_{1,2} = f_r \pm (\Delta f_h / 2) \qquad (4.9)$$

where d distance between the plate and the wall [m]

These quantities are shown in figure 4.8. The absorption coefficient a is given by:

$$\frac{\alpha}{\alpha_{max}} = \frac{1}{1 + (f_r/\Delta f_h)^2 \cdot (f/f_r - f_r/f)^2} \qquad (4.10)$$

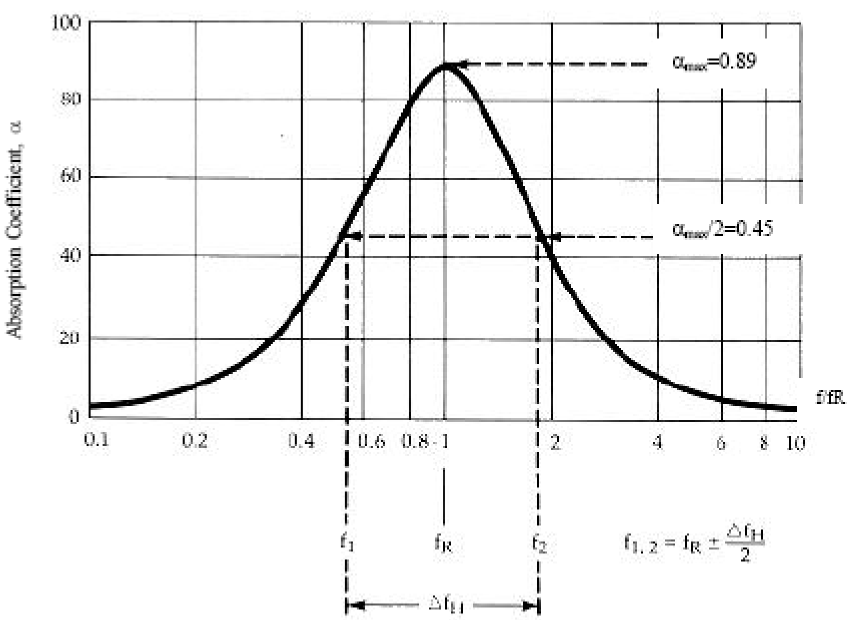


Figure 4.8

Curve of absorption defining α_{max}, f_1, f_2 and Δf_h (reference [5])

The higher the resonance frequency, or the greater the flow resistance of the filling, or the greater the depth of the airspace, the wider the frequency band over which high sound absorption will occur (see formula (4.8)).

The left plots in figure 4.9 show the influence of the resistance ratio on the absorption curve. All absorbers have the same dimensions and consequently the same resonance frequency. Values of R/ρc greater than unity lead to broader absorption curves, while values less than unity give narrower curves.

The right plots in figure 4.9 show the influence of the depth d of the air cavity volume on the absorption curve. All absorbers are filled with an absorptive material having R/ρc = 1, but with different values of d, while keeping the same resonance frequency. The choice of R/ρc = 1 causes the sound absorption at the resonance (f/f_r=1) to be 100% in all cases. But higher values for d lead to absorption curves that are broader; and lower values for d lead to narrower curves.

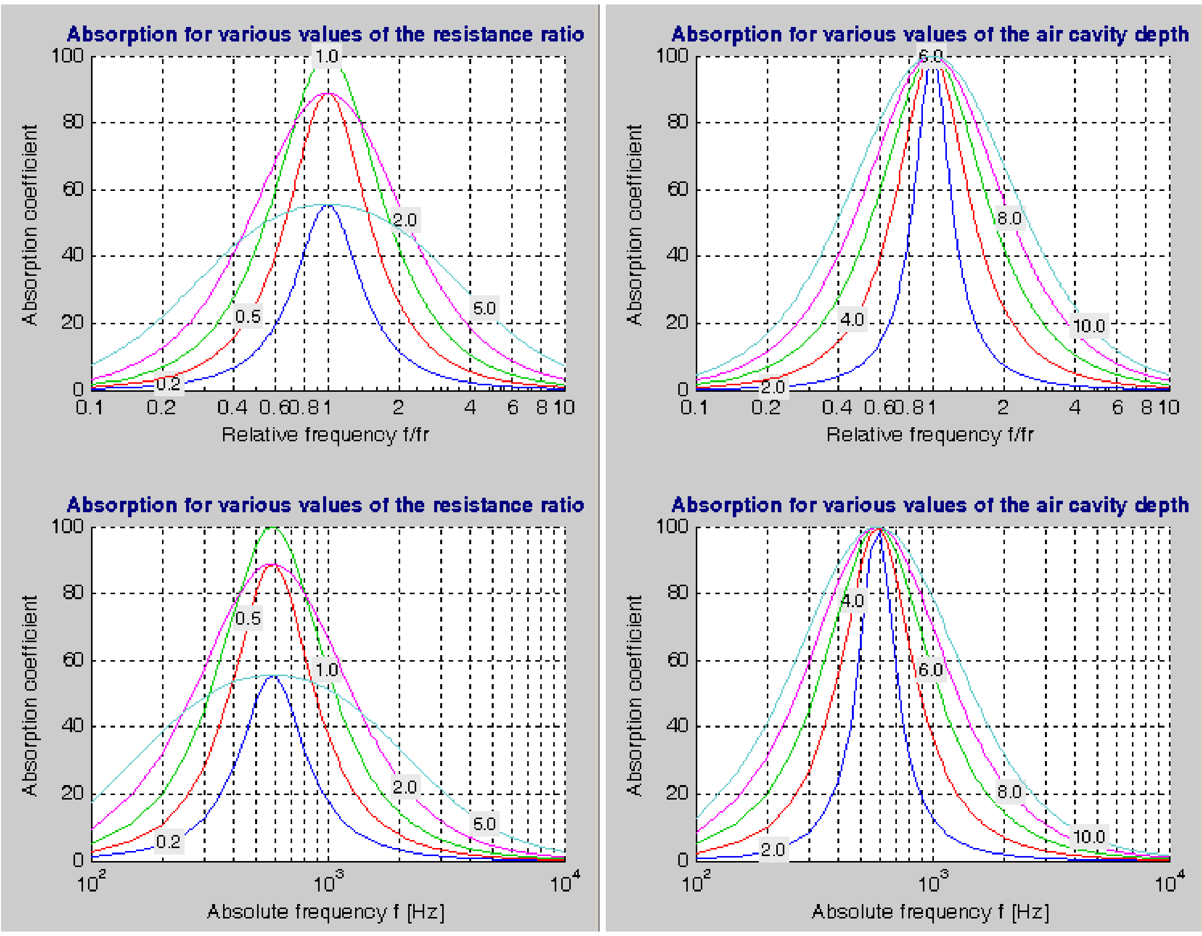


Figure 4.9

Influence of the sound absorptive material and the depth of the air cavity volume on the absorption curve

4.2.1.4 Proper choice for R

It would not be favourable to choose R smaller than ρc, because then both the maximum absorption at resonance and the width of the absorption curve would be decreased.

But if R is greater than ρc, then again the maximum absorption at resonance decreases, but the curve is wider, which may be desirable.

If both the maximum absorption at resonance and the half power bandwidth are optimized, by forming their product:

$$\alpha_{max} \cdot \Delta f_H = 2 \cdot \pi \cdot \left[4 \cdot (R/\rho c) \cdot (1 + R/\rho c)\right] \cdot (d/c) \cdot f_R{}^2 \tag{4.11}$$

It can be seen that it would still be good to choose a large value for R.

However, even a choice of infinitely high R would yield a result for the product that is only twice that for the matching case, R/ρc = 1. And R cannot be too great, since a too-strongly damped resonator is no resonator at all.

The conclusion can be made that a choice of R/ρc around 2 to 3 will give the best compromise between a high maximum sound absorption at resonance and a broad half power bandwidth.

4.2.2 Description of the perforated panel absorber design

The design of the perforated panel absorber results from a number of compromises and considerations related to:

- The frequency range that has to be covered;
- The characteristics and dimensions of the absorptive material;
- The specifications of the perforated panel;
- …

4.2.2.1 Frequency range

The frequency range that has to be covered lies between 31,5 Hz and 63 Hz. These are the dominant frequencies for buses passing by, arriving, idling and departing at bus stops (see chapter 1).

4.2.2.2 Selection of the absorptive material

As mentioned before, a choice of absorptive material such that the ratio $R/\rho c$ is around 2 to 3 will give the best compromise between a high maximum sound absorption at resonance and a broad half power bandwidth.

The following material has been selected for its adequate flow resistance characteristics:

- Isover Mupan Façade ($7x10^3$ Ns/m^4) – 0,12 m – see appendix B, figure B.1;

As the characteristic impedance of the air is equal to 408 Ns/m^3, a 0,12 m thick layer of the absorptive material with a flow resistivity of 7 $x10^3$ Ns/m^4 will give a ratio $R/\rho c$ of 2.

4.2.2.3 Specifications of perforated panel

The required specifications can be calculated in order to make the absorber resonate at the desired frequencies: between 31,5 Hz and 63 Hz.

For a given resonance frequency and a given flow resistance of the filling, the depth of the airspace must be as large as possible to have a wider frequency band over which high sound absorption will occur (see formula (4.8)). The depth of the air space is of course limited for practical reasons.

Design 1

For the first design, a metal panel with following specifications is chosen:

– Sheet thickness l = 30 mm = 0,03 m;
– Distance between the openings b = 0.04 m ($V = b^2 \cdot d$);
– Distance between the panel and the wall d = 1,0 m.

From the formula (4.4), the radius r of the openings (depending on the resonant frequency f_r) can be calculated.

In table 4.4 all the dimensions are given for perforated metal panels, so that they resonate at the desired frequencies of 31,5 and 63 Hz:

f_r [Hz]	l [m]	b [m]	d [m]	r [m]	perforation rate [%]
31,5	0,03	0,04	1,0	0,0025	1,1
63	0,03	0,04	1,0	0,0050	5,1

Table 4.4 Perforated panel absorber design 1

A perforated panel absorber can be developed with the specifications in table 4.4. One panel can be made with two different radii of the openings so that the two dominant frequencies of 31,5 and 63 Hz will be absorbed.

In the figures 4.10 and 4.11 a plot is given of the absorption coefficient α versus respectively the relative frequency f/f_r and the absolute frequency f. The red plot shows the absorption coefficient for the part of the panel where the radius of the openings is 0,0025 m (f_r = 31,5 Hz), the green plot shows the absorption coefficient for the part of the panel where the radius of the openings is 0,0050 m (f_r = 63 Hz) for a distance of 1 m in front of the wall.

From these figures it can be seen that the perforated panel with the two different radii of the openings will have an absorption value that is higher than 0,8 for frequencies that lie between 25 and 110 Hz.

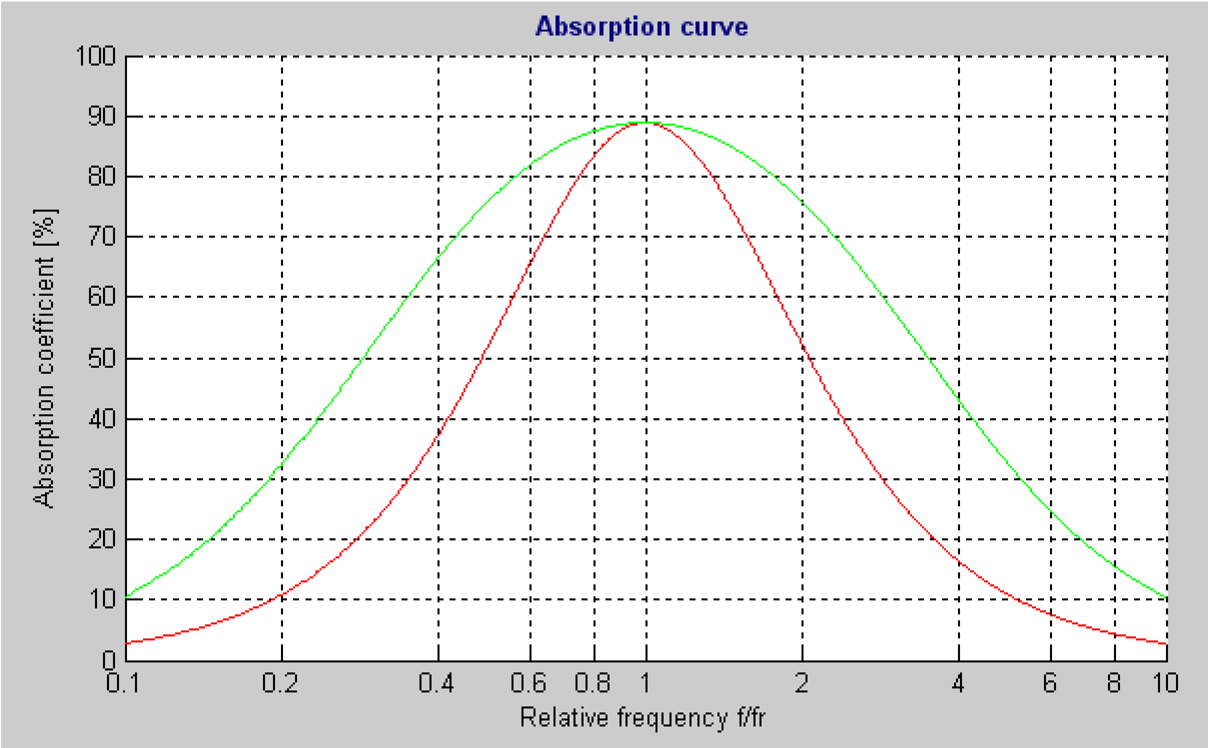


Figure 4.10

Perforated panel absorber design 1: absorption coefficient versus relative frequency

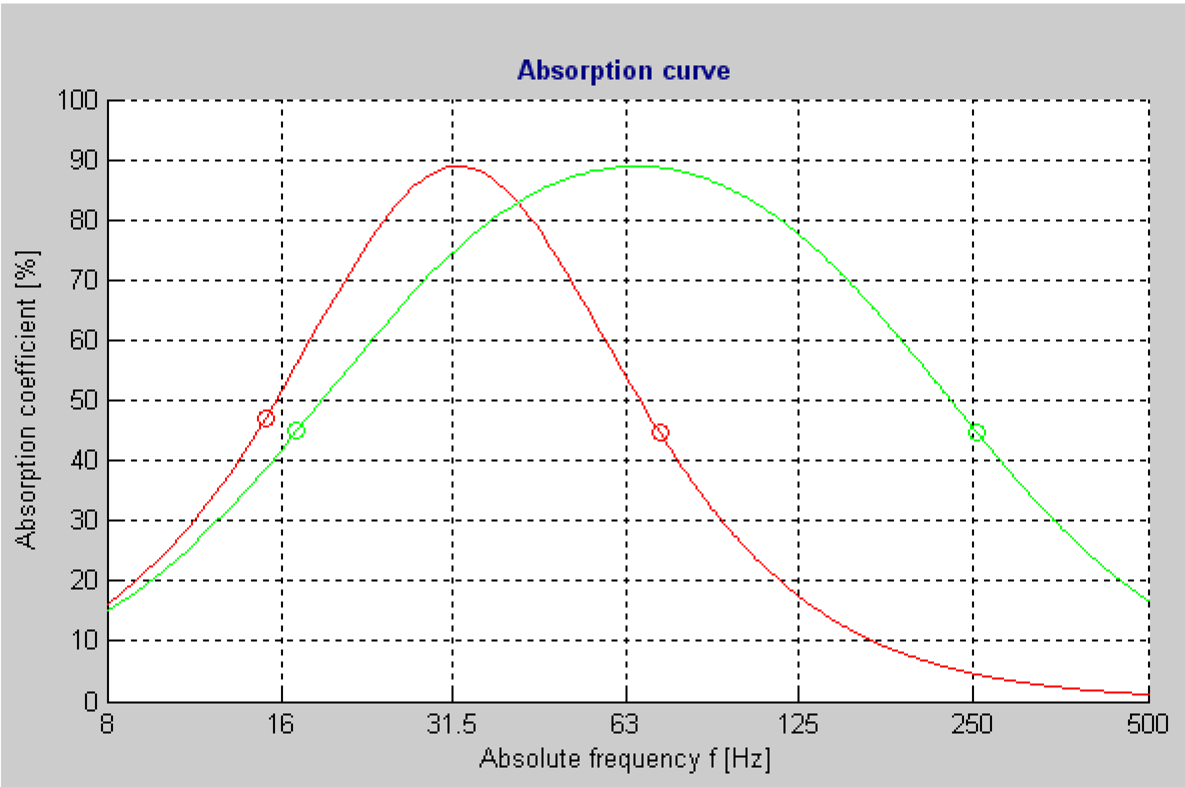


Figure 4.11

Perforated panel absorber design 1: absorption coefficient versus absolute frequency

Design 2

Another design, where the distance between the panel and the wall is smaller, can be made: d = 0,6 m instead of 1,0 m. To keep a perforation rate of more than 1 percent, the panel has to be made thicker, the perforations must be larger and the distance between the perforations is increased.

f_r [Hz]	l [m]	b [m]	d [m]	r [m]	perforation rate [%]
31,5	0,04	0,14	0,6	0,0080	1,1
63	0,04	0,14	0,6	0,0180	5,5

Table 4.5 Perforated panel absorber design 2

In the figures 4.12 and 4.13 a plot is given of the absorption coefficient α versus respectively the relative frequency f/f_r and the absolute frequency f. The red plot shows the absorption coefficient for the part of the panel where the radius of the openings is 0,0080 m (f_r = 31,5 Hz), the green plot shows the absorption coefficient for the part of the panel where the radius of the openings is 0,0180 m (f_r = 63 Hz) for a distance of 0,6 m in front of the wall.

From these figures it can be seen that the perforated panel with the two different radii of the openings will have an absorption value that is higher than 0,75 for frequencies that lie between 25 and 100 Hz.

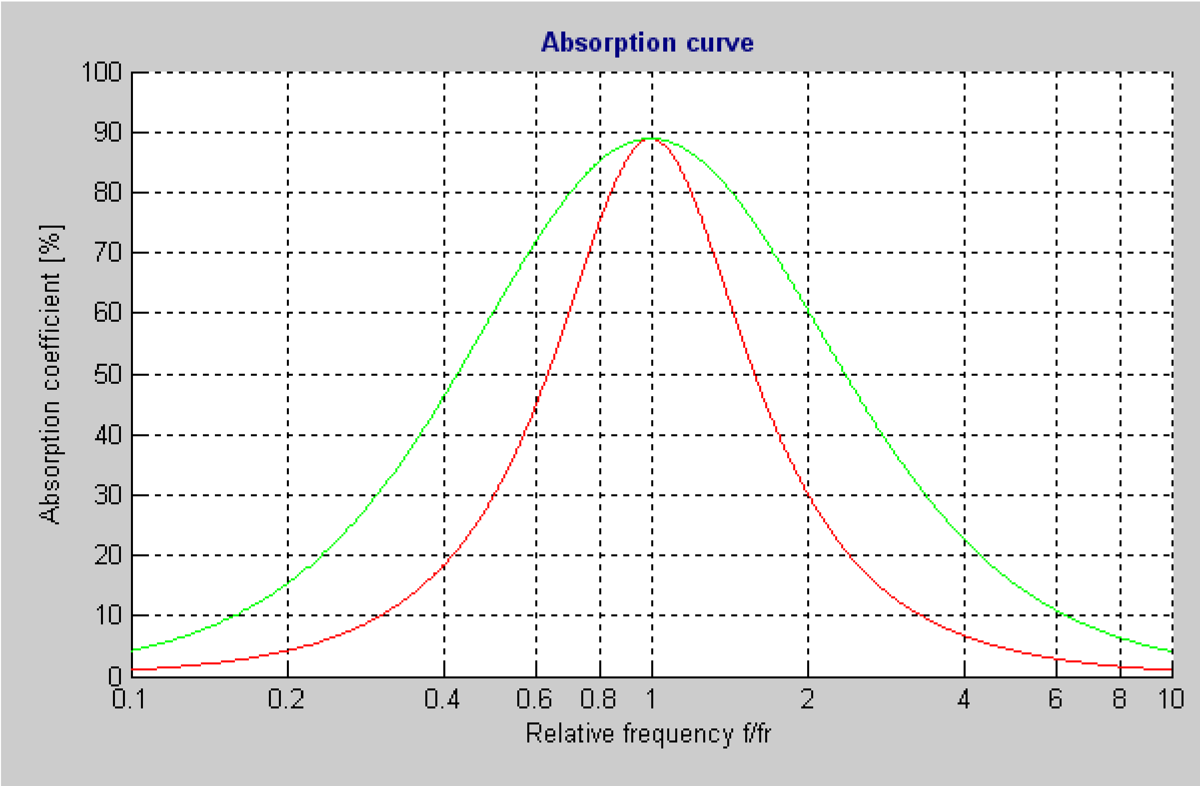


Figure 4.12

Perforated panel absorber design 2: absorption coefficient versus relative frequency

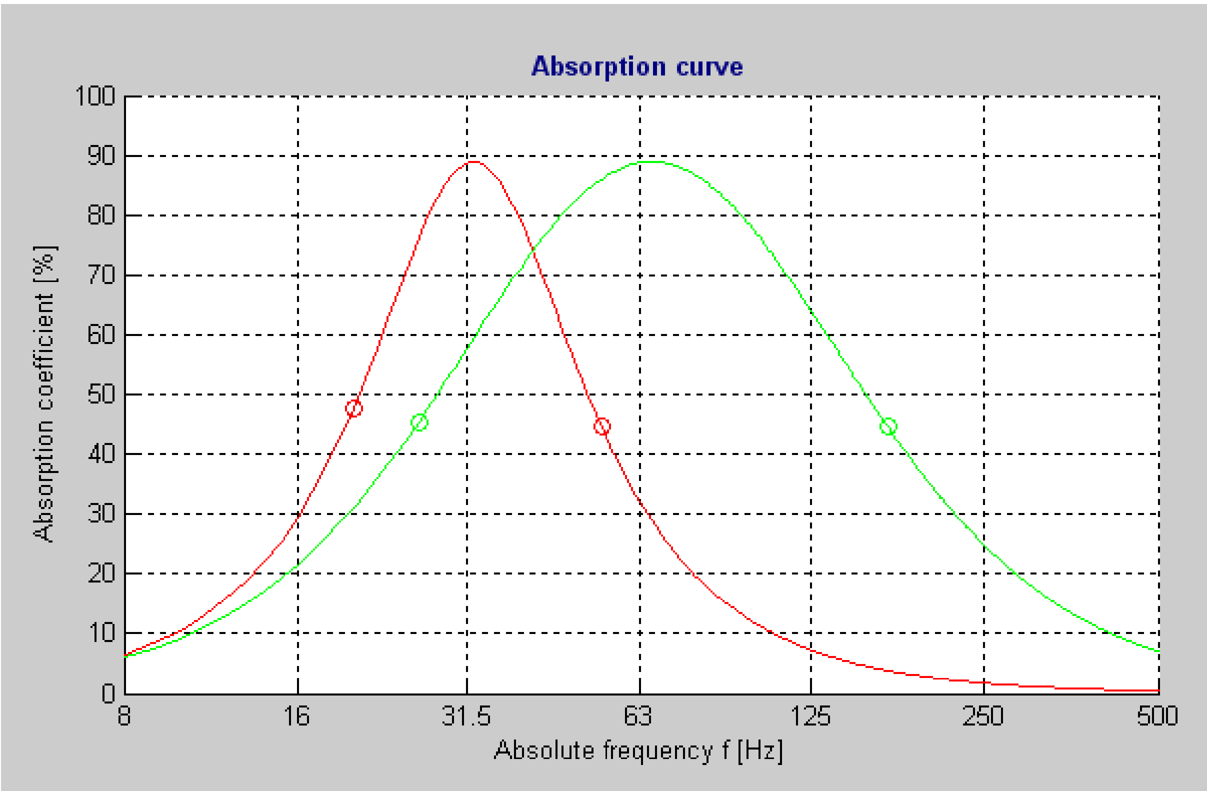


Figure 4.13

Perforated panel absorber design 2: absorption coefficient versus absolute frequency

4.3 APPLICATIONS

Both absorber types, the solid panel absorber and the perforated panel absorber, can be used for two typical applications.

The first application consists of panels next to the drive-line of buses and trucks.

The second application consists of panels placed on the balcony of dwellings.

The absorbers are for both applications designed to have a resonance frequency that matches the frequencies of the traffic noise.

4.3.1 Panels next to drive-line

In order to decrease the noise radiated from buses and trucks to the façades of the buildings, the sound could be absorbed at the source: namely with panels next to the drive-line of buses and trucks.

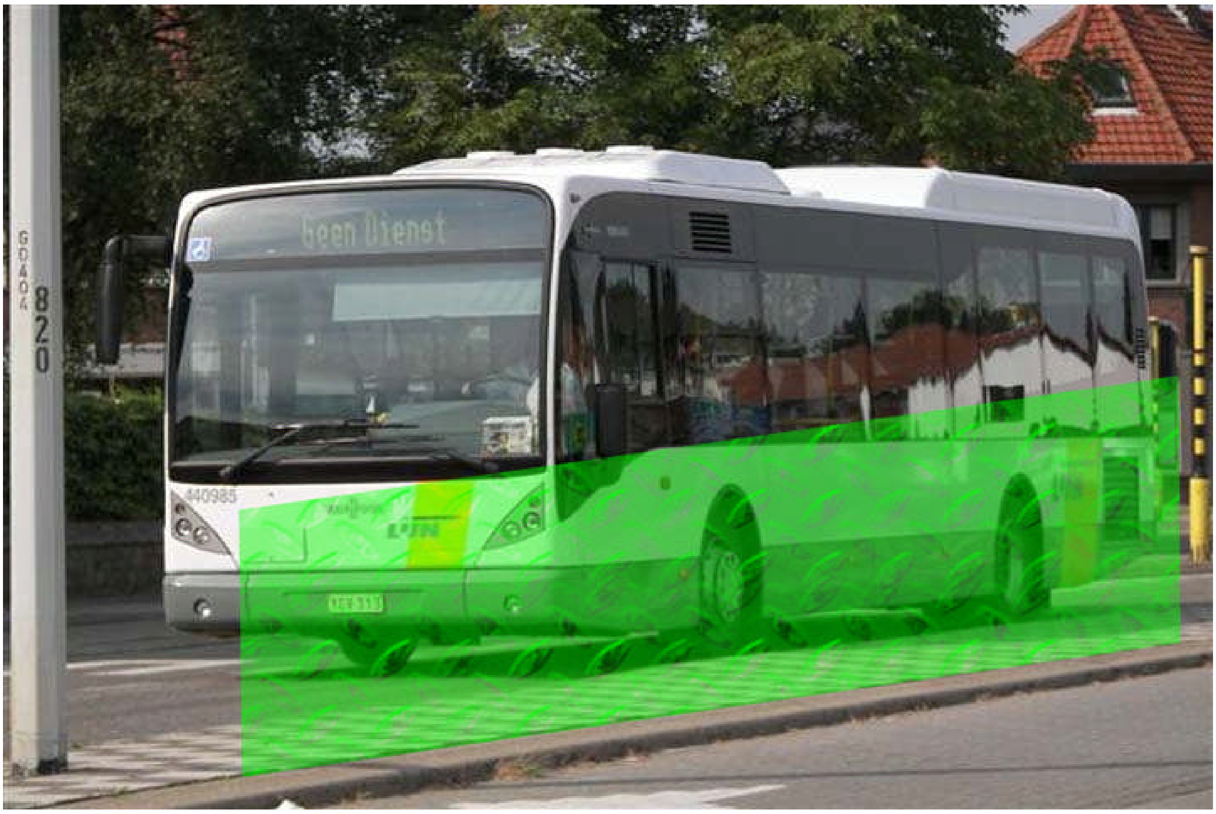


Figure 4.14 Panels next to drive line

4.3.2 Panels on balconies of dwellings

An application of both panel absorbers consists of placing them in front of balcony surfaces. See figure 4.15: incident sound rays (blue lines) reflect on parts of the balcony before they reach the window. The sound rays are absorbed by the panel absorbers installed against the balcony surfaces facing the building façade (green lines). Thus the low frequency noise from trucks and buses is prevented from entering the building through the windows.

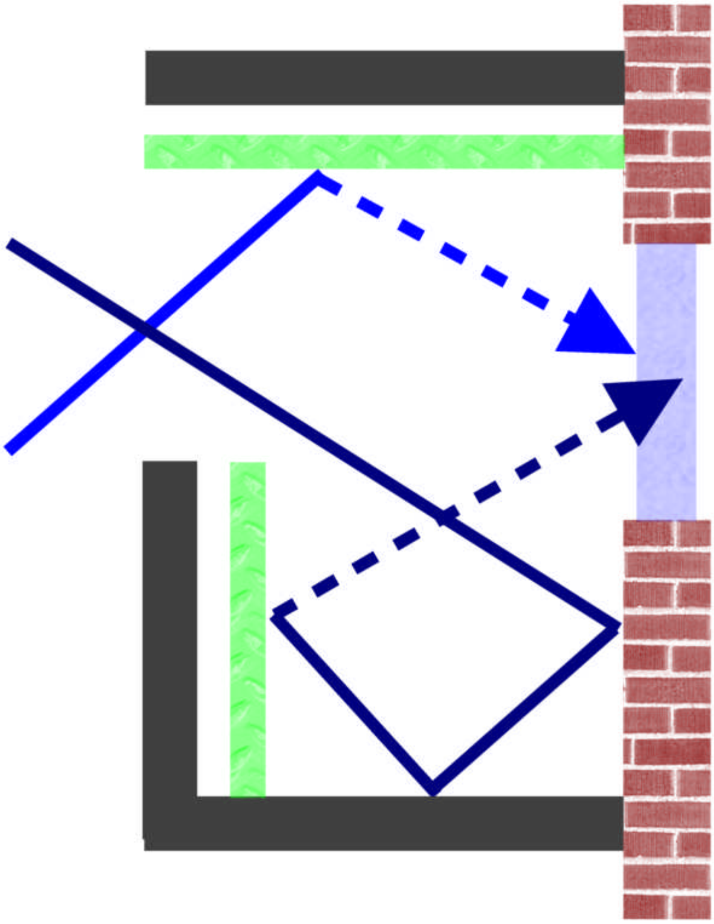


Figure 4.15

Sound rays reflecting on balconies

5 CONCLUSIONS

Trucks and buses are major contributors to traffic noise. At low speeds, the engine and exhaust typically produce low frequency noise (LFN) with dominant frequencies between 31,5 Hz en 63 Hz.

Commonly used window types do not perform well when it comes to low frequency sound insulation. Trucks and buses passing by at low speeds and at close proximity to building façades therefore generate noise inside the building with a high low frequency content.

In the CityHush project, two LFN absorbers for installation on the façade are developed to reduce the LFN around the exposed windows, so they will transmit less to the inside of the building: the solid panel absorber and the perforated panel absorber.

Both LFN absorbers are designed to have a resonance frequency that matches the frequencies of the traffic noise. The resonance frequency of the solid panel absorber depends on the mass of the panel and on the depth of the cavity behind. For the perforated panel absorber it is the combination of the perforations and the cavity behind that determines the resonance frequency. Both systems are known to be efficient absorbers for LFN.

Tables 5.1, 5.2 and 5.3 give possible dimensions for respectively an aluminium solid panel absorber and two perforated panel absorbers designed to absorb sound at frequencies of 31,5 Hz and 63 Hz.

f_r [Hz]	d [m]	Material	ρ [kg/m³]	t [m]	m'' [kg/m²]
31,5	0,90	Aluminium	2800	0,0014	4
63	0,30	Aluminium	2800	0,0011	3

Table 5.1 Solid panel absorber design

f_r [Hz]	l [m]	b [m]	d [m]	r [m]	perforation rate [%]
31,5	0,03	0,04	1,0	0,0025	1,1
63	0,03	0,04	1,0	0,0050	5,1

Table 5.2 Perforated panel absorber design 1

f_r [Hz]	l [m]	b [m]	d [m]	r [m]	perforation rate [%]
31,5	0,04	0,14	0,6	0,0080	1,1
63	0,04	0,14	0,6	0,0180	5,5

Table 5.3 Perforated panel absorber design 2

Both LFN absorbers, the solid panel absorber and the perforated panel absorber, can be used for two applications. The first application consists of panels next to the drive-line of buses and trucks. The second application consists of panels placed on the balcony of dwellings.

In the months M12 to M24, the prototypes of both LFN absorbers will be built and tested in the lab. The characteristics and dimensions of the absorbers will be verified and fine-tuned so that the LFN absorbers will resonate at the desired frequencies and to maximise the absorption ratio.

6 REFERENCES

[1] Nederlandse Stichting Geluidhinder., "NSG-Richtlijn Laagfrequent Geluid (in Dutch, Trans: "Guidelines for Low-Frequency Noise")", Nederlandse Stichting Geluidhinder, The Netherlands, (1999).

[2] Socialstyrelsen., "Buller Inomhus (in Swedish, Trans: "Low-Frequency Noise in Dwellings")", SOSFS 2005:6 (M) Allmänna råd Socialstyrelsen, Sweden, (2005).

[3] K.B. Ginn, M. Sc., "Architectural Acoustics", Brüel & Kjær, Denmark, (1978).

[4] W. Fasold, H. Winkler, "Bauphysikalische Entwerfslehre Band 5 Raumakustik", VEB Verlag für Bauwesen, Berlin, (1976).

[5] T.J. Schultz, "Acoustical uses for perforated metals: Principles and Applications", Industrial Perforators Association, (1986).

APPENDIX A: FIGURES MEASUREMENT CAMPAIGN

PROJECT: CityHush Comment: Bus pass by O
REC: rec15.dat
TIME: 16/02/2010 16:50:33

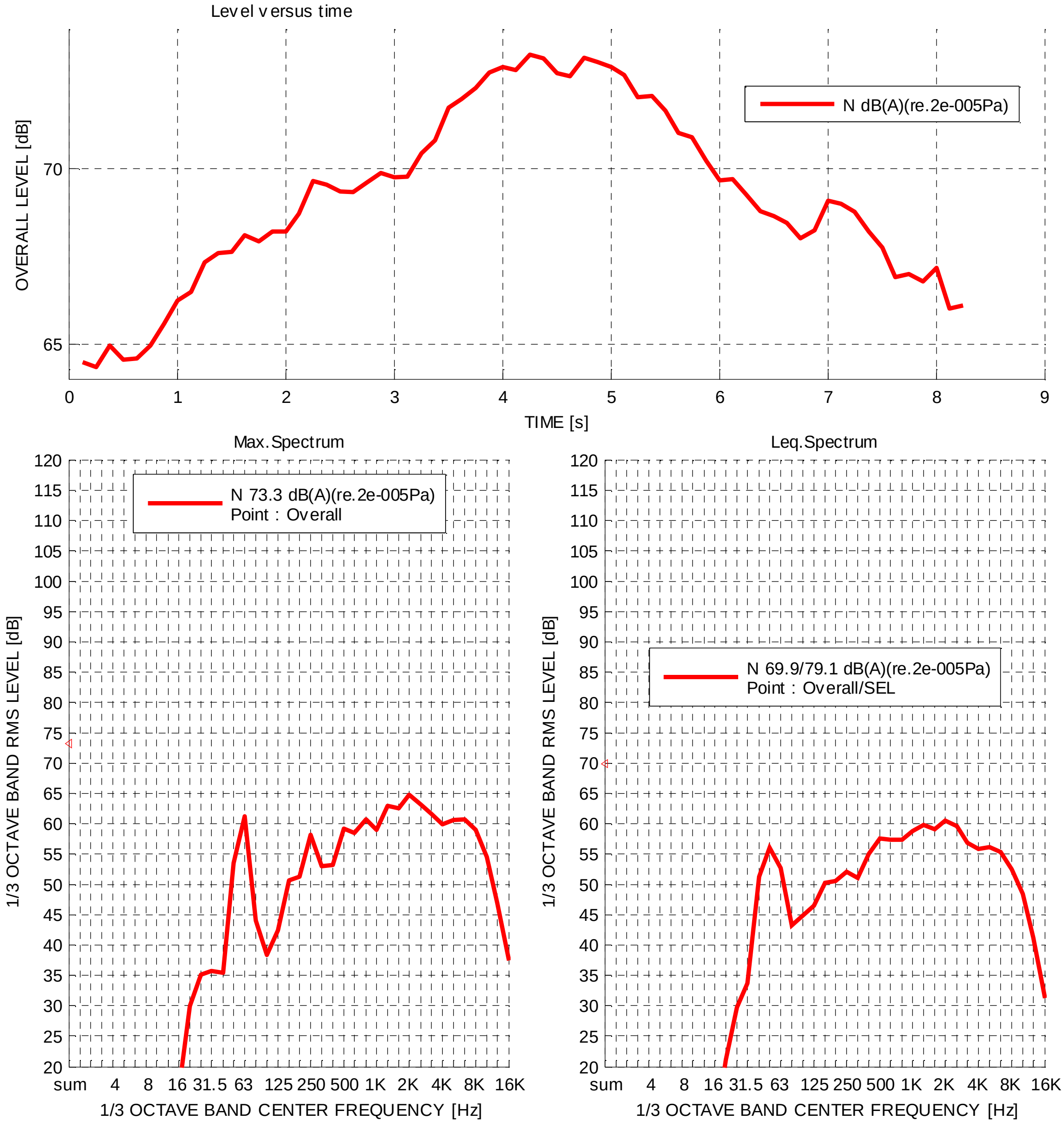


Figure A.1

Bus pass-by

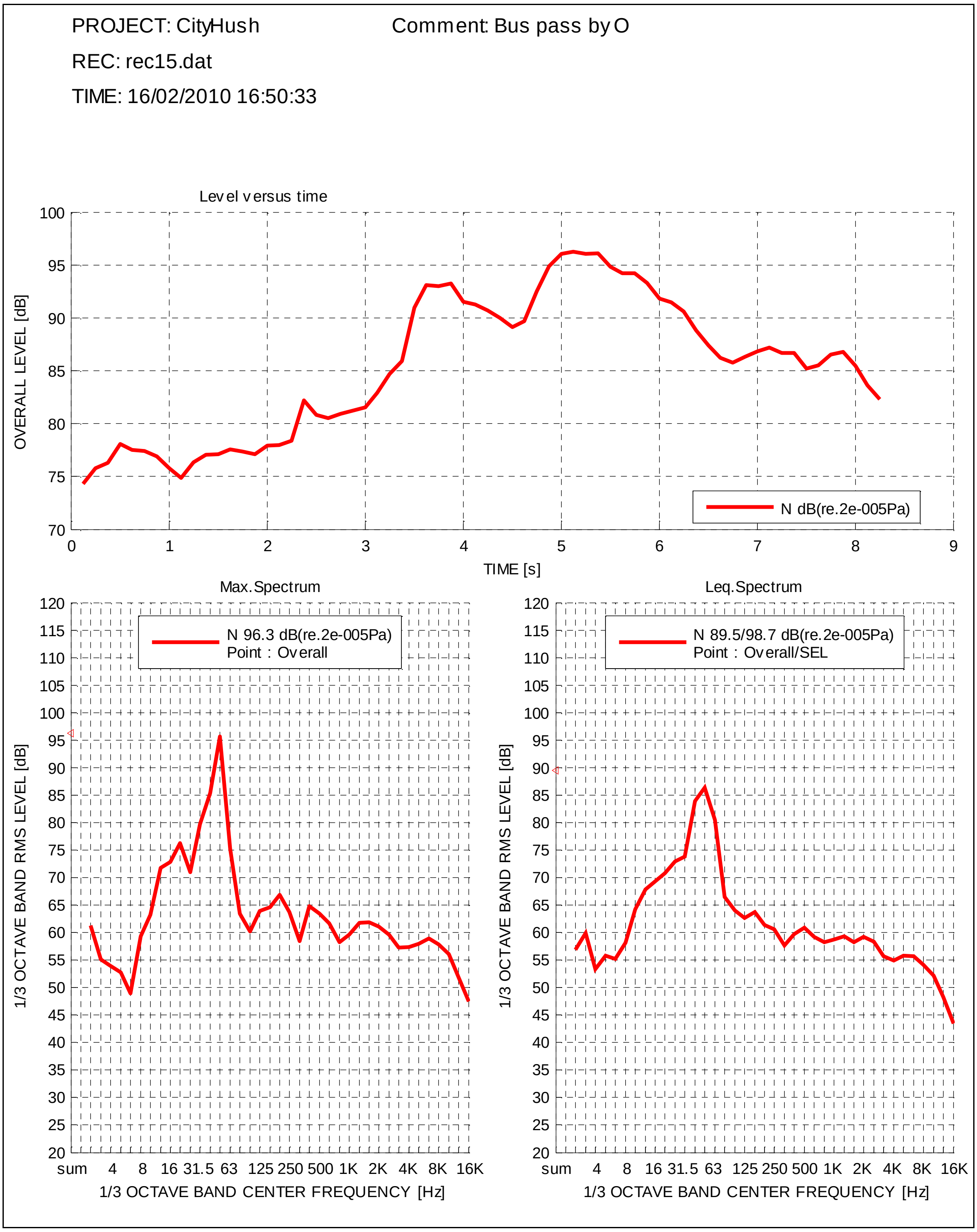


Figure A.2

Bus pass-by

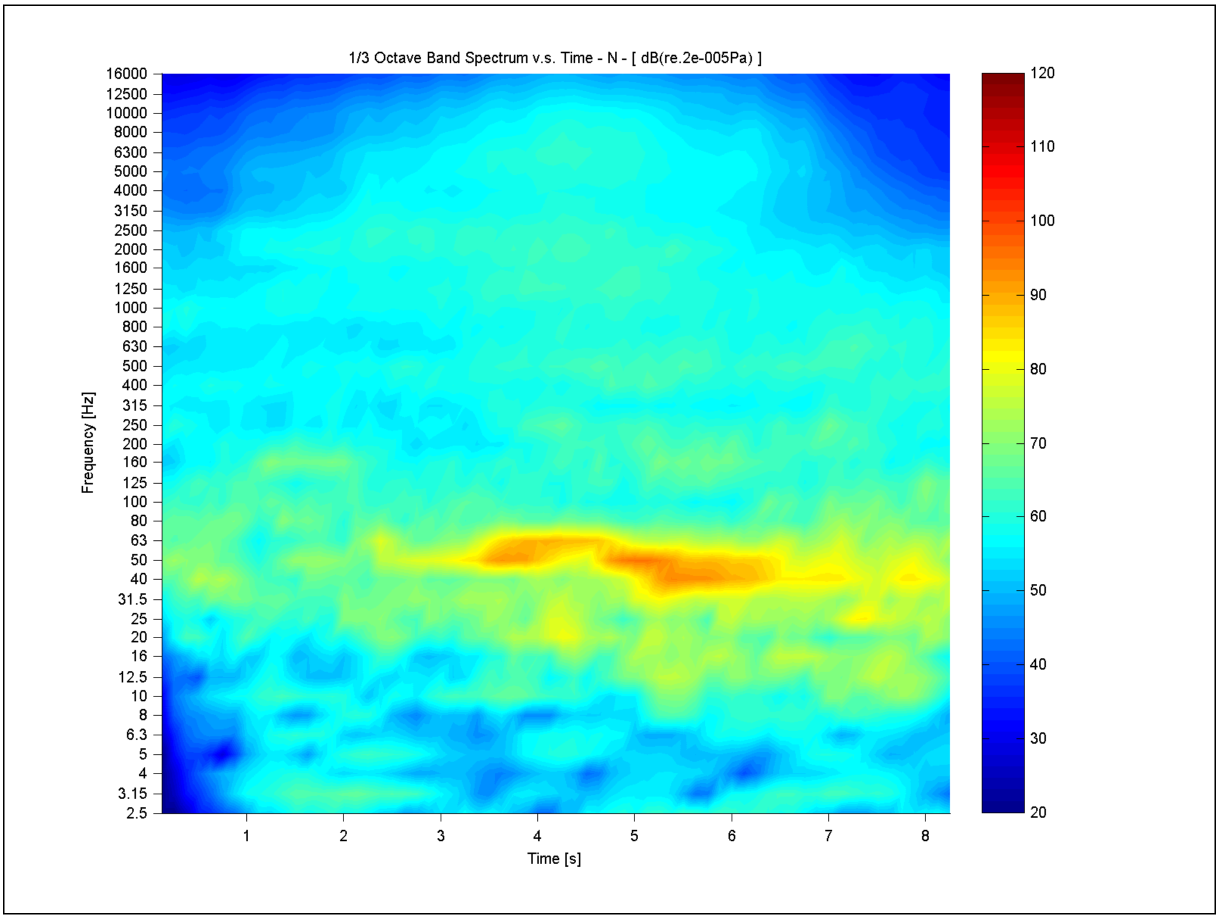


Figure A.3

Bus pass-by

PROJECT: CityHush

Comment: Bus idling O

REC: rec19.dat

TIME: 16/02/2010 16:56:25

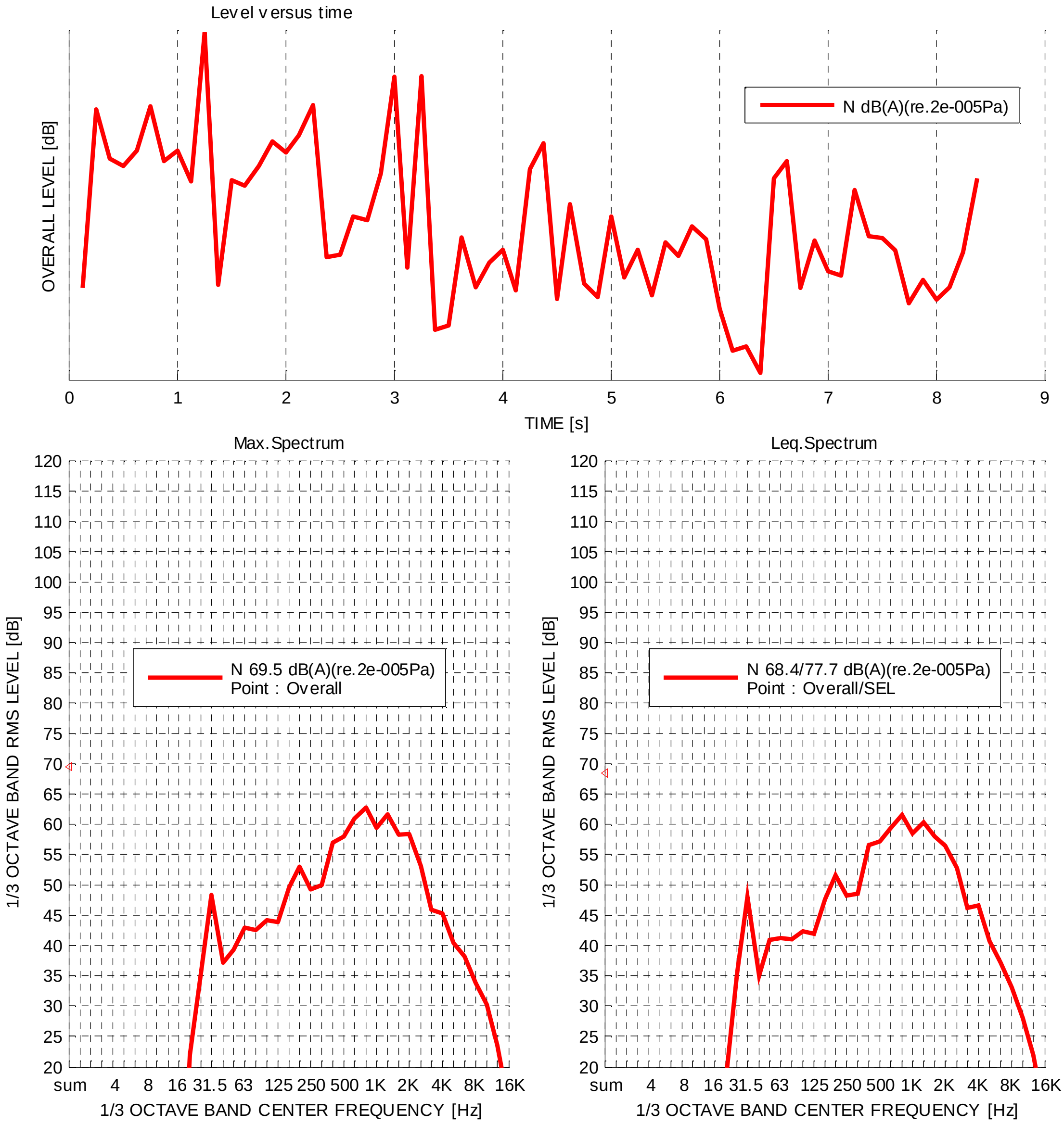


Figure A.4

Bus idling

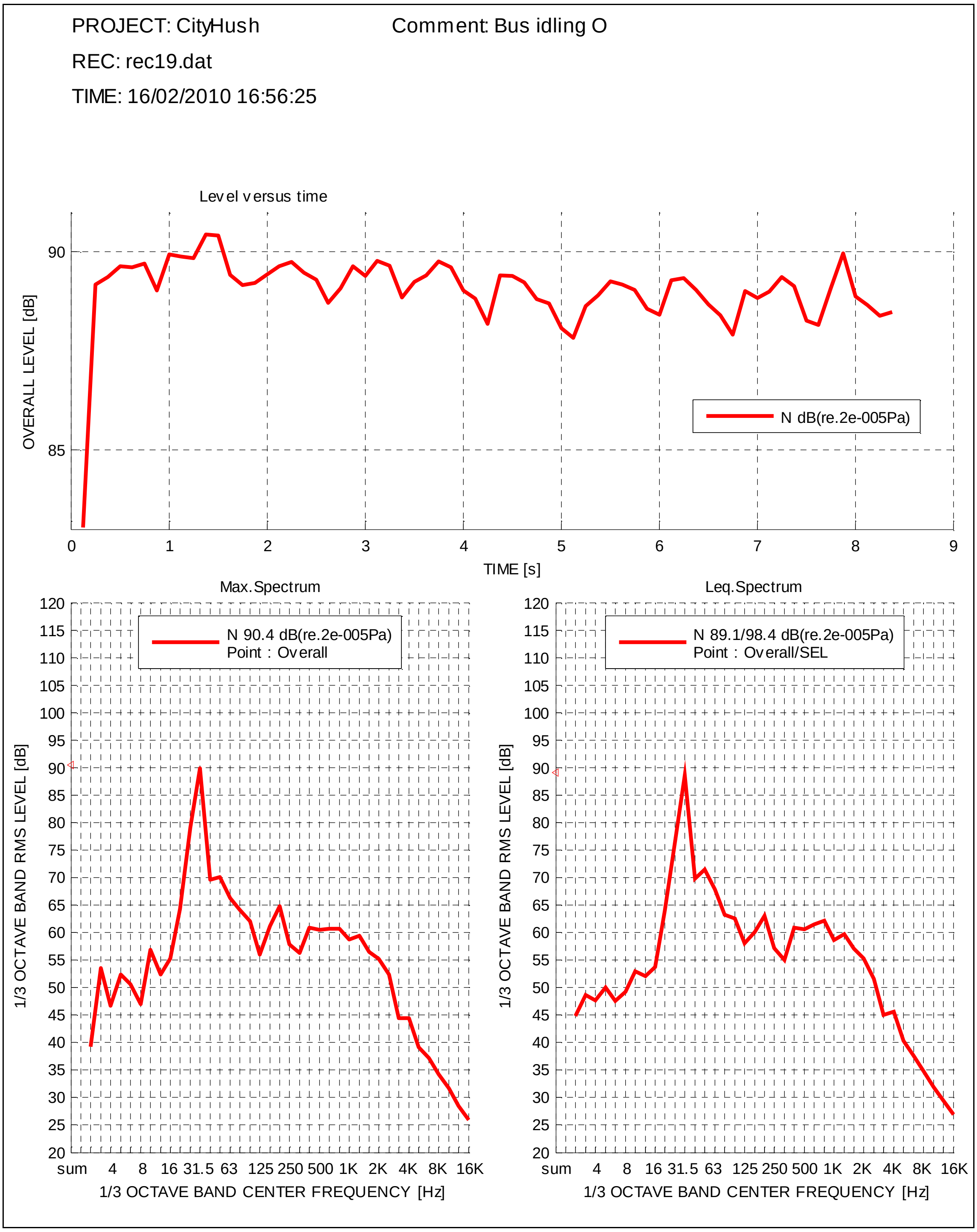


Figure A.5

Bus idling

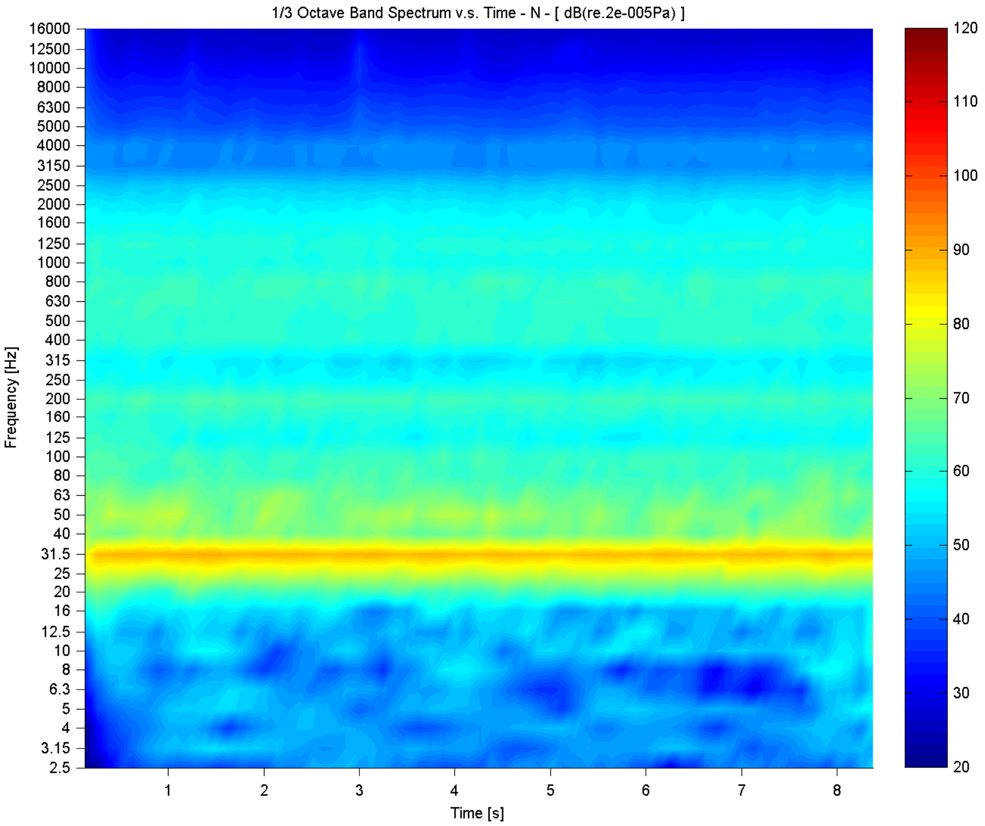


Figure A.6

Bus idling

PROJECT: CityHush

Comment: Departure bus O

REC: rec20.dat

TIME: 16/02/2010 16:56:54

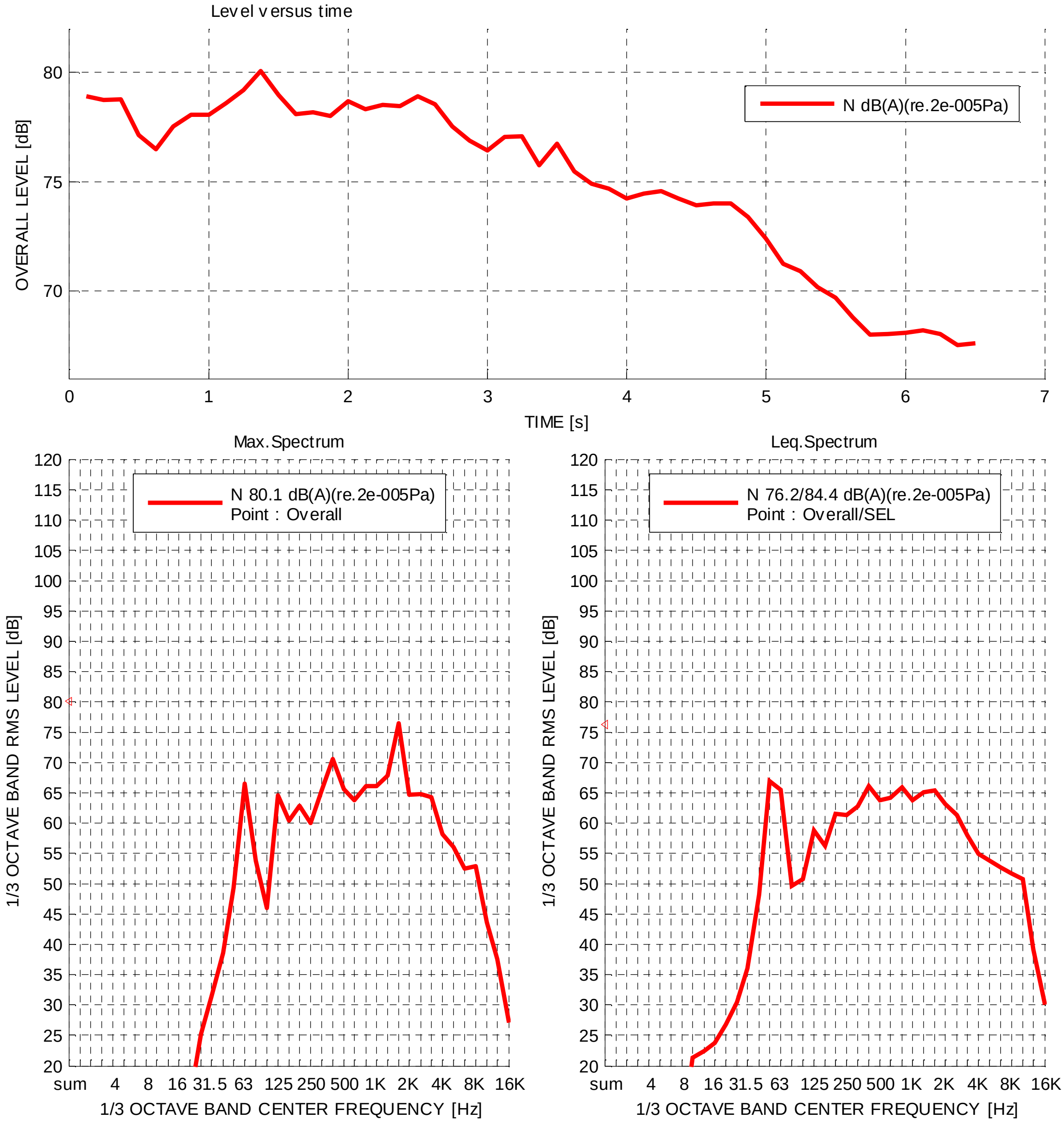


Figure A.7

Bus departure

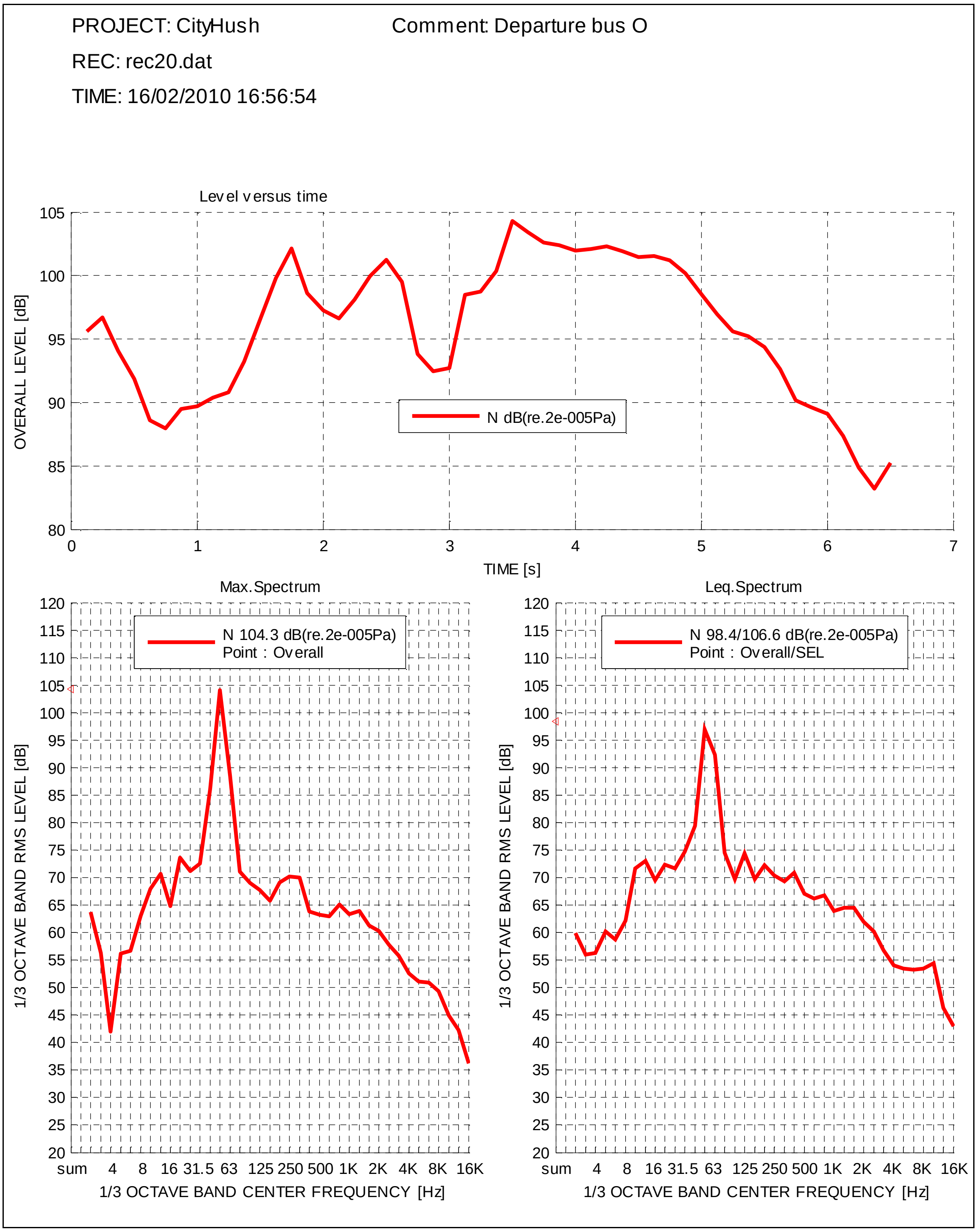


Figure A.8

Bus departure

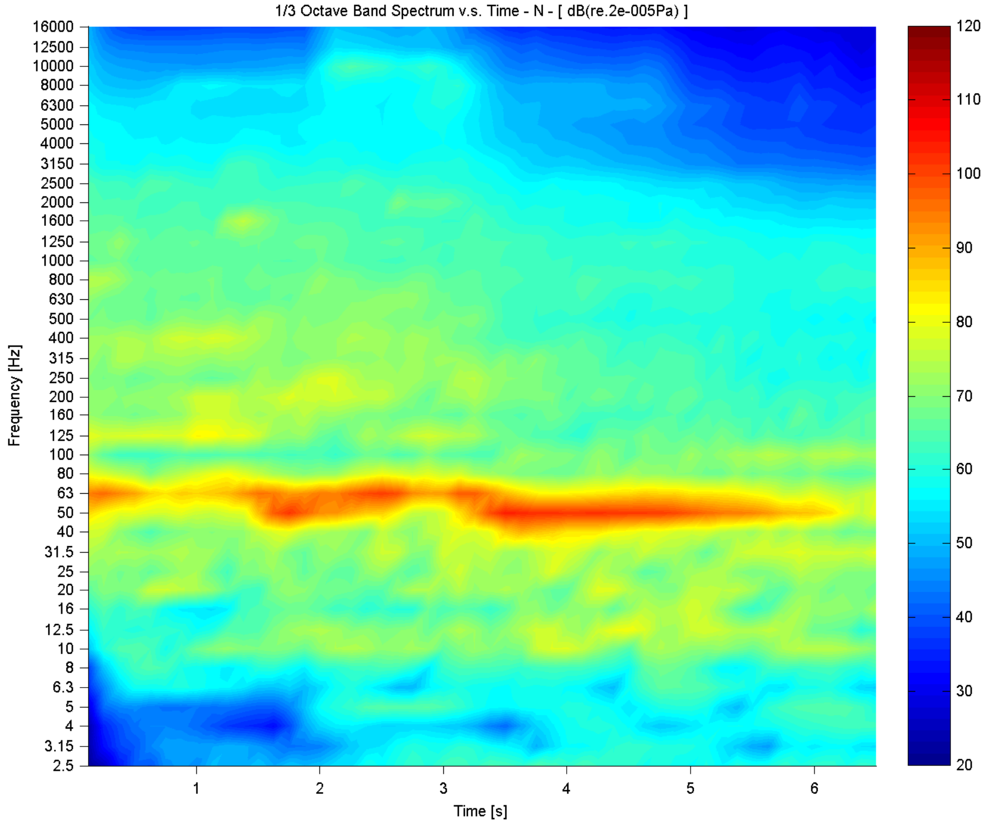


Figure A.9

Bus departure

PROJECT: CityHush
REC: rec11.dat
TIME: 16/02/2010 16:42:15

Comment: Arrival bus + idling + departure O

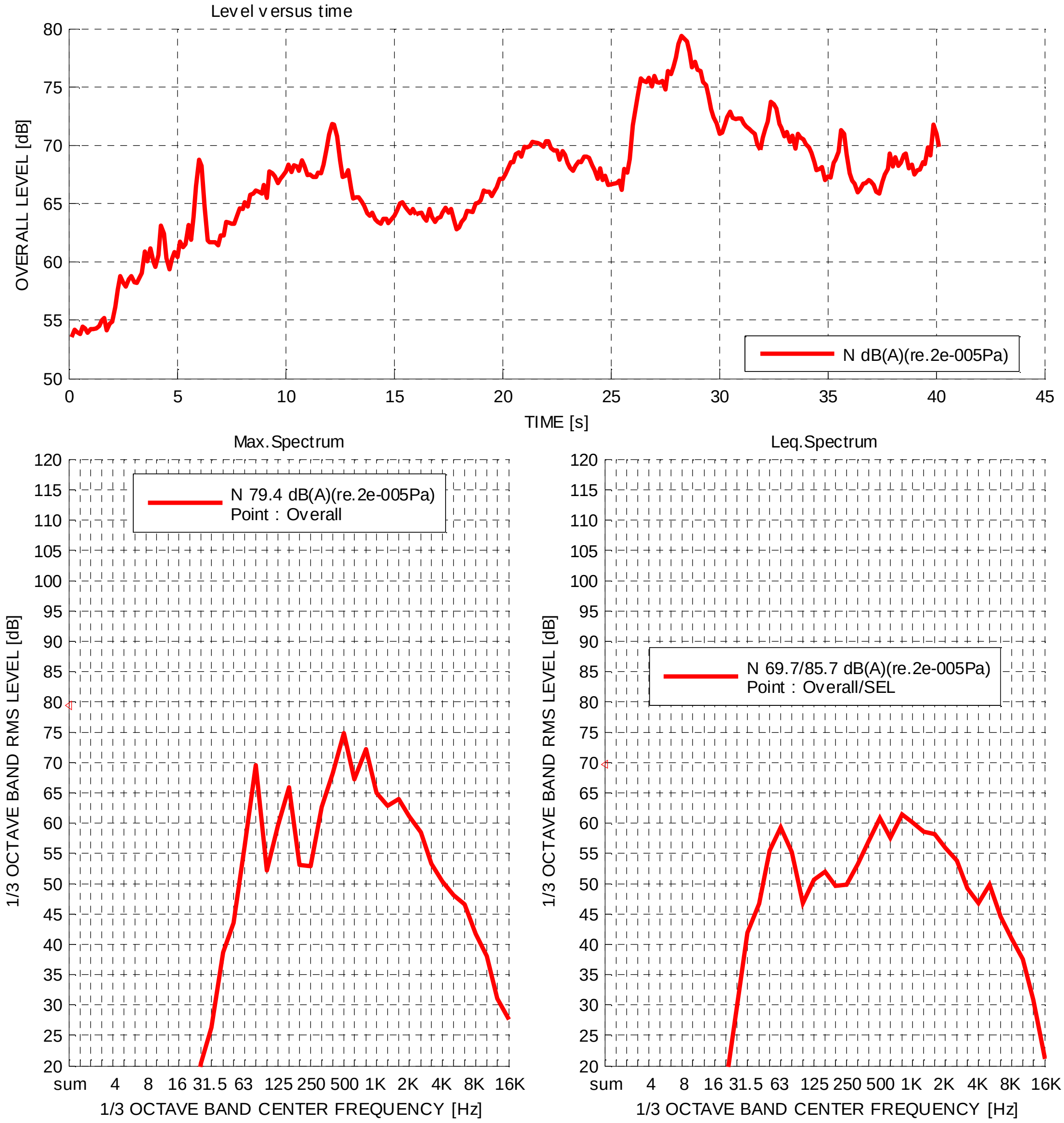


Figure A.10

Bus arrival + idling + departure

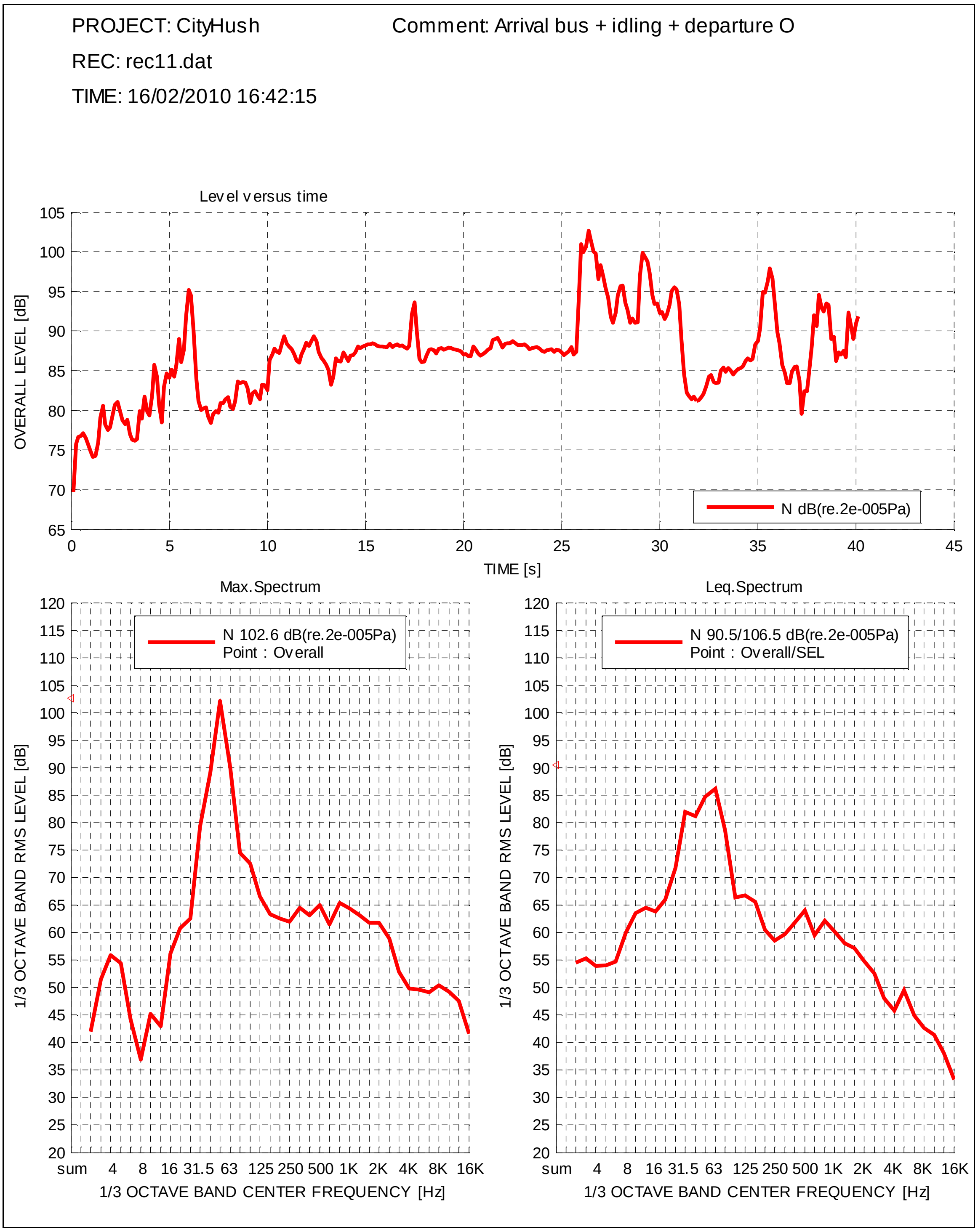


Figure A.11

Bus arrival + idling + departure

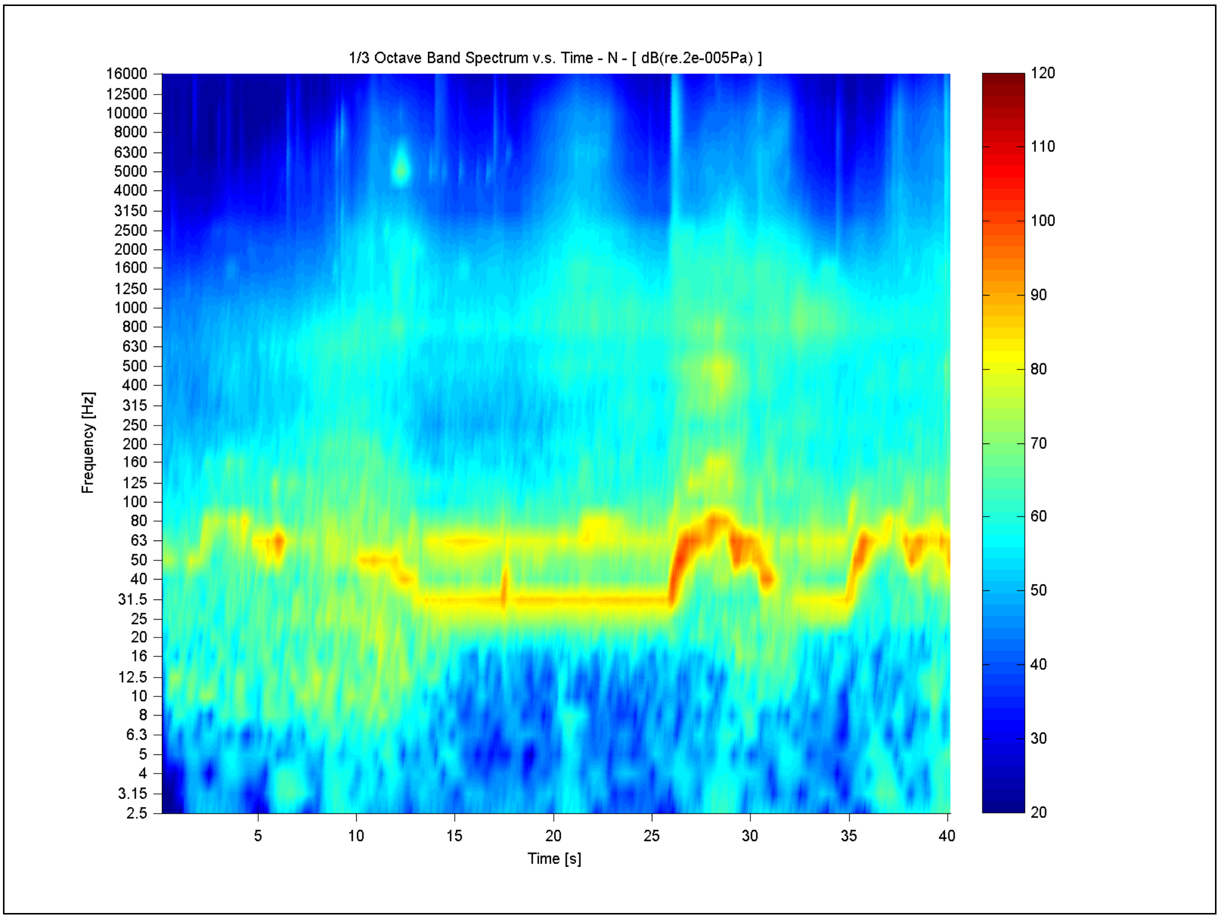


Figure A.12

Bus arrival + idling + departure

APPENDIX B: ABSORPTIVE MATERIAL

Mupan façade

Thermische isolatie van spouwmuren
Isolation thermique des murs creux

λ32 THERMAL INSULATION

PRODUCTOMSCHRIJVING

Harde glaswolplaat, langs de ene zijde bekleed met een zwart, kleurvast glasvlies en langs de andere kant bekleed met een geel Vetrotex® glasvlies.

TOEPASSING

Isover mupan façade is de optimale plaat voor een veilige spouwmuurisolatie.

PRODUCTEIGENSCHAPPEN

De eigenschappen van de Isover-producten gemeten volgens de STS 08.82.5 (ééngemaakte technische specificaties) worden gehomologeerd door de BUtgb onder het nr. ATG/H 557.

Thermische eigenschappen

λ_D = 0.032W/mK

Dikte (mm)	140	120	100	80	70	60	50	40
R_D (m²K/W)	4.35	3.75	3.10	2.50	2.15	1.85	1.55	1.25

Akoestische eigenschappen

Specifieke luchtweerstand : ongeveer 7x10³ Ns/m⁴

Brandveiligheid

Klasse A1 volgens NBN EN 13501-1
Klasse A0 volgens de oude norm NBN S 21.203

Overige eigenschappen

- Vochtgedrag
  - Niet capillair
  - Waterafstotend
  - Niet hygroscopisch
  - Weerstandsfactor bij waterdampdiffusie : μ=1,5
- Rotvrij
- Vormvast, geen verzakkingen
- Onaantastbaar door knaagdieren en micro-organismen
- Niet corrosief

DESCRIPTION PRODUIT

Panneau rigide en laine de verre, recouvert sur une face d'un voile de verre noir résistant aux intempéries, sur l'autre face d'un voile de verre jaune Vetrotex®.

APPLICATION

Isover mupan façade convient parfaitement pour isoler un mur creux en toute sécurité.

PROPRIETES PRODUIT

Les propriétés des produits Isover, mesurées conformément aux STS 08.82.5 (spécifications techniques unifiées) sont couvertes par l'Homologation suivie délivrée par l'UBAtc sous le numéro ATG/H 557.

Propriétés thermiques

λ_D = 0.032W/mK

Epaisseur (mm)	140	120	100	80	70	60	50	40
R_D (m²K/W)	4.35	3.75	3.10	2.50	2.15	1.85	1.55	1.25

Propriétés acoustiques

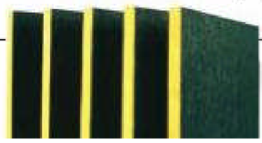
Résistance spécifique à l'air environ 7x10³ Ns/m⁴

Sécurité au feu

Classe A1 selon NBN EN 13501-1
Classe A0 selon l'ancienne norme NBN S 21.203

Autres propriétés

- Comportement à l'humidité
  - Non capillaire
  - Hydrofugé
  - Non hygroscopique
  - Facteur de résistance à la diffusion de vapeur d'eau : μ=1,5
- Imputrescible
- Dimensionnellement stable
- Inattaquable par les rongeurs et micro-organismes
- Non corrosif



ISOVER

Figure B.1

Data sheet for the *Isover Mupan Façade* material