Acoustics in sports halls

absorbing behaviour of perforated steel panels

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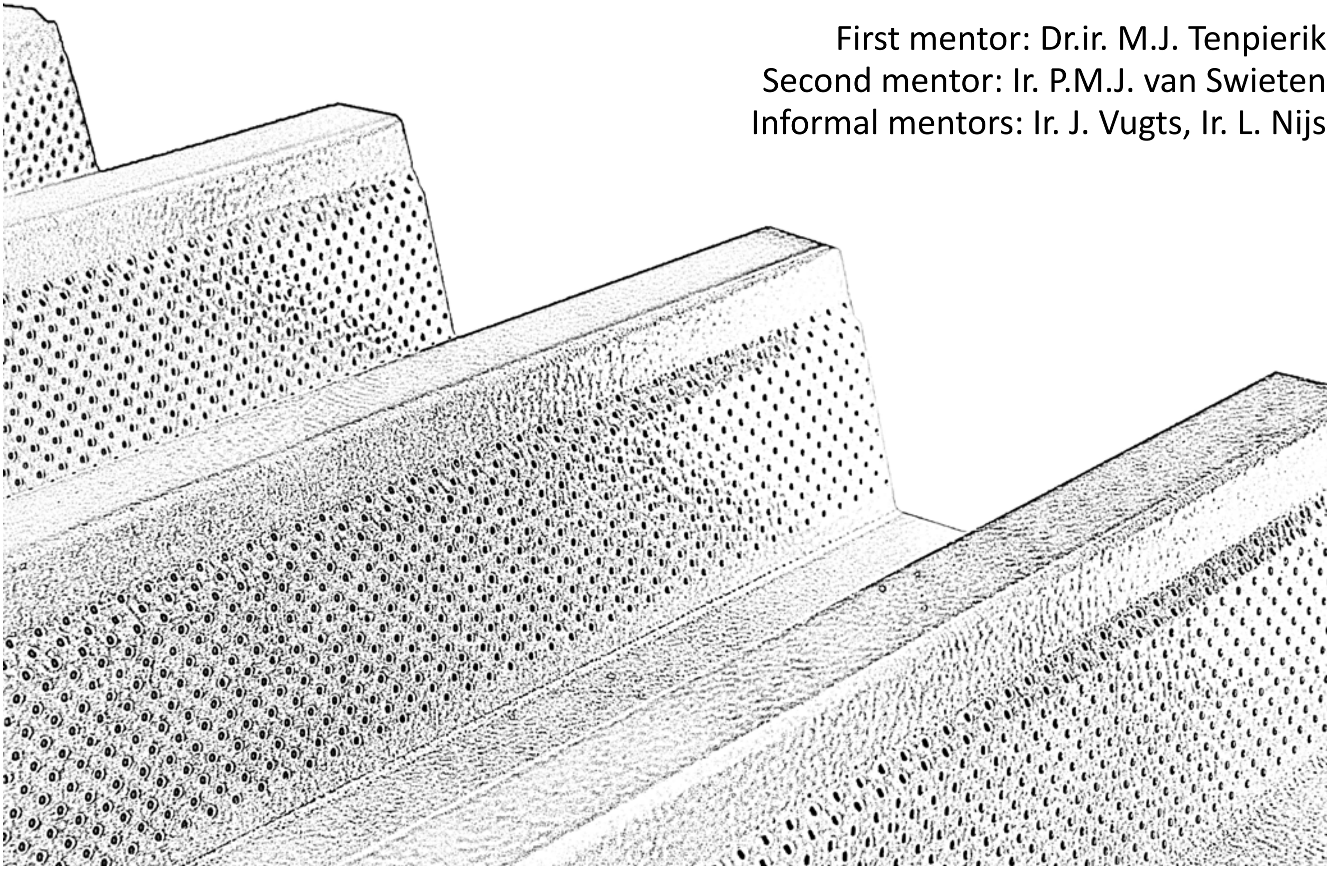

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Preface

When I decided acoustics would be my graduation direction, I had no idea of the research I wanted to do. Jeroen Vugts invited me at LBP|SIGHT to talk about some interesting topics. At the end of the conversation, after a lot of nice ideas, we talked about a problem he had found while measuring the reverberation time of many sports halls in the Netherlands. Namely, that perforated steel panels often used in sports halls, seem to behave differently than expected from laboratory results.
I found it an interesting topic and after a meeting with Martin Tenpierik, I decided to make it my graduation topic. Because of the time limit, this graduation research mainly focuses on the sound absorbing behaviour of perforated steel panels, often used on walls or as roof of sports halls.

Since the start in October, I have worked with a lot of enthusiasm and energy on the project. To be honest, I am proud of the final result.

With many thanks to: Martin Tenpierik for being my inspired first mentor, Peter van Swieten for being my second mentor, Jeroen Vugts of LBP|SIGHT for being my very helpful informal third mentor, Lau Nijs for being a great help with all of his expertise, LBP|SIGHT for providing the laboratory, the laboratory employees, Jeroen Neggers of ISA Sport for providing several useful standards and last but not least my family, friends and colleagues for their support.

Sponsors:
SAB: André de Jongh. (Perforated roof panels type SAB 106R+/750 P3L-B and SAB 106R+/750 P4L-B.)
Rockwool: Roger Peeters. (Cannelurevulling and Rhinox roof insulation panels.)

This report is created with the use of the graduation report of Eline Nesselaar, Sander Uittenbosch and Peter Koers.

Yvonne Wattez
Delft, June 2012

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Summary

Problem statement

Sports teachers (and other users) often suffer from bad acoustics in sports halls. A survey revealed that most complaints are about tiredness, throat problems and hearing problems. Bad acoustics in sports halls are mainly caused by an uneven distribution of the sound absorbing material in a hall. Often, the roof absorbs most of the sound where the floor and walls up to 3 meter reflect most of the sound. This lower part of the walls needs to be strong, flat and hard to rebound balls. In this way, vertically reflected sound is being absorbed faster than horizontally reflected sound. This difference causes sagging Schroeder curves and therefore long reverberation times. Dutch standards for acoustics are mainly based on reverberation times, so sagging curves are unwanted.

Besides the problem of the long reverberation times, another problem is found. Measurements show that sports halls constructed with stone-like materials absorb sound according to the expectations of the material properties. Sports halls constructed with perforated steel sound absorbing panels give different results. These panels seem to behave differently on sound absorption than expected. Especially low frequencies seem to be absorbed extremely well by the panels. The research question therefore is: *'Why do perforated steel sound absorbing panels seem to behave differently in absorption coefficient in sports halls than expected from laboratory test results?'* Because this question is part of the larger subject 'acoustics in sports halls', it is necessary to investigate the acoustical behaviour of perforated steel panels for a broad demarcation of the subject.

Approach

In order to find an answer to the research question, two hypotheses are tested by measurements. The measurements are done in a laboratory and in a scale model. The results are analysed. The conclusions give insight in the sound absorbing behaviour of perforated steel sound absorbing panels. Besides that, the conclusions give guidelines to the design phase of this graduation project.

Hypotheses:

1. A perforated steel panel behaves differently in practice than in a laboratory situation on absorption coefficient because the shape of the panels causes sound absorption of parallel striking sound based on a

phase shift principle.
2. A perforated steel panel behaves differently in practice than in a laboratory situation on absorption coefficient because the backing construction has influence on the result.

Laboratory measurements
Absorption coefficient measurements are done in the reverberation room of the acoustical laboratory. Different samples of the roof structure are tested. By comparing the results, we get insight in the influence of different parts of the roof structure. Besides that, hypothesis one can be tested: the influence of the backing construction will become clear.

Scale model measurements
The scale model makes it possible to test the influence of different shaped roof structures in a small model. By using the same, reflecting material for all walls, we get insight in the influence of changes in roof shapes. Some of the roof structures are (partly) covered with a sound absorbing material to compare the results of those to the other variations. This research should give an answer to hypothesis two.

Results
The scale model measurement results show that profiled structures cause a considerable decrease in local sound pressure compared to a hard, flat surface. This decrease is largest for low frequencies.
The surface with sound absorbing material on top causes the biggest decrease, which is caused by the sound absorption of the facing material. The 'sound absorption by shape' is caused by the phenomena of diffusion and interference. It is not visible in reverberation times (T20), just in histograms and Schroeder curves.

The laboratory measurements show that the influence of the thermal insulation layer; a part of the backing construction, is large on (low frequent) sound absorption. The rock wool panels give a high peak in the sound absorption graph for 100 Hz. This very good sound absorption is probably caused by the panel being its own panel-resonator and porous absorber in one. The influence of the 'cannelurevulling' is small, like the influence of the vapour barrier and different perforation degrees.

Conclusions

Perforated steel sound absorbing panels absorb more low frequent sound than expected. This good sound absorption is probably achieved by the profiled shape of the steel panels in combination with the backing thermal insulation.

The profiled shape causes the effect of sound absorption caused by the effects of diffusion and interference. The backing construction absorbs a large part of the low frequent sound because the hard, stiff and heavy rock wool panels are probably a porous material and panel-resonator in one.

The research does not give strong guidelines to improve the roof structure. The gained knowledge is used in the design of a wall panel. The walls up to three meters are still the weakest link when talking about acoustics in sports halls. The designed panel should improve this situation.

I. The problem

1. Introduction

Acoustics

Acoustics is the interdisciplinary science that deals with the study of all mechanical waves in gases, liquids and solids, including vibration, sound, ultrasound and infrasound. The architectural acoustics field can be subdivided too. This graduation project deals with a part of the room acoustics theory: it includes acoustics of concert halls, theatres, auditoria and sports halls.

Acoustics in sports halls

The acoustics of sports halls are designed by an architect, often with the help of an engineer. Since 2008, acoustics in sports facilities is hot news. The standards for those facilities are based on the reverberation time. As found in some literature, the reverberation time is a constant, easy to measure, but possibly not very useful parameter to use for sports halls. The differences in reverberation time of the designed situation and the realized situation are still big. This proves why it is not easy to design acoustics.

Sports halls

There are different types of sports facilities. In this graduation project, three different types are mentioned. A gymnasium is a small sports facility, mainly used by primary schools. A gym is a 'normal' sports hall, with one hall that is not dividable. A sports hall is a big sports facility that can be used as one big hall, but also as two or three different smaller gyms by dividing it with descending flexible walls.

The sizes of a gymnasium, gym, and sports hall differ between: 12 x 21 x 5,5 meter and 28 x 70 x 9 meter. (ISA-Sport 2002) Nowadays, these types have different standards for reverberation time depending on the volume of the room.

2. Description of the problem

The overall problem on which this graduation is based, consists of three parts:

1. Complaints from users
2. Designing acoustics
3. The acoustical behaviour of perforated steel panels

2.1. Complaints from users

In 2010, teachers and employees of a sports hall in Rijssen-Holten got publicity because of a protest action where all students were wearing earmuffs while sporting. By this action, the employees tried to get attention for the bad acoustics in sports halls which, in their opinion, is the cause of their hearing problems.

The KVLO, the 'Koninklijke Vereninging voor leraren Lichamelijke Opvoeding' started a survey among PE teachers to find out whether there were complaints or not. This survey resulted in about 250 complaints possibly related to acoustics in sports halls. Most complaints were: throat problems, tiredness and hearing problems. (Zandstra 2010)

The complaints from PE teachers are mainly caused by the acoustics of a sports hall. Because of the hard materials used in sports halls, reflections of sound can be heard. Reflections from the walls will arrive later than the direct sound. This difference can help the teacher because it can increase the sound level. On the other hand, it can cause complaints: the late reflections will arrive 'too late' and they will overlap the next syllable of the word. Then, the speech intelligibility decreases. This speech intelligibility is a term often used in room acoustics. It depends on the level of background noise, the reverberation time and the shape of the room. Besides that, it also depends on the person speaking and the person listening.

Background noise level

Although the main issue seems to be the reverberation time measured in sports halls, also the background noise needs to be discussed. Sander Uittenbosch (2009) found that the background noise level measured in different sports halls is too high. This parameter should not be ignored when solving the problems.

Speech intelligibility

The low speech intelligibility in sport halls may be linked to the complaints of PE teachers. Those are mainly: throat problems, tiredness and hearing problems. That sports sound is the reason for hearing problems is hard to prove. Though, the other two complaints (throat problems and tiredness) are closely related to the acoustics. The teachers need to shout to make themselves heard. This can cause throat problems. The tiredness is mainly caused by the continuous sounds and echoes and the need to speak out loud all day.

2.2. Designing acoustics

This section will explain in short why designing acoustics is not easy. This section will give some information about different tools that are available when designing acoustics in sports halls, given below. It is a short summary of an investigation of two sports halls. The whole case study can be found in Attachment C.

1. Standards
2. Formula's
3. Computer programmes
4. Measurements of comparable situations

2.2.1. Standards

Almost all standards for acoustics in rooms are based on the reverberation time in the room and the background noise level. Standards for sports halls are comparable. Wether this is a good way to define acoustics of a sports hall, is the question. For more information see chapter 3.

Legislation and standards for acoustics in sports halls are relatively new. Standards for this type of rooms exist since the last decade. Attachment A gives more information about the history of the legislation and standards.

ISA-Sport 2005
The newest version of the standard: 'Standards gymnastics and sports halls and parts of sports halls with educational use.' is used in The Netherlands since 2005. This standard is also called: ISA-US1-BF1. It is created by ISA-Sport.

The standard is based on the reverberation time because a better option is not found yet. With the help and advice of some experts, the reverberation time is defined per volume of the hall. This is important, because a sports hall twice the size should not have twice the reverberation time. The very big sports halls may have a very long reverberation time in that case. The acoustical quality is not linear with the volume. Besides the advise to introduce the volume in the standard for reverberation time, the average absorption coefficient was advised to be at least 0,25. (Nijs 2004)

The ISA-Sport standard includes: information on the location, the sports equipment, dressing rooms, and acoustics of the sports facility:

- The average absorption coefficient [alpha] of the materials in the sports hall has to be at least 0,25.
- The reverberation time depends on volume and absorbing behaviour of the room. The average reverberation time in frequency band of 125-4000 Hz may not be higher than 1,0 seconds for a sports hall of 14 x 22 x 5.5 m.
- The reverberation time per frequency band (Tmax/fb) is calculated by Tav divided by Tmax/fb. This has to be or greater than 0,7.
- The background noise level must not be higher than 40 dB(A). This applies to external sounds like outdoor traffic and internal sound sources like installation systems.
- The noise insulation index between rooms for physical education and other residences/ classrooms should be 10 dB(A), preferably 15dB(A).

The investigated sports halls are of type C1: 24 x 44 m.

Kind of room		**Size [m] (w x l x h)**	**Reverberation time [s]**
A.1	Gymnastics	≤ 14 x 22 x 5.5	≤ 1,0
A.2	Gym	13 x 22 x 7	≤ 1,1
A.3	1/3 sports hall /gym	14 x 24 x 7	≤ 1,2
B.1	Gym	16 x 28 x 7	≤ 1,3
B.2	Gym	22 x 28	≤ 1,4
B.3	2/3 sports hall	32 x 28	≤ 1,5
C.1	Sports hall	24 x 44	≤ 1,6
C.2	Sports hall	28 x 48 x 7	≤ 1,7
C.3	Sports hall	28 x 48 x 9	≤ 1,9
D.1	Sports hall	28 x 88 x 7	≤ 2,0
D.2	Sports hall	35 x 80 x 10	≤ 2,3

2.2.2. Formulas

When designing reverberation times, different formulas can be used. Most formulas use the term 1/6. This could be replaced by the more precise 55,3/c or 0,161. The correction for air attenuation '4mV', as described in Sabine, needs to be used for big rooms in combination with Sabines formula (or other formulas).
The Dutch standard NEN-12354-6 describes a calculation model to estimate the total equivalent sound absorption area or reverberation time of enclosed spaces in buildings.

Sabine

Reverberation time formula based on a cube. This formula is world-wide used to design acoustics.

$$RT = \frac{1}{6} \cdot \frac{V}{A + 4mV}$$

with: RT= reverberation time [s]
V= volume of room [m^3]
A= total absorption of materials in room = $\sum(\alpha_i S_i)$
with: α_i = absorption coefficient of element i [-]
S_i = surface of element i [m^2]
$4mV$ = correction for air attenuation

Sabine corrected by Lau Nijs

This correction is created because Sabine is based on a cube. Sports halls normally have a rectangular shape. By replacing the surface factor S by a certain part of the volume $V^{2/3}$, this formula becomes more realistic.

$$RT = \frac{1}{6} \frac{V}{\bar{\alpha} \cdot 6V^{2/3}}$$

with: RT= reverberation time [s]
V= volume of room [m^3]
$\bar{\alpha}$ = average absorption coefficient

Eyring 1

This formula of Eyring works with the mean value of all absorption coefficients.

$$RT = \frac{1}{6} \frac{V}{S \cdot \ln\left[(1-\bar{\alpha})^{-1}\right]}$$

with: RT= reverberation time [s]
V= volume of room [m^3]
S= total surface [m^2]
$\bar{\alpha}$ = mean value of all absorption coefficients in the room = $\frac{\sum(\alpha_i S_i)}{\sum S_i}$ [-]

Eyring 2

A different formula of Eyring that works with the frequency dependent absorption coefficient.

$$RT = \frac{1}{6}\frac{V}{\sum S_i \cdot \ln\left[(1-\alpha_i)^{-1}\right]}$$

with: RT= reverberation time [s]
V= volume of room [m^3]
S_i = surface of element i [m^2]
α_i = absorption coefficient of element i [-]

Kosten

Specifically developed for concert halls and theatres.

$$RT = \frac{1}{6}\frac{V}{S_s \cdot \alpha_{eq}}$$

with: RT= reverberation time [s]
V= volume of room [m^3]
S_s= by audience occupied surface [m^2]
α_{eq}= equivalent absorption coefficient of this surface [-]

Fitzroy

$$RT = \frac{x}{S}\frac{kV}{-S \cdot \ln\left(1-\alpha_{x,av}\right)} + \frac{y}{S}\frac{kV}{-S \cdot \ln\left(1-\alpha_{y,av}\right)} + \frac{z}{S}\frac{kV}{-S \cdot \ln\left(1-\alpha_{z,av}\right)}$$

with: RT= reverberation time [s]
k= constant = 55,3 /c [s/m]
with: c= velocity of sound in air = $20{,}1\sqrt{(273+t)}$
with: t= temperature
V= volume of room [m^3]
S= total surface [m^2]
x= total surface side walls [m^2]
$\alpha_{x,av}$= average absorption coefficient of side walls [-]= $\frac{\sum(\alpha_{x,i}S_{x,i})}{\sum S_{x,i}}$ [-]
y= total surface crosscut walls [m^2]
$\alpha_{y,av}$= average absorption coefficient of crosscut walls [-]= $\frac{\sum(\alpha_{y,i}S_{y,i})}{\sum S_{y,i}}$ [-]
z= total surface floor/ceiling [m^2]
$\alpha_{z,av}$= average absorption coefficient of floor/ceiling [-]= $\frac{\sum(\alpha_{z,i}S_{z,i})}{\sum S_{z,i}}$ [-]

2.2.3. Computer programmes

Besides formulas, computer programmes can be a useful tool to design acoustics too. In the comparison of the two halls (Attachment C), the programmes ODEON and Catt Acoustic are used. An important input value is the scattering coefficient, which gives information about the roughness of the surface and so the diffusion of the sound. This parameter is often filled out with a standard value of 0,1 but it is very important for the results. For example: the higher the scattering coefficient, the shorter the reverberation time.

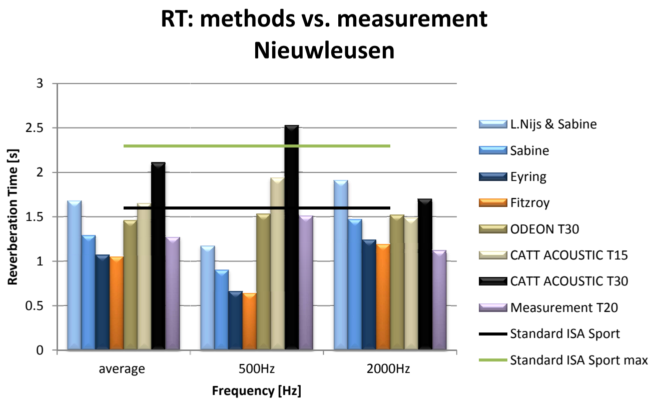
2.2.4. Measurements of comparable situations

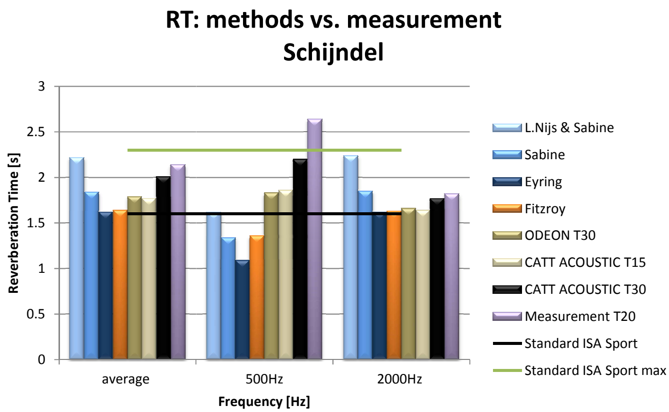
Finally, measurements of comparable situations can improve the design.

2.2.5. Comparison of all tools

The graph below shows reverberation times for different frequency bands for two sports halls. The measured reverberation time is shown on the right in purple. A quick look is enough to see that all tools give different results on the same hall. This is in short the problem of designing acoustics. It is just not easy to find a suitable tool and to use it in a appropriate way.

For the whole investigation of the use of different tools and parameters in practise, see Attachment C.





2.3. The acoustical behaviour of perforated steel panels

The last part of the bigger problem; acoustics in sports halls, is the part of the material properties. The acoustical behaviour of materials used in sports halls is very important for the acoustical quality of the hall. As in Sabines formula is written: the reverberation time depends for a large part on the amount of sound absorbing surface in the room.

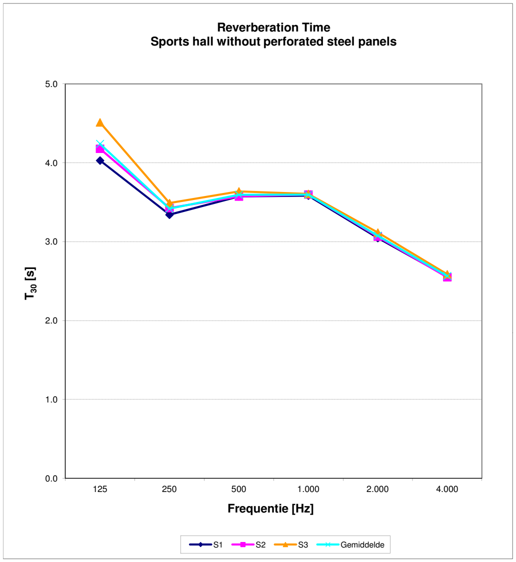
This graduation report explains the story of acoustics in sports halls by investigating the sound absorbing properties of a perforated roof structure. Why are those properties interesting for the total problem? This chapter explains why it is important to understand the differences between halls constructed with stone-like materials and halls constructed with perforated steel panels.

Demarcation of the subject

The acoustical problem found in sports halls is a relatively new topic. Therefore, it is important to understand as many phenomena as possible according to the topic. Since something 'strange' is found by comparing many sports halls, it is interesting to understand why the results are this 'strange'. In this way the demarcation of the subject is a tool to improve the acoustics of a sports hall.

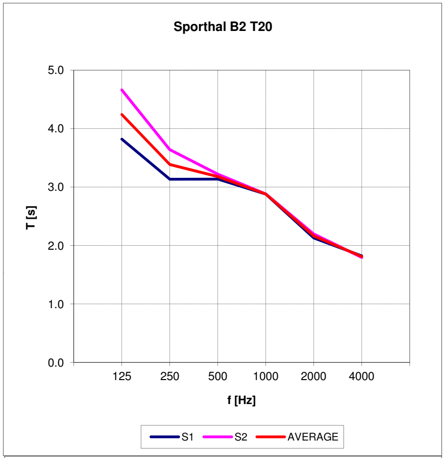
More sound absorbing material?

But what is exactly the 'strange' phenomenon found in sports halls constructed with perforated steel panels? Well, the results shown on the next page give an overview of the reverberation times in different sports halls, organized by size (see also chapter 2.2.1). The expected reverberation time for a sports hall without perforated steel panels on the walls or roof, is shown in figure 1. A long reverberation time for low frequencies and a shorter reverberation time for high frequencies. Short sound waves can be absorbed more easily than long sound waves.

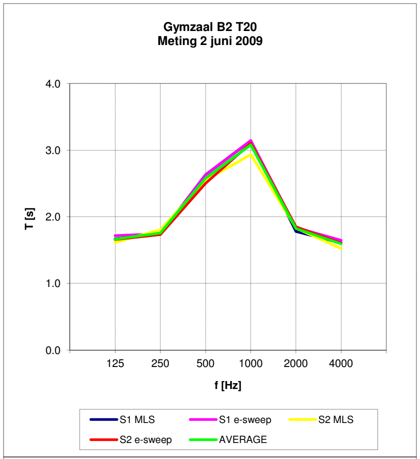


Fig,1. *Reverberation time of a 'stone-like' sports hall.*

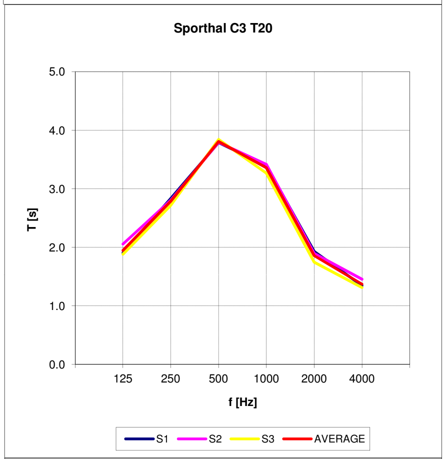
A sports hall constructed with perforated steel panels is expected to give similar results. Laboratory tests do not give any reason



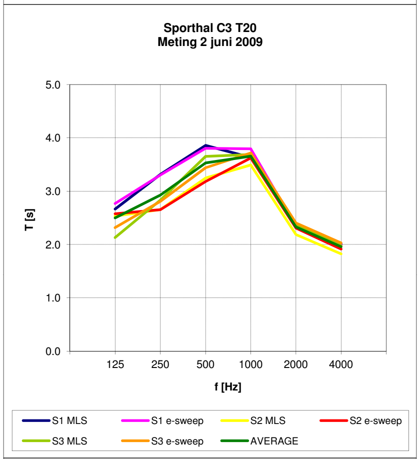
Fig,2. Sport hall 1
Walls: perfo steel (1x), concrete block (3x).
Roof: perfo steel



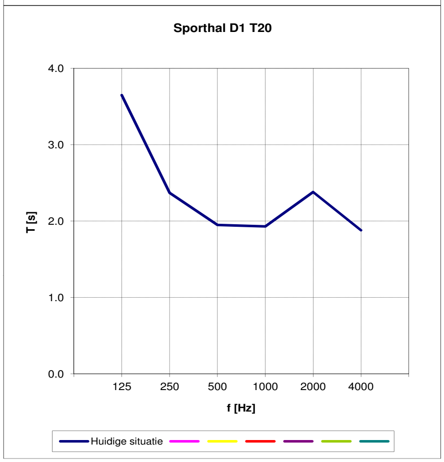
Fig,3. Sport hall 2
Walls: perfo steel (3x) first 3m plastered.
Roof: perfo steel.



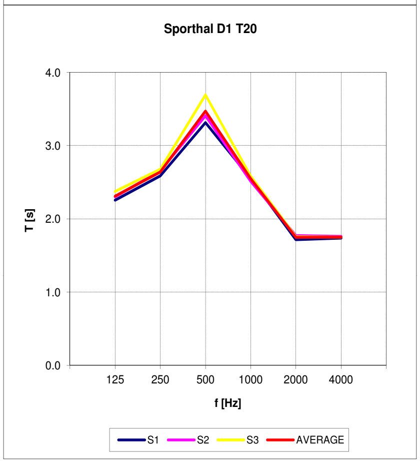
Fig,4. Sport hall 3
Walls: perfo steel (3x), concrete block (1x).
Roof: perfo steel



Fig,5. Sport hall 4
Walls: perfo steel (3x) first 3m plastered.
Roof: perfo steel.



Fig,6. Sport hall 5
Walls: plastered limestone/concrete.
Roof: perfo steel filled with mineral wool.



Fig,7. Sports hall 6
Walls: perfo steel.
Roof: perfo steel.

to have other expectations. Though, as shown in the graphs, some of the sports halls give a very short reverberation time for low frequent sound. Actually, the whole shape of the graph is different from expected.

In practice, sports halls with some type of perforated panels thus seem to have more sound absorption than one would expect based on lab measurements. The question is: why is this?

3. Literature study

Designing acoustics is not an easy job. Especially because the designed acoustic for a room often behaves differently than expected after being built. Also for sports halls, this is a big problem. When a hall is designed for a reverberation time of 1,0 seconds and it appears to be 1,5 seconds in practice, the speech intelligibility is decreased drastically. This could lead to complaints from users.

Is it possible to find a way of designing that leads to results closer to reality? This chapter will focus on the different points of view found in literature: different articles, congress papers and books have been studied. This literature study is a part of the demarcation of the subject.

3.1. Sabines theory

When we combine the found problem (the differences between the designed and the measured reverberation time) with Sabines reverberation time formula, it could lead to a direction for a solution.

$$RT = \frac{1}{6} \cdot \frac{V}{A + 4mV}$$

The reverberation time depends on: the volume of the hall (green), the absorption of the materials in the hall (blue) and the absorption of the air in the hall (pink).

The volume of a hall is relatively simple to measure. We assume this factor to be known. The absorption of air in a sports hall could cause a difference in result, but we assume the volume of the air and the air attenuation coefficient to be known. So, these factors could not mainly cause the difference. What is left is the reverberation time itself and the sound absorption of the materials in the hall.

3.2. Reverberation time

Dutch articles (Nijs 2004) (Luyckx and Vercammen 2011) (Nijs 2009) (Ruiter and Noordermeer 2010) (Nijs 2010) (Rychtarikova 2004) and graduation reports (Uittenbosch 2009) (Nesselaar 2001) mainly focus on the reverberation time to be the reason for the found difference. The authors share the same conclusions:

Not a suitable parameter

The reverberation time is not a suitable quantity to be used as a standard for acoustics in sports halls. Although it is a very constant and easy to measure parameter, it is not always true that a sports hall that exceeds the limit has bad acoustics. Though, the other way around; when sports halls stay within the limits, the acoustics is generally good.

Other reasons why the reverberation time is not a suitable parameter for sports halls are: the parameter is ment for music, not for noisy environments. The reverberation time depends on the volume of the hall. The reverberation time gives little information about multi source environments. And reverberation times are hard to predict from ray-tracing computer models. (Rychtarikova 2004)

Sound field

The sound field in a sports hall is not diffuse, so for example flutter echoes could have a big influence on the reverberation time measurements in practice. This is explained in section 3.5.

Sabine's formula

The formula of Sabine does not fit the problem of a sports hall because it focuses on a cubic room with evenly spread sound absorbing material together with a perfectly diffuse sound field. One of the authors tried to change the formula of Sabine in such a way that it fits better for the situation in sports halls (Nijs 2009). To check this corrected version of Sabine, it is used in the comparison in Attachment C.

3.3. Absorption coefficient

On the other hand, the absorption is not often questioned. Absorption coefficients of materials are measured in laboratory situations. These values are normally assumed to be right. Absorption coefficients are the link between the theory and the designers:

'For the architect the absorption coefficients of the materials are by far the most important values. In most cases these properties can be found in product documentations. Therefore it would be very convenient if acoustical standards were given in absorption coefficients as well. However, there is one big problem: to verify if an architect and product manufacturer have done their work properly, absorption coefficients must be measured if the room has been actually built. But, in-situ measurement of absorption coefficients is very difficult and therefore the reverberation time is used on most occasions.' (Rychtarikova 2004)

However, in a lab these are typically measured in a (as far as possible) diffuse sound field. Sports halls typically do not have such sound fields.

3.4. Other quantity to define acoustics in sports halls

So, the reverberation time seems to be unsuitable. But what is a good additional quantity for acoustics in sports halls?

Speech intelligibility is very important in sports facilities because signals, warnings and instructions need to be heard by everybody. Also, when the speech intelligibility is good, PE teachers do not have to shout to make themselves heard. This will expectedly decrease the complaint level on throat problems and tiredness. Besides that, there will be less dangerous situations due to misunderstandings.

In one of his articles (Nijs 2010), Lau Nijs studies the use of the STI in sports halls. STI is an abbreviation for Speech Transmission Index, a quantity for speech intelligibility. This parameter gives a good overall idea of the acoustics, because the speech intelligibility is an important parameter in a sports hall. Although STI values give a good overview, it is not easy to measure the parameter (where the reverberation time is easy to measure).

Besides that, the acoustical quality of a sports hall is not definable with just one STI value. More measurements in the same hall, on different locations need to be done. All scores together give the best overview. Or, are results measured at a distance of 10m from the source or an average value for the STI in a hall acceptable too? More research needs to be done on the use of the STI parameter to define acoustics in sports halls.

Defining a STI measurement method for sports halls

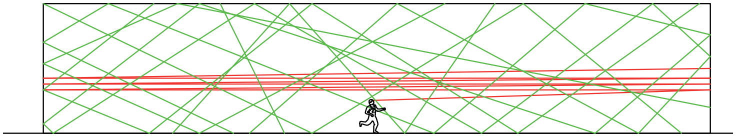
Although the method to calculate the STI is defined, it is not defined for sports halls. A 'normal' STI measurement could be done with and without background noise. For sports halls no background noise level is determined. This is also not an easy job, because how should this background noise level be, representing a bouncing basketball, screaming kids and a peeping shoes?

And the sound source, should it generate the sound pressure of a normal talking person or one with a raised voice? Besides those undefined parameters, L. Nijs mentions a few more.

It clearly shows how new this way of defining acoustics in sports halls is. It could work, but it needs more investigation. This also applies to other parameters that could define acoustics in sports halls like the strength (G). The parameter could be part of a solution but so far, there is not a suitable parameter found. The reverberation time still seems to be the best for now, a suitable additional parameter has not yet been found.

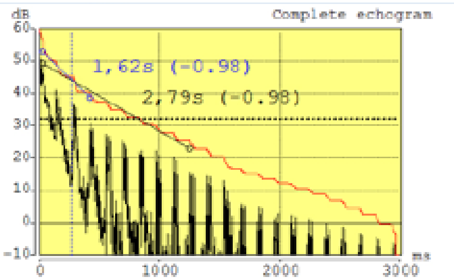
3.5. Flutter echoes

Back to the main problem: the reverberation time. Why is it so different from expected? A possible reason are flutter echoes. This type of echo reflects more than once between two parallel walls. When this keeps on going, the reverberation time could increase. See figure 1 on the right.

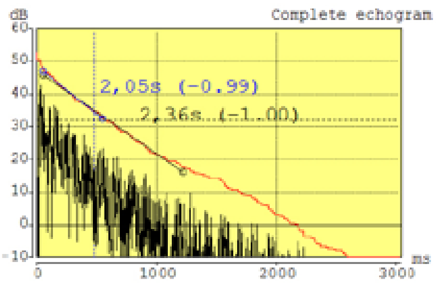


Fig,1. *Scheme of flutter echo*

When measuring reverberation times, an echogram shows the decrease of sound pressure in a room after the sound source has been switched off. With the use of tangents (see the black and blue line in figure 2 and 3), the reverberation time can be calculated. The needed difference of 60 dB can now be extrapolated from a difference of 15 dB (T15) in blue, or with a difference of 30 dB (T30) in black.



Fig,2. *Echogram with flutter echo simulated in Catt Acoustic*



Fig,3. *Echogram without flutter echo.*

The echogram in figure 2 shows a clear flutter, in figure 3 a 'normal' sound pressure decrease is shown. The curve of an echogram of a room without flutters should be an almost straight line downwards. The echogram in figure 2 shows a sagging curve.
Early reflections (between roof and floor) are absorbed strongly by the roof. Late reflections (between walls) are mostly reflected. Therefore, the horizontal sound field has a large influence on the shape of the curve. The tangents according to which the reverberation time is calculated, have different angles. Therefore, the reverberation time in the hall with a flutter echo cannot be clearly defined.

Flutter echoes are often discussed in articles. Lau Nijs (Nijs 2009) suggested that those echoes exist at high frequencies. Evert de Ruiter and Marc Noordermeer explained their findings in an answering article (De Ruiter 2010). Flutter echoes are also found in low frequency bands. Whether flutter echoes in low frequencies can occur depends on the size of the hall. Low frequencies exist of long sound wave lengths. In order to become a flutter echo, the wave should fit in the room.

The effect of a flutter could generate longer reverberation times. In this way, a reverberation time of a 'good' hall could exceed the limit. Striking though is that humans can hardly hear flutter echoes. This is only possible when the sound pressure level between the peaks and the 'normal' decay curve are big, minimal 10 dB. Also, the type of sound is important. A talking person cannot easily cause audible flutters because the sound is inconstant and rapid. The time difference between two peaks is also important for the audibility of a flutter. Though, an impulsive sound of a bouncing basketball could cause an audible flutter. So, a flutter can cause an undesirable effect on the reverberation time but it is still unclear whether it causes an audible nuisance.

Why can flutter echoes exist in sports halls? A sports hall is a type of building with parallel walls. Because of the parallel positioning, an echo can reflect endlessly if it is not absorbed. This brings me to the second characteristic of a sports hall: the hard materials. The floor should be hard and flat. It hardly absorbs sound. The walls should also be strong, balls could easily break down soft, porous or light materials. Those materials with a more open structure have better sound absorbing properties. This is conflicting.

Most seen solution to this problem is a sports hall with absorbing material on the walls from 3 meters above the ground together with ceilings that are made of a sound absorbing material. In this way, there is enough sound absorbing material in the sports hall. Though, a flutter echo can still occur in the lower 3 meters. Gym equipment or other sound scattering materials can help out in this case. When placing a mat against the wall, the walls are more absorbing and less parallel. This stops flutter echoes. (Luykx and Vercammen 2011)

Standing waves

A part of the difference could be explained by the fact that ray tracing models cannot simulate standing waves. As Lau Nijs writes in his later article: 'This has the unpleasant consequence that we have no precise calculation model for noise in a gym that is still on the drawing board.' (Nijs 2010) Standing waves can occur between two parallel walls in a sports hall. The sound should be constant. A talking person creates inconstant sound. Therefore, standing waves can hardly exist. The note of Lau Nijs is true, but maybe not always applicable to sports halls.

Conclusion

The need to find a suitable parameter to define acoustics in sports halls is clear. Though, this is not the main topic of this graduation report. The broad description of the problem above helps to understand the complex theory of sound in sports halls. The main topic of this graduation is the absorption coefficient of perforated steel panels. Some measurements give remarkable differences on the absorption coefficients of this material compared to the laboratory measurements. The sound absorbing panels seem to behave differently from expected from laboratory results. This graduation report will focus on this difference. The goal is to understand what causes this difference.

4. Learning agreement

4.1. Problem statement

Perforated steel panels seem to behave differently in practise than expected from a laboratory situation.

4.2. Question

Why do perforated steel sound absorbing panels seem to behave differently in absorption coefficient in sports halls than expected from laboratory test results?

4.3. Goal

The goal of this research is to determine why a difference in sound absorption behaviour of perforated steel panels exists between practical applications and lab situations. And, understanding what causes this difference in behaviour.

Hypotheses:

1. A perforated steel panel behaves differently in practice than in a laboratory situation on absorption coefficient because the shape of the panels causes sound absorption of parallel striking sound based on a phase shift principle.

2. A perforated steel panel behaves differently in practice than in a laboratory situation on absorption coefficient because the backing construction has influence on the result.

In order to find an answer to the research question, the hypoteses are tested by measurements. The measurements are done in a laboratory and in a scale model. The results are analysed. The conclusions give insight in the sound absorbing behaviour of perforated steel sound absorbing panels. Besides that, the conclusions give guidelines to the design phase of this graduation project.

4.4. Research method

Laboratory measurements

Absorption coefficient measurements are done in the reverberation room of the acoustical laboratory. Different samples of the roof structure are tested. By comparing the results, we get insight in the influence of different parts of the roof structure. Besides that, hypothesis one can be tested: the influence of the backing construction will become clear.

Scale model measurements

The scale model makes it possible to test the influence of different shaped roof structures in a small model. By using the same, reflecting material for all walls, we get insight in the influence of changes in roof shapes. Some of the roof structures are (partly) covered with a sound absorbing material to compare the results of those to the other variations. This research should give an answer to hypothesis two.

4.5. Relevance

Scientific relevance

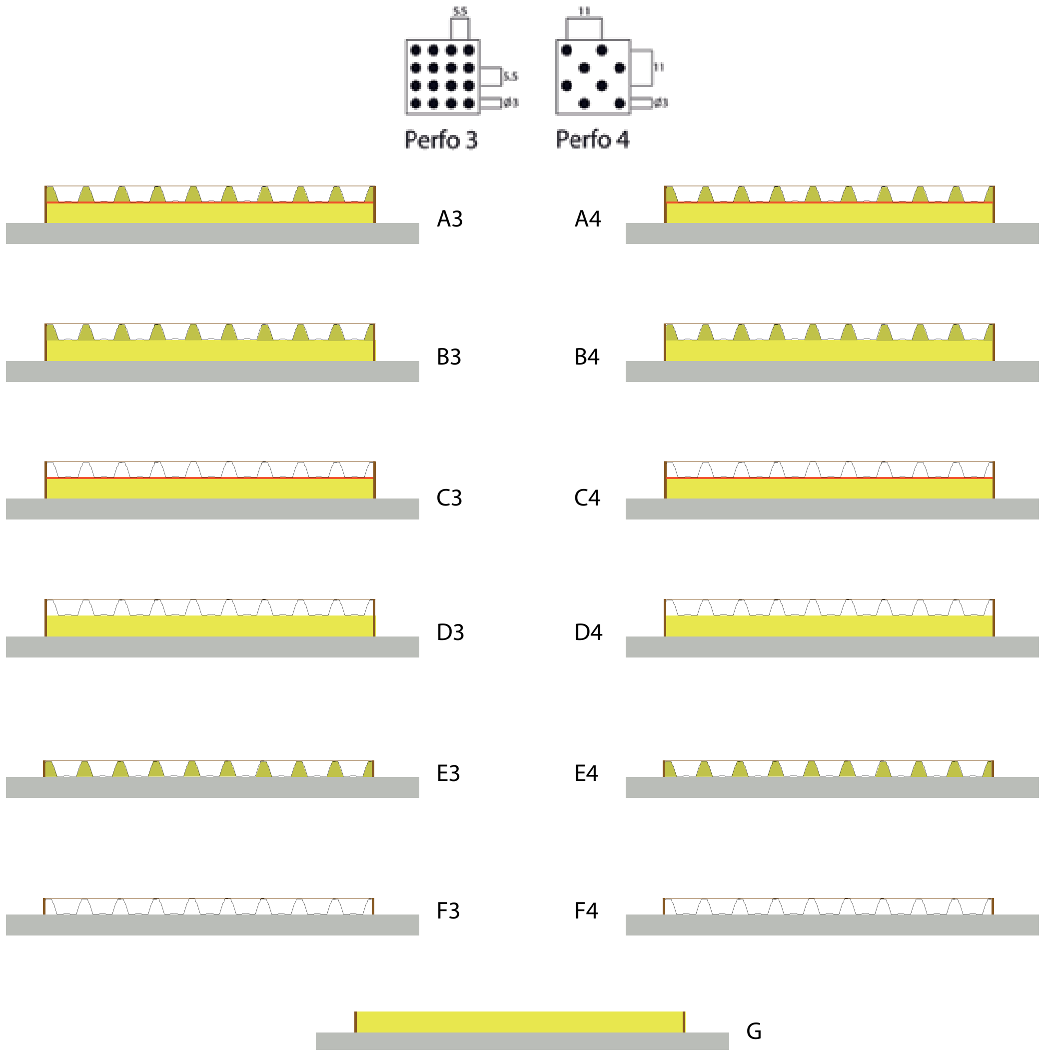
The sound absorption coefficient [α] is used by i.e. architects, designers, researchers and consultancy companies to predict the acoustical quality of a space. More knowledge on the behaviour of this type of sound absorbing material could lead to a more precise design of the acoustics in sports halls where this type of material is used. More knowledge on the sound absorbing behaviour could help creating a single-numbered value for designing acoustics. This value could be used by designers, engineers etc. This would make it easy to compare the acoustics of different spaces.

Social relevance

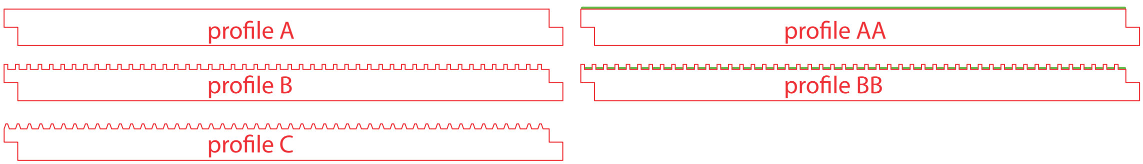
When sound in sports halls acts exactly as designed for, sports halls can be designed in a way that physical complaints of PE teachers are brought to a minimum.

Financial relevance

A more precise design could also prevent from additional costs for adjustments on the acoustics afterwards.



Fig,1. *Different test samples for laboratory meaurements.*



Fig,2. *Different roof structures for scale model meaurements.*

4.6. Design

After the research, a design needs to be made. The design should improve acoustics in sports halls. The more precise topic of the design depends on the results of the research; the conclusions should give guidelines for it.

The research is finished when the goal is reached and when a design, based on the conclusions of the research, is made. The results of the research and the design will be presented in a report and a presentation on July 29th 2012.

II. Theory

5. Important parameters

This chapter gives the definition of the most important parameters for acoustics in sports halls. The case study in Attachment C, uses more parameters. Those parameters are explained in Attachment B.

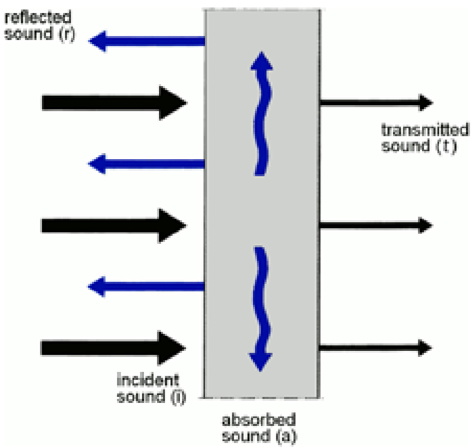
5.1. Absorption coefficient

Definition

Absorption coefficient α is a measure of the efficiency of a surface or material in absorbing sound. (Everest 2001)

How does it work?

The sound absorption coefficient describes the efficiency of the material to absorb sound. Absorption is the property that changes the acoustic energy of sound waves into another form, often heat. The other part of the acoustical energy is reflected or transmitted. The acoustic absorption of a material depends on the frequency of the sound, the size, location and shape of the material. In practice, the α varies between 0.001 for concrete and about 0.99 for mineral wool. An open window is considered a perfect absorber because sound passing through it never returns to the room. It would have an absorption coefficient of 1.0. Ten square meter of open window would give 10 sabins of absorbance. The absorption coefficient is the most important room acoustical variable for architects.



Fig,1. *Sound absorption (a). (Linden 2006)*

Sports halls

Values for sound absorption coefficients in sports halls differ per material, but the average value should be at least 0,28 (Rychtarikova 2004).

5.2. Reverberation time T

Definition

The reverberation time is defined as the time that expires till a sound pressure has decayed by 60 dB after the sound source has been switched off. (Linden 2006)

How does it work?

The reverberation time depends on the absorption in the room; the more absorbing material, the shorter the reverberation time. The activities that take place in a room determine which reverberation time is best for that room. The size of the room (hall) affects the actual reverberation time of a room and it plays a role in determining the required reverberation time for the room, as does the diffusivity of the room.

Formulas

In the late 1890s, Sabine developed a reverberation equation. The reverberation time was intend to express musical quality. This formula is not perfect for sports halls where noise levels are the main acoustical 'quality'. Besides that, Sabines formulas is based on a cubic room, a perfect diffuse sound field and evenly distributed absorption. Nothing of this is true in a sports hall.

So, the formula has disadvantages, but the developed alternatives by Eyring, Kosten and Fitzroy have disadvantages too (ISA-Sport 2002). Sabine's formula often comes closest to reality.

The formulas itself will be discussed in the next chapter.

Sports halls

The required reverberation time for sports halls depends on the size of the hall. ISA-Sport defined the maximal values per size in the standard ISA-N/A1.1. See figure 2.

Kind of room		Size [m] (w x l x h)	Reverberation time [s]
A.1	Gymnastics	≤ 14 x 22 x 5.5	≤ 1,0
A.2	Gym	13 x 22 x 7	≤ 1,1
A.3	1/3 sports hall /gym	14 x 24 x 7	≤ 1,2
B.1	Gym	16 x 28 x 7	≤ 1,3
B.2	Gym	22 x 28	≤ 1,4
B.3	2/3 sports hall	32 x 28	≤ 1,5
C.1	Sports hall	24 x 44	≤ 1,6
C.2	Sports hall	28 x 48 x 7	≤ 1,7
C.3	Sports hall	28 x 48 x 9	≤ 1,9
D.1	Sports hall	28 x 88 x 7	≤ 2,0
D.2	Sports hall	35 x 80 x 10	≤ 2,3

Fig,2. *Required reverberation time in sports halls . (ISA-Sport 2005)*

5.2.1. Reverberation time T20,T30

Although the reverberation time is defined as the time it takes for the sound to decay 60dB, this is seldom possible to measure due to the unavoidable background noise or too low power of the source. The reverberation time is therefore often based on the decay rate for a range of 20 or 30 dB starting 5dB below the stationary level. The value is afterwards extrapolated to 60dB.

5.3. Sound pressure level Lp

Definition

Sound pressure or acoustic pressure is the local pressure deviation from the ambient (average, or equilibrium) atmospheric pressure caused by a sound wave. Sound pressure in air can be measured using a microphone. The SI unit for sound pressure p is the pascal (symbol: Pa).

Sound pressure level (SPL) or sound level is a logarithmic measure of the effective sound pressure of a sound relative to a reference value. It is measured in decibels (dB) above a standard reference level. The commonly used “zero” reference sound pressure in air is 20 µPa RMS, which is usually considered the threshold of human hearing (at 1 kHz). (Wikipedia 2012)

How does it work?

The human ear therefore has a ‘measurement range’ for sound pressure from $2 \cdot 10^{-5}$ to 200 Pa. These extremes are separated by a factor of 10^7. Because it is difficult to work with figures that are so widely spaced, sound pressure cannot be directly used as a measurement of the sound intensity. A logarithmic measurement has therefore been introduced: the sound pressure level (Lp).

Formula

$p_0 = 2 \cdot 10^{-5}$ Pa = 0.00002 Pa

Because the square of the effective sound pressure is a measure of the intensity of the sound, the following expression is used to determine the sound pressure level (Lp):

$$\frac{p^2{}_{eff}}{p_0^2}$$

The sound pressure level is the logarithm of this relationship multiplied by 10:

$$L_p = 10\log\left(\frac{p^2{}_{eff}}{p_0^2}\right)[dB]$$

5.4. Equivalent sound pressure level LA,eq:

Definition

LA,eq is the equivalent continuous A-weighted sound pressure level. It is the energetic average over a certain time. A-weighted values are corrected to the human hearing.

Formula

$$L_{eq} = 10\log\left(\frac{1}{T}\cdot\int\frac{p_t^2}{p_0^2}\cdot dt\right)$$

L_{eq} = is the equivalent sound pressure level in dB or dB(A)

T = the exposure time in s

p_t = the effective sound pressure in Pa during the exposure time

p_0 = the reference sound pressure in Pa

Sports halls

In sports halls, sound pressure levels between 80 and 115 dB(A) can occur (Zandstra 2010). Above 80 dB(A) hearing protection needs to be provided. More about this in Attachment A: ‘Legislation and standards’.

5.5. Early Decay Time EDT

Definition

The EDT parameter is the reverberation time, measured over the first 10dB of the decay. This value is then multiplied with 6 in order to be comparable to T.

How does it work?

The EDT gives a more subjective evaluation of the reverberation time. The EDT is computed over every octave band and expressed in [ms]. The EDT parameter is important for music, but it can also be important for sports halls. By comparing the EDT and the reverberation time, we are able to define whether the room is diffuse or not. When the EDT and reverberation time are the same, there is a diffuse sound field in the room.

5.6. Speech Intelligibility

Speech intelligibility

It is very important in a sports hall that everybody can hear what a coach says. Besides that, the players should be able to understand each other too. Speech intelligibility depends on the reverberation time in the hall, the background noise level and the shape of the room. Moreover, the articulation of the speaker and the loudness of his/her voice and the hearing qualities of the listener are important too.

Speech intelligibility parameters

Worldwide several parameters for speech intelligibility are in use: STI, SII, SIL, Alcons etc. In Europe, the STI is mostly used because it corresponds closely to subjective measurements.

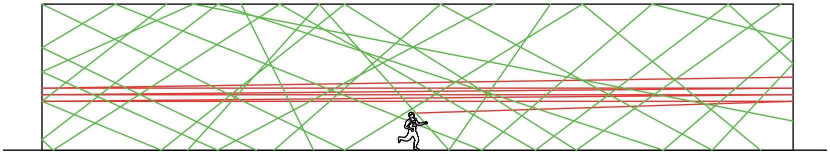
5.7. Flutter echoes

Definition

A repetitive echo set up by parallel reflecting surfaces. (Everest 2001)

How does it work?

Reflections of sound exist in every room. A flutter echo is a special type of reflection. A flutter echo will occur in between two parallel hard surfaces. The sound will reflect in between the two surfaces, back and forth. Flutter echoes can be reduced by adding more sound absorption or by more sound diffusion, or by creating nonparallel surfaces.



Fig,3. *Flutter echo: red.*

6. Measurement methods

Measurements on reverberation time, background noise and absorption coefficient need to be done according to a certain standard. In this way every measurement is done in the same way, so that the results are comparable all over the world. This chapter will describe the different measurement methods based on the standards.

6.1. Reverberation time measurement in sports halls

The method for measuring reverberation times in sports halls is explained in standard ISA-N/A 1.1 of ISA-Sport. Necessary for this type of measurements are: a positioned sound source that produces a signal and a positioned microphone. The impulse response can be derived from the difference between the produced and received signal.

The reverberation time is calculated from the measured T20 (see chapter 5.2.1.).
The sound source, an omni directional source, produces a certain signal from which the impulse response can be calculated using the correlation method. Reverberation times are calculated from this impuls response.

The software calculates the reverberation time per frequency band (125, 250, 500, 1000, 2000, 4000 Hz). Results of the measurements need to be registered.

6.1.1. Procedural

After preparing the measurements, all doors are closed. During the measurements only two persons are allowed to be in the hall. Measurements need to be done in a conventional situation. This means that:

- the temperature is according to the usage situation,
- the ventilation system is working for at least 50% of its capacity,
- the heating is turned on,

- when available, the cooling system is turned on,

When a sports hall can be divided into different parts, the reverberation time of the individual parts should be measured too.

According to standard ISA-N/A 1.1, sports halls can be devided into different categories depending on size and volume. See figure 1 below.

categorie		breedte x lengte [m]	hoogte [m]	inhoud [m^3]
A.1	gymnastieklokaal	14 x 22	5,5	≤ 1.700
A.2	sportzaal	13 x 22	7	1.701-2.100
A.3	1/3 sporthal / sportzaal	14 x 24	7	2.101-2.400
B.1	sportzaal	16 x 28	7	2.401-3.200
B.2	sportzaal	22 x 28	7	3.201-4.350
B.3.	2/3 sporthal	32 x 28	7	4.351-6.300
C.1	sporthal	24 x 44	7	6.301-7.400
C.2	sporthal	28 x 48	7	7.401-9.500
C.3	sporthal	28 x 48	9	9.501-12.400
D.1	sporthal	28 x 88	7	12.401-17.250
D.2	sporthal	32 x 88	10	17.251-29.000
E	overig			≥ 29.001

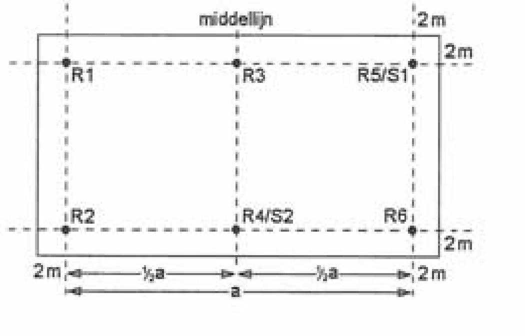
Fig,1. *Categories of sports halls. (ISA-Sport 2005)*

6.1.2. Source and microphone positions

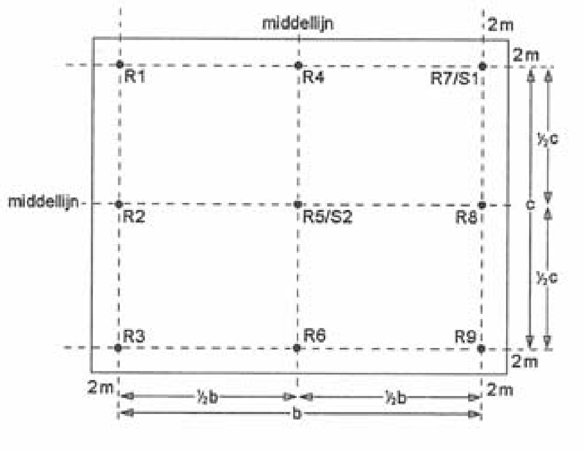
Depending on the category of the sports hall, different source (S) and microphone (R) positions are explained in figures 2 - 4.

In any case, all sources and microphones should be placed at a minimal distance of 2 meters from the wall. Besides that, the distance between the source and microphone positions in category A should be minimal 4 m, in category B 6m and in category C, D and E minimal 8 m.

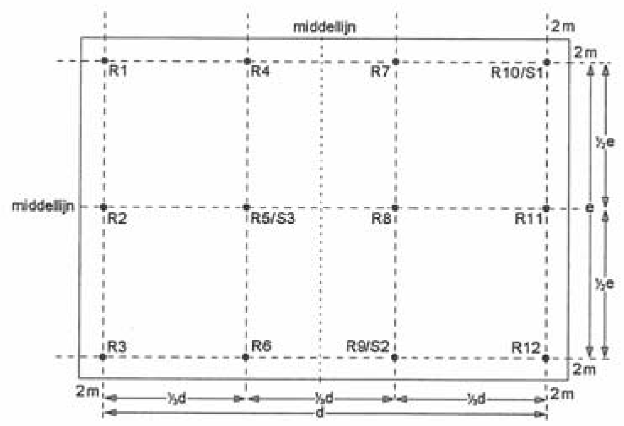
Per source position, all microphone positions should be measured except for the microphone position on the position of the source. The microphone and the middle of the source should be placed at a height of 1,5 m above the sports floor.



Fig,2. Microphone and source positions for category A. (ISA-Sport 2005)



Fig,3. Microphone and source positions for category C,D and E. (ISA-Sport 2005)



Fig,4. Microphone and source positions for category B. (ISA-Sport 2005)

6.1.3. Calculation and presentation of results

The average reverberation time is calculated as a arithmetic mean of all source and microphone positions over all frequency bands (125, 250, 500, 1000, 2000 and 4000 Hz). It is described in 0,1 seconds.
The reverberation time per frequency band is calculated as a arithmetic mean of all source and microphone positions per frequency band (125, 250, 500, 1000, 2000 and 4000 Hz). It is described in 0,1 seconds.

6.2. Background noise in sports halls

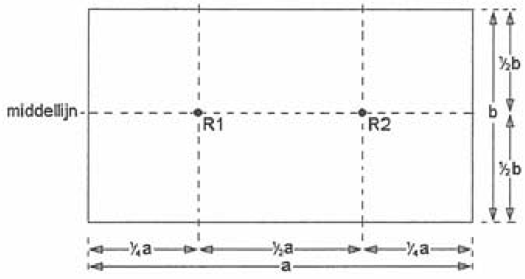
The method for measuring background noise in sports halls is explained in standard ISA-N/A 1.1 of ISA-Sport. The background noise is the sound pressure level that is present in the sports hall at the moment that no activities take place. It is a value expressed in dB(A). The background noise is measured by a microphone.

The background noise consists of:
- environmental noise,
- installation noise,
- other noise.
During the measurement, extraordinary conditions should be avoided.

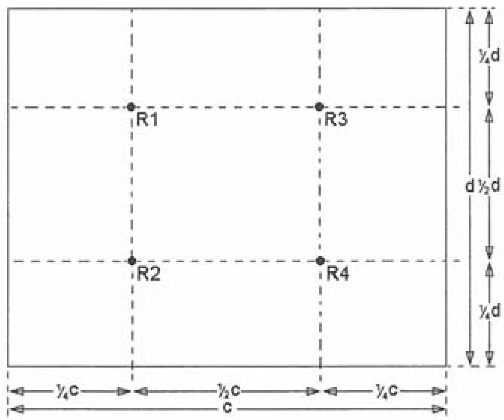
Procedural
Same as for reverberation time measurements. See section 6.1.1, except that there is no omni-directional sound source producing the signal.

Microphone positions
The microphone positions (R) depend on the category of the sports hall. See figures 5-7. The measurements should be done at a height of 1.5 m above the sports floor.

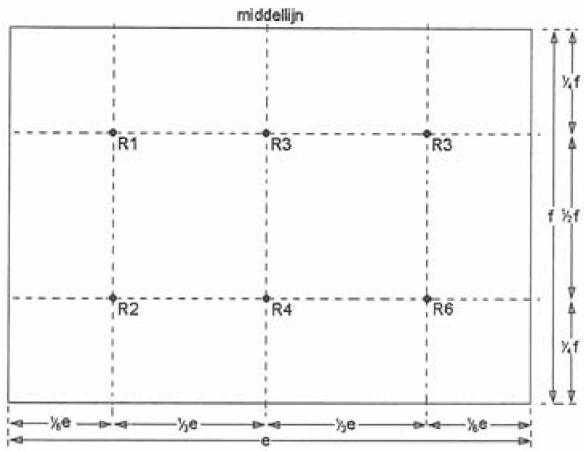
Calculation and presentation of results
The background noise is measured per microphone position and is presented as a energetic mean of three measurement results per microphone position. The sound pressure level is described in 0,1 dB(A).



Fig,5. *Microphone positions for category A. (ISA-Sport 2005)*



Fig,6. *Microphone positions for category B. (ISA-Sport 2005)*



Fig,7. *Microphone positions for category C,D and E. (ISA-Sport 2005)*

7. Sound absorption

A desired reverberation time is designed by selecting the proper wall and ceiling materials based on their sound absorbing behaviour in combination with the volume of and the function in the room. Since achieving this desired reverberation time in some sports halls is a problem, this chapter will focus on the theory behind sound absorption. Section 7.1. starts with an introduction on the topic. After that, different sound absorbing mechanisms will be explained in section 7.2. Finally, the sound absorbing properties of air will be discussed in section 7.2.6.

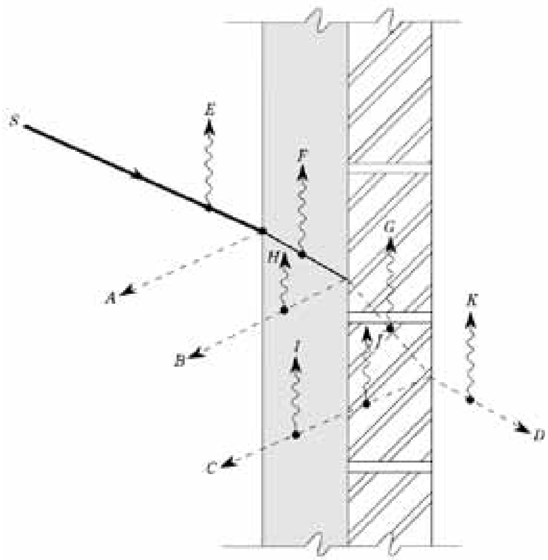
7.1. Introduction on sound absorption

All materials have sound absorbing properties. Incident sound energy which is not absorbed will be reflected or transmitted. A material's sound absorbing properties can be described as a sound absorption coefficient in a particular frequency range. The coefficient can be viewed as a percentage of sound being absorbed, where 1.00 is complete absorption (100%) and 0.01 is minimal (1%).

The sound absorption coefficient α describes the efficiency of the material to absorb sound. Absorbing sound is the property that changes the acoustic energy of sound waves into another form, often heat. The other part of the acoustical energy is reflected or transmitted.

7.1.1. Dissipation of sound energy

‘When sound wave S hits a wall (such as in figure 1), what happens to the energy it contains? If the sound wave is travelling in air and it strikes a concrete block wall covered with an acoustical material, there is first a reflected component (A) returned to the air from the surface of the acoustical material. Of course, there is a certain heat loss (E) in the air that is appreciable only at the higher audio frequencies. Some of the sound penetrates the acoustical material represented by the shaded layer in figure 1. The direction of travel of the sound is refracted downward because the acoustical material is denser than air. There is heat lost (F) by the frictional resistance the acoustical material offers to the vibration of air particles. As the sound ray strikes the surface of the concrete blocks, two things happen: a component is reflected (B), and the ray is bent strongly downward as it enters the much denser concrete blocks. Of course, there is further heat loss (G) within the concrete blocks. As the ray travels on, getting weaker all the time, it strikes the concrete-air boundary and undergoes another reflection (C) and refraction (D) with heat lost (I, J, and K) in three media.’ (Everest 2001)



Fig,1. *Incident sound being absorbed, transmitted and reflected. (Everest 2001)*

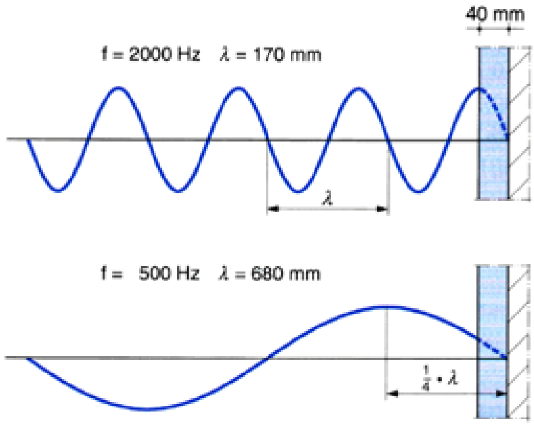
The sound absorption coefficient can be calculated from the absorbed energy and the incident energy.

$$\alpha = \frac{W_a}{W_i}$$

In a normal situation the absorbed, transmitted and reflected sound equals one: a+t+r=1 but it is different in case of a sports hall, where transmitted sound might end in the outer air and ‘disappears’. This is a kind of sound absorption too: a =1− r, where a also includes transmission.

7.1.2. Absorbing the majority of energy

A number of characteristic values can be derived in relation to wave motions, specifically, the frequency (f), the wavelength (λ) and the propagation speed (c).The number of vibrations per second is known as the frequency (f). This is expressed in Hertz (Hz). The wavelength is the distance between two points that are in an identical state (are in phase). The propagation speed (c) of longitudinal waves is the same for all frequencies and depends on the medium and the temperature of the medium.

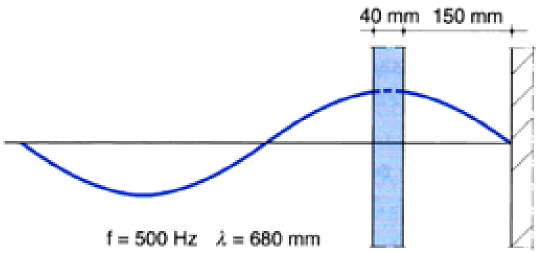


Fig,2. *Influence of frequency and layer thickness on sound absorption. (A.C. van der Linden 2006)*

The following relationship exists between the propagation speed, the frequency and the wavelength:

$c = f * \lambda$ [m/s]

Here: c is the propagation speed of the sound in m/s, f is the frequency in Hz λ is the wavelength in m. (Linden 2006)



Fig,3. *Better sound absorption without increasing the material thickness. (A.C. van der Linden 2006)*

The thickness of the material is important when it needs to absorb sound. The particle speed is at its maximum at ¼ λ from the wall. This is where the majority of the energy is. This quarter wavelength differs per frequency.

The sound absorbing material works best when the thickness is the same as this ¼ λ. This means that for low frequencies, like bouncing basketballs, a very thick layer of sound absorbing material must be added to the wall. However, this is not always possible and necessary. When a cavity construction is used, it is possible to apply the material exactly at the place where the majority of energy is.

7.2. Sound absorbing mechanisms

The acoustic absorption of a material depends on the frequency of the sound, the size, location and shape of the material. In practice, the α varies between 0.001 for concrete and about 0.99 for mineral wool. An open window is considered a perfect absorber because sound passing through it never returns to the room. It would have an absorption coefficient of 1.0. Ten square meter of open window would give 10 sabins of absorbance. The absorption coefficient of a material is normally tested in a laboratory according to prescribed standards (ISO 354_2003). Then, manufacturer can advertise with the official properties of the material. The properties are not often questioned. Architects use these properties to form an idea of the acoustical quality within their design. But, on which priniciples are most of the sound absorbing materials based? The book 'Bouwfysica' of A.C. van der Linden (2006) describes the basics of sound absorption very clearly: 'Absorption of sound is, nothing more than the conversion of vibrations into heat. In principle, sound absorption can be achieved in two ways:

- by friction with air movement in porous materials.
- by means of resonance.'

This section will focus on friction, resonance and sound absorption of air.

7.2.1. Friction

'When a sound wave penetrates a porous material there is friction between the coming and going air particles in the pores of the material. This friction causes the sound energy (movement) to be converted into heat. The sound is absorbed by the material.' (A.C. van der Linden 2006)

In order to ensure that the sound can penetrate the material it must be as porous as possible. Too much noise must not be reflected at transitions between air and material. This is indicated using the airflow resistance of the material. Absorbing material must have a low airflow resistance so that the transition from the air to the material is such that not too much sound is reflected. On the other hand, the airflow resistance must not be too low, otherwise there will not be sufficient friction and absorption will be inadequate.

Porous materials

Porous absorptive materials absorb sound according to the basics of friction and airflow resistance. The materials are usually fuzzy, fibrous materials in the form of boards, foams, fabrics, carpets, cushions, etc. The cells of the material should have an open structure, like fibers. If the fibres are too loosely packed, there will be little energy lost as heat. On the other hand, if they are packed too densely, penetration suffers and the air motion cannot generate enough friction to be effective. Between these extremes are many materials that are very good absorbers of sound. These are commonly composed of cellulose or mineral fibre.

Covering sound absorbing materials

The principle of sound absorption mentioned above is very useful in sports halls, although sound absorbing materials are often very soft and fragile. Especially in sports halls, mechanical damage is a realistic threat. To protect the material, a layer of perforated steel, hardboard or plaster can be added. The absorbing behaviour should not be less because of the additional layer, so the percentage of perforations must be greater than approximately 20% and the intermediate distance between the openings must not be greater than 20 mm. In sports halls, often wooden laths with open spaces in between are used. Sometimes, the sound absorbing material is visible trough the openings.

Then something on the aesthetics of a sports hall. The look of a facility becomes more and more important, fancy materials and colours are used. Painting a sound absorbing material could have a very bad influence on the sound absorption. Paint can close the openings in the material. Besides that, the airflow resistance will increased so that sound waves are no longer able to penetrate the material.

7.2.2. Resonance

Objects have a frequency at which they start to vibrate (their natural frequency). When an object is struck by a vibration that is equal to this natural frequency, the object will start to vibrate. This phenomenon is known as resonance.

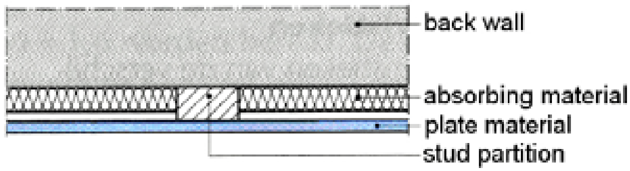
Resonators typically act to absorb sound in a narrow frequency range. Resonators include some perforated materials and materials that have openings (holes and slots). The classic example of a resonator is the Helmholtz resonator, which has the shape of a bottle. The resonant frequency is governed by the size of the opening, the length of the neck and the volume of air trapped in the chamber.

A mass-spring system exists of a specific oscillation value. When the mass starts moving, it will start oscillating in a specific speed; the resonant frequency. An outer alternating power with the same frequency can easily cause an intense vibration. The energy at the resonance frequency is maximal, so systems that are based on this principle can absorb sound very efficiently; for example plate resonators.

A combination of both friction and resonance is possible. An example of this are the old-fashion soft-board tiles. Although they had very good sound absorbing properties, fire safety for ceilings made them inadequate.

Plate resonators

'Sheet material such as plywood, chipboard, metal etc., can also have a sound absorbing effect when they are fitted in front of an air cavity. The sheet forms a mass-spring system with the underlying air layer (see figure 3). The natural frequencies of these structures fall in the range from approximately 50-500 Hz. In addition, all kinds of bending waves occur in the sheet with specific natural frequencies.' (A.C. van der Linden 2006)



Fig,4. *Un-perforated panels: absorption in a frequency range from approximately 50-500 Hz. (A.C. van der Linden 2006)*

$$f_r = \frac{1}{2\pi}\sqrt{\frac{1.4p_0}{m_1 D}} = \frac{60}{\sqrt{m_1 D}}$$

With:

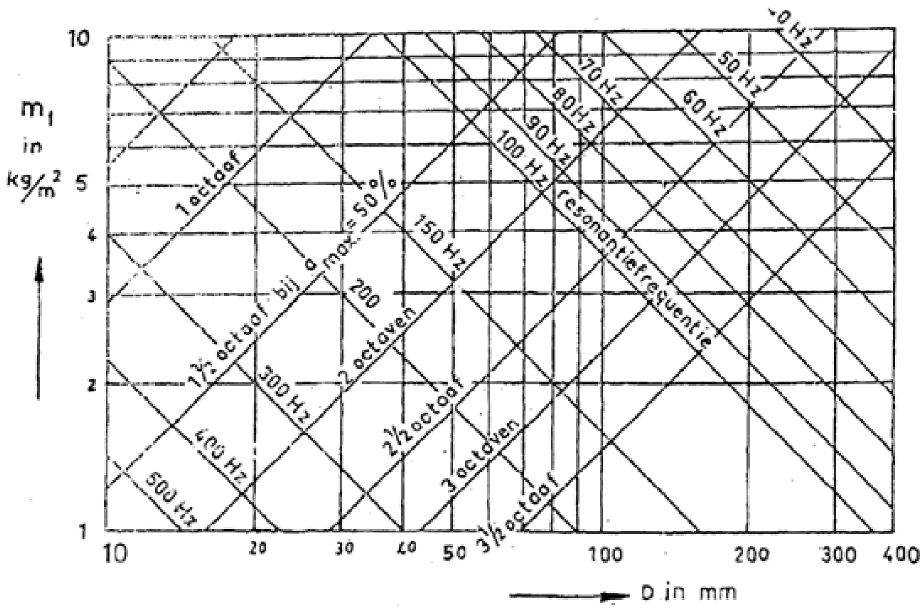
m_1 = mass in kg/m^2

D = distance to rigid backing wall

p_0= 10^5 [Pa]

The resonance frequency (f_r) of a plate resonator can be calculated by the following formula. This formula works for only plate resonators with a cavity filled with air or a porous material with a low airflow resistance.

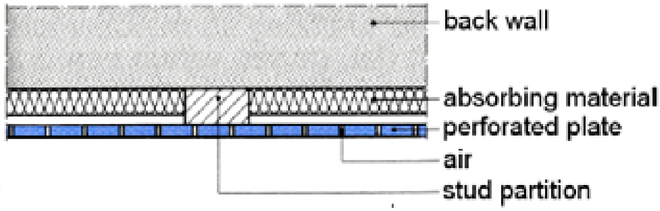
By changing the D and m1, the frequency and the amount of octaves around fr will change. The graph shows what happens when the cavity is filled with a sound absorbing material so that the sound absorption coefficient is 0.50. This filling lowers the frequency peak and makes it broader. The broader the peak, the more sound of different frequencies it will absorb.



Fig,5. Frequency f_r at which a not-perforated panel with mass m and backing air layer with thickness D, maximal absorbs. (Nederhof 2005)

7.2.3. Perforated panels/Helmholtz resonators

The way in which perforated panels work is based on the resonance principle. The air in the holes forms a kind of mass that can vibrate on the air layer that lies behind it, acting as a kind of spring. This type of mass-spring system has a natural frequency (or resonance frequency). Because of the operation of the mass-spring system, these structures can absorb effectively in the frequency range from approximately 300-1500 Hz. The high frequencies (up to 1500 Hz) can be absorbed because the mass of the air is smaller than for plate resonators. However, the percentage of perforations must not be too high (5 to 10%). The absorption (by the porous material) of higher frequencies is, perhaps, lost by this degree of perforation, but in return there is a gain in absorption in the lower and middle frequencies.



Fig,6. *Perforated panels: good absorption in a frequency range from approximately 300-1500Hz. (A.C. van der Linden 2006)*

The perforated panel shown in figure 5 is a type of Helmholtz resonator. The incident sound wave results in an alternating air pressure. This sometimes positive, sometimes negative pressure activates the air in the perforations: it will start vibrating. This will result in a changing air pressure in the backing (porous) sound absorbing material. This pressure equalizing mechanism by the air in the perforations works easier than when the whole panel needs to vibrate. The vibrating mass is very small, the graph in figure 4 shows that this results in a high natural frequency and an absorption with a wider frequency range.

The resonance frequency can be estimated from:

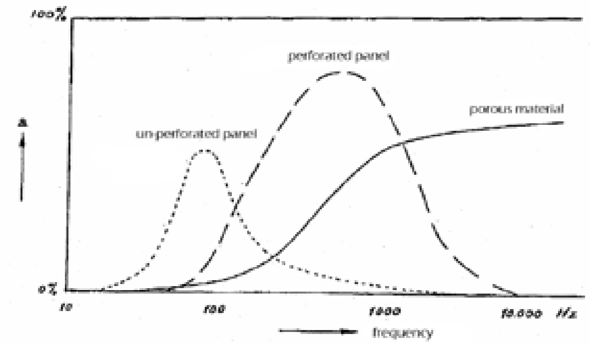
$$f_{ms} \approx \frac{c_{air}}{2\pi}\sqrt{\frac{e}{d_{cav}d_p}} \approx 54\sqrt{\frac{e}{d_{cav}d_p}}$$

7.2.4. Perforation grade

The amount and size of perforations in a perforated panel can make a big difference in the sound absorbing behaviour. A perforated panel mentioned later, works as a Helmholtz resonator when the plate consists of 5- 10% of perforations. When the perforation degree is greater than 30%, the facing could be ignored. The backing sound absorbing material then works like a porous material. Recent research focuses on developing a micro perforated panel (Onen 2010). The diameter of the perforations in this panel differ from 0.05 to 1 mm. The degree of perforations varies between 0.5 and 2%.

7.2.5. Different mechanisms compared

The sound absorbing behaviour of porous materials, un-perforated and perforated panels compared gives the following graph. The chart below gives an overview of the main properties of the three mentioned sound absorbing mechanisms.



Fig,7. *Different mechanisms compared (Nederhof 2005)*

Porous materials	**Un-perforated panels**	**Perforated panels**
Textile, carpets, glass wool, rock wool porous plaster etc.	Triplex, hardboard, aluminium or chipboard on layer of air	Plate materials with perforations or slots, perforation degree < 30%
Absorbing: high frequencies	Absorbing: low frequencies	Absorbing: mid frequencies
Painting not allowed	Painting allowed	Painting allowed, but openings should stay open.

7.2.6. Air

The absorption of sound waves in the air of sports halls is in this case significant since sports halls are large rooms. But how does this phenomenon work? A summary of this physical phenomenon:
The propagation losses will be characterised by the decrease of the intensity in a plane sound wave. The equation of this intensity is:

$$I(x) = I_0 \exp(-mx)$$

Where x denotes the path length and I_0 is the incident intensity for x = 0 m

With: $m = R_{air} / (4.343c)$ [m-1]

Where *c* is the velocity of sound in m/second and R_{air} is the decay rate of sound due only to the absorption of sound in air. (Harris 1967)

The equation is based on the assumption that the changes in the state of a volume of gas take place adiabatically, i.e. there is no heat exchange between neighbouring volume elements. The equation states that a compressed volume element has a slightly higher temperature than an element which is rarefied by the action of the sound wave. Although the temperature differences occurring at normal sound intensities amount to small fractions of a degree centigrade only, they cause a heat flow because of the finite thermal conductivity of the air. This flow is directed from the warmer to the cooler volume elements. The changes of state are therefore not taking place entirely adiabatically. According to basic principles of physics, the energy transported by these thermal currents cannot be reconverted completely into mechanical, i.e. into acoustical energy; some energy is lost to the sound wave. The corresponding portion of the attenuation constant m increases with the square of the frequency.

Under normal conditions the above mentioned cause of attenuation in air is negligibly small compared with the attenuation caused by what is called ‘thermal relaxation’. This physical phenomenon can be described shortly: when a gas is suddenly compressed, i.e. when its internal energy is increased, the whole additional energy will be stored at first in a form called translational energy. After that, the energy will be divided over the other so-called energy stores. In this way, the establishment of the new equilibrium will be done. This process takes some time. This time delay of acceptation or delivering of energy at wrong moments is less important for lower than for higher frequencies.

This process is a sort of ‘internal heat conduction’ which weakens the sound wave. The maximum weakening of the sound wave will take place when the duration of one sound period is comparable with a specific time, the above explained ‘relaxation time’. (Kuttruff 2000)

Relative Humidity (%)	Frequency (kHz)						
	0.5	1	2	3	4	6	8
40	0.60	1.07	2.58	5.03	8.40	17.71	30.00
50	0.63	1.08	2.28	4.20	6.84	14.26	24.29
60	0.64	1.11	2.14	3.72	5.91	12.08	20.52
70	0.64	1.15	2.08	3.45	5.32	10.62	17.91

Fig,8. *Attenuation constant of air at 20 °C and normal atmospheric pressure, in 10^{-3} m^{-1}. (Bass 1995)*

Since the sound in sports halls does not only exist of high frequencies, we are also interested in the absorption of sound in air at low frequencies. This value is less important for small rooms, but for sports halls it has to be taken into account. In the calculation below, we see that even for lower frequencies, 125 a remarkable part of the sound is absorbed by the air at 20 °C and a relative humidity of 50%: (sports hall of 24 x 44 x 9 meter)

125 Hz: 0.00003 m^{-1} * 9504 m^3 =
2.9 m^2 open window = 2.9 Sabins (metric).

500Hz: 0.00161 m^{-1} * 9504 m^3 =
15.3 m^2 open window = 15.3 Sabins (metric).

2000Hz: 0.00960 m^{-1} * 9504 m^3 =
91.2 m^2 open window = 91.2 Sabins (metric).

TABULATION OF DATA FOR
ABSORPTION OF SOUND IN AIR VERSUS PERCENT RELATIVE HUMIDITY
AT 20 DEGREES CENTIGRADE FOR A FREQUENCY OF 125 HERTZ

(1) TEMP DEGR CENT	(2) FREQ HERTZ	(3) REL HUM PER CENT	(4) ATTEN COEF PER METER	(5) 4M PER METER	(6) ATTEN COEF PER FOOT	(7) 4M PER FOOT	(8) ATTEN DB PER 100 METER	(9) ATTEN DB PER 1000 FEET	(10) DECAY RATE DB PER SECOND
20	125	50.0	.00007	.00030	.00002	.00009	.033	.102	.115
20	250	50.0	.00016	.00067	.00005	.00020	.073	.223	.252
20	500	50.0	.00040	.00161	.00012	.00049	.175	.534	.601
20	1000	50.0	.00097	.00388	.00029	.00118	.422	1.287	1.449
20	2000	50.0	.00240	.00960	.00073	.00292	1.042	3.179	3.580
20	4000	50.0	.00611	.02444	.00186	.00745	2.653	8.089	9.109

Fig,9. *Absorption of sound in air (Harris 1967)*

Most reverberation time formulas include the term ‘A+4mV’ which is the correction for attenuation of air. So, in these cases, the absorption of air is taken into account.

8. Perforated panels

Sound absorbers consisting of a perforated facing backed with a porous material have been used in different forms for many years. As very efficient tuned systems, they can achieve high values of absorption at the resonance frequency.

This chapter will focus on the more physical side of perforated sound absorbers. What factors have influence on the absorption? To explain this theory, an experimental study of W.A. Davern (1977) is used. The results of this study illustrate the different parameters and their properties. But first, the difficult theory behind Helmholtz resonators is explained.

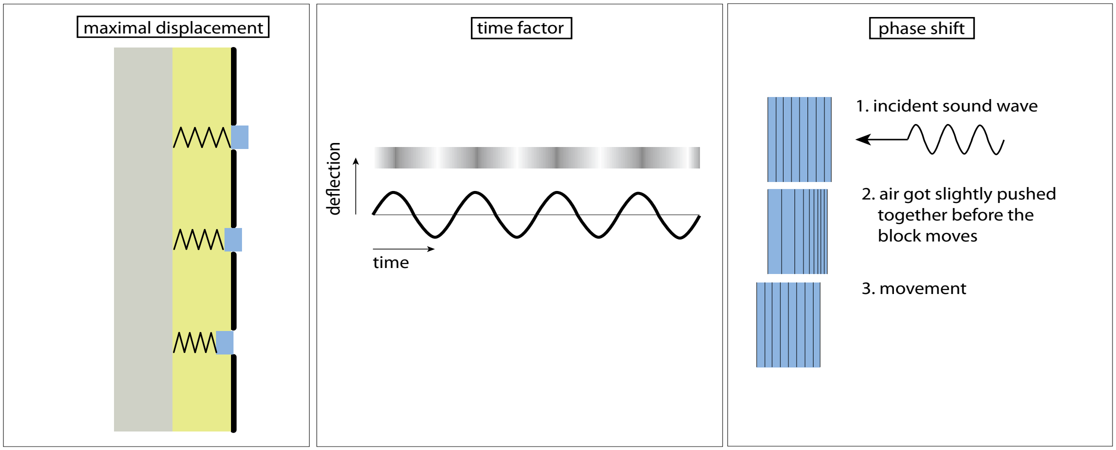
8.1. Helmholtz resonator, the theory

A perforated steel panel like the profiles often used in sports halls, works with the principle of a Helmholtz resonator. When we blow over a bottle, for example, a tone can be heard. This is the exact same principle. A little block of air, created by a hole in the facing (or in the neck of the bottle), can vibrate on the backing layer of porous material (or on the air in the bottle). The oscillation of the block of air can be calculated by a general formula, see figure 1.
The movement and so the maximal displacement can be derived by formula A. The greater the displacement, the higher the acoustical impedance. The absorption coefficient can be calculated from this impedance. See formula B for calculating the absorption coefficient of a perforated panel. An other way to calculate alpha is by formula C which is based on the displacement. Both formulas are comparable although the impedance is theoretically more correct.

8.1.1. Displacement

The incident wave forces the air in the hole to a damped oscillation. The displacement depends on the following factors:

$$\underbrace{U(t)=\hat{u}}_{\text{maximal displacement}}\cdot\cos(\underbrace{\omega t}_{\text{time factor}}+\underbrace{\beta}_{\text{phase shift}})$$



Fig,1. *Displacement of the little block of air in the perforation of the facing.*

1. The maximal displacement

The forces acting on the resonator are: the incident sound wave, the force of the spring and the resistance between the air and the sides of the holes in the facing. Adding those forces results in a differential equation. See formula A. From this formula, the maximal displacement can be derived.

2. Time factor

The air in the perforation moves from one side to the other caused by the incident sound wave. Whether the deflection is maximal or minimal depends on the time. Since the pressure and speed fluctuate in time, often described as a sinus, complex numbers are often used in formulas to describe the 'time factor'.

3. Phase shift

The third factor is the phase shift. When the sound wave hits the block of air in the perforation, the air volume will slightly pushed together on that side. The block will not directly move. Therefore, the movement is delayed. This is called phase shift.

When the natural frequency of the Helmholtz resonator and the frequency of the incident sound wave match, most of the energy will be absorbed. (Haupl 2008)

Formula A differential equation of movement

-force of spring - resistance in perforation + soundwave = a certain movement

$$-k \cdot u - r \cdot \frac{d}{dt} \cdot u + F_0 \cdot \cos(2\pi \cdot f_e \cdot t) = m_{perf} \cdot \frac{d^2}{dt^2} \cdot u$$

$-k =$ spring constant
$u =$ deflection
$r =$ resistance constant
$F_0 =$ natural freqency of Helmholtz resonator
$f_e =$ freqency of incident sound wave
$t =$ time
$m_{perf} =$ mass of air in perforation

Impedance

The acoustical impedance is the ratio of the sound pressure at a boundary surface to the sound flux (flow velocity of the particles or volume velocity, times area) through the surface.

$$Z_a = \frac{p(t)}{v(t) \cdot S}$$

Specific acoustical impedance is the ratio of the sound pressure at a point to the sound flux through that point.

$$Z_s = \frac{p(t)}{v(t)}$$

Since impedance Z is a so called complex number, it consists of a real and a imaginary component. Complex numbers have some calculation rules that are necessary to understand the theory behind the calculation of the absorption coefficient.

Z= impedance, a complex number (z)
Z'= the real component of the impedance (a)
Z''= the imaginary component of the impedance (i*b)
z= a+i*b or Z=Z'+Z''

Formula B for sound absorption of a perforated panel.

The absorption coefficient of Helmholtz resonators can be calculated from:

$$\alpha = \frac{4Z'_{1R;panel}Z_0}{(Z'_{R;panel}+Z_0)^2+Z''^2_{1R;panel}}$$

Z_0= the characteristic impedance of air = ρc

$Z_{1R;panel}$= the impedance of the resonator in total = Z_R/h

$Z_R = Z_V + Z_M$

h= the percentage perforations

Z_V = impedance of the air cavity ($Z_V = Z_V' + Z_V''$)

Z_M = impedance of the air in the perforation ($Z_M = Z_M' + Z_M''$)

$Z'_{R;panel} = Z'_M + Z'_V$ = the real component of the complex number of Z_R

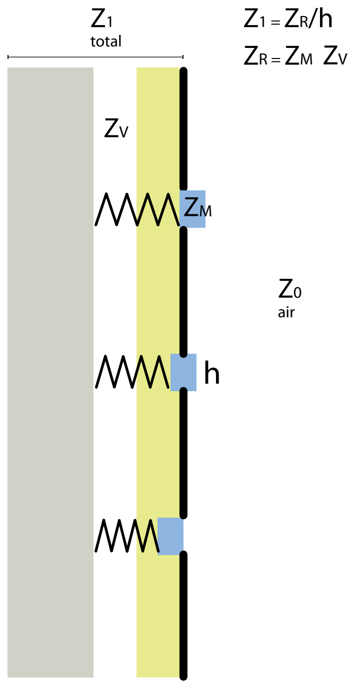
$Z''_{1R;panel} = Z''_M + Z''_V$ = the imaginary component of the complex number of Z_R

$$Z_v' = 0$$

$$Z_v'' = -\frac{\rho c^2}{\omega d_{cav}}$$

$$Z_m' = \rho\sqrt{8\nu\omega}\left(1+\frac{d_p}{2r}\right)$$

$$Z_m'' = \omega\rho\left[d_p + 2r\Delta l + \sqrt{\frac{8\nu}{\omega}\left(1+\frac{d_p}{2r}\right)}\right]$$



Fig,2. *Helmholtz resonator*

r= radius of perforation

v= viscosity of air

Δl= part of air outside of perforation with influence

For determining impedances, I refer to the book ‘Noise and vibration control engineering’ of Beranek and Ver, and ‘Acoustic Absorbers and Diffusers: Theory, Design and Application’ of Cox and d’Antonio.

Fig,3. *Blowing over a bottle: a Helmholtz resonator (http://blog.playdation.com)*

Formula C for sound absorption of a perforated panel. (Formula C follows from formula A.)

$$\alpha = \frac{1}{1 + \dfrac{\omega^2 \left(d_p + 0.5\sqrt{\pi A_{perf}}\right)}{32 e \omega_{ms} \eta_{perf} c_{air}} \left[\left(\dfrac{\omega_{ms}^2}{\omega^2} - 1 \right)^2 + \left(2\eta_{perf} \dfrac{\omega_{ms}}{\omega} \right)^2 \right]} \qquad \omega_{ms} = \sqrt{\frac{E_{cav} e}{d_{cav} \rho_{air} \left(d_p + 0.5\sqrt{\pi A_{perf}}\right)}}$$

ω	circle frequency = $2\pi f$ [rad/s]
d_p	thickness of perforated facing [m]
A_{perf}	area of one perforation [m²]
e	percentage perforations [-]
ω_{ms}	the circle frequency at which mass-spring resonance occurs = $2\pi f_{ms}$[rad/s]
η_{perf}	a resistance constant for energy dissipation[-]
c_{air}	speed of sound in air [m/s]
E_{cav}	modulus of elasticity of the cavity [Pa]
d_{cav}	thickness of cavity behind perforated facing [m]
ρ_{air}	the density of air [kg/m³]

When the cavity is filled with just air or an open absorbing material like mineral wool, the formula for resonance frequency can be written as:
($E_{cav} = \rho_{air} c_{air}^2$)

$$\omega_{ms} = c_{air} \sqrt{\frac{e}{d_{cav} \left(d_p + 0.5\sqrt{\pi A_{perf}}\right)}}$$

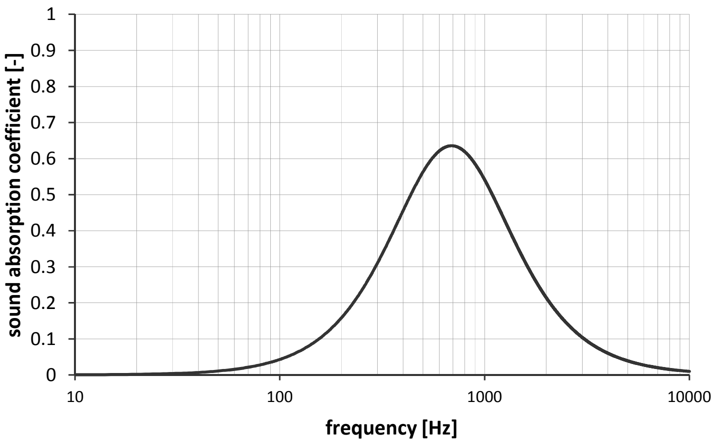
Or if the second term behind the parentheses is neglected:

$$\omega_{ms} = c_{air} \sqrt{\frac{e}{d_{cav} d_p}}$$

And thus the frequency:

$$f_{ms} \approx \frac{c_{air}}{2\pi} \sqrt{\frac{e}{d_{cav} d_p}} \approx 54 \sqrt{\frac{e}{d_{cav} d_p}}$$

(also mentioned in chapter 7.2.3)



8.1.2. Most important parameters

The formula for calculating the sound absorption coefficient of a perforated steel panel exists of a lot of different variables. The most important parameters will be explained in this section. As mentioned before, an experimental study of W.A. Davern (1977) is used to illustrate the workings of these parameters.

- ***Percentage perforation***

The more perforations the higher the resonance frequency.

- ***Thickness of the facing***

The thicker, the lower the frequency of the absorption peak.

- ***Density of the porous backing material***

The denser, the broader the tuning of the system.

- ***Air space between the facing and backing material***

Adding an airspace decreases the absorbing properties of the system.

- ***Impervious layer between the facing and backing material***

Adding an impervious layer decreases the absorbing properties of the system drastically.

(These conclusions are found for specific test samples. Other measurement setups give different results. The principles of perforation degree, thickness of facing and air space stay the same. The effect of the impervious layer strongly depends on the type of layer.)

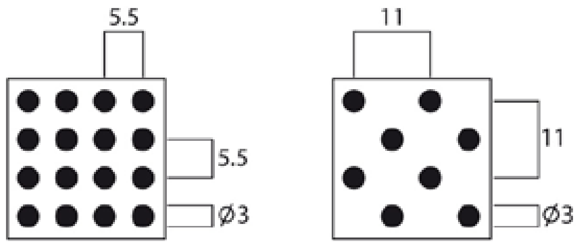
Knowledge on these factors can be used in achieving a good design.

8.1.3. Percentage perforation

Changing the percentage of the open area of the facing is one of the principal ways of changing the frequency at which resonance occurs. The effect of increasing the percentage perforation, in the research of W.A. Davern (1977) by enlarging the holes, is to raise the resonance frequency. As a result, the low frequency absorption is decreased.

When we aim for the low frequencies to absorb, the percentage perforation should be small. Figure 3 shows how the absorption behaviour changes with the percentage perforation. Even the characteristics of the backing material without facing is given: the 100% perforation.

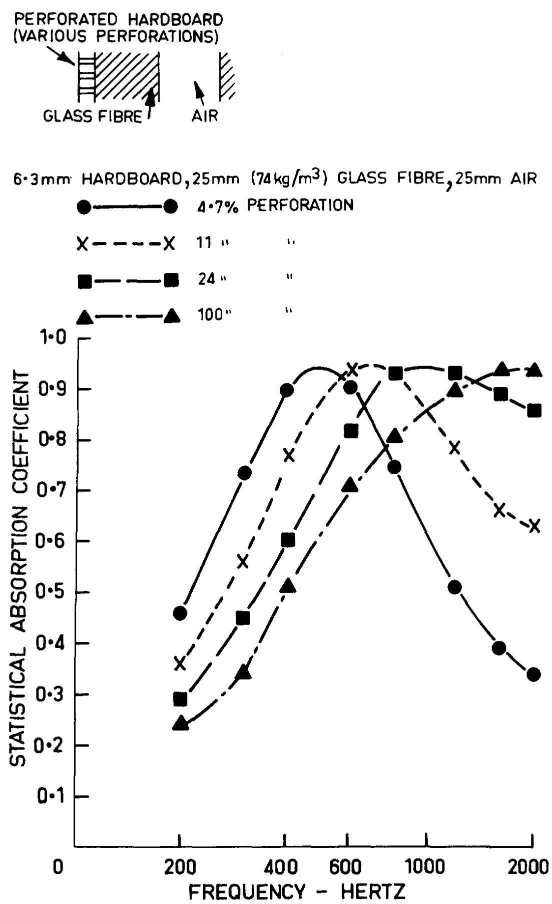
The effect of hole size is not great. So, a panel perforated with a lot of small holes and a panel perforated with less bigger holes are comparable except for very high sound pressures and at the extremes holes sizes.



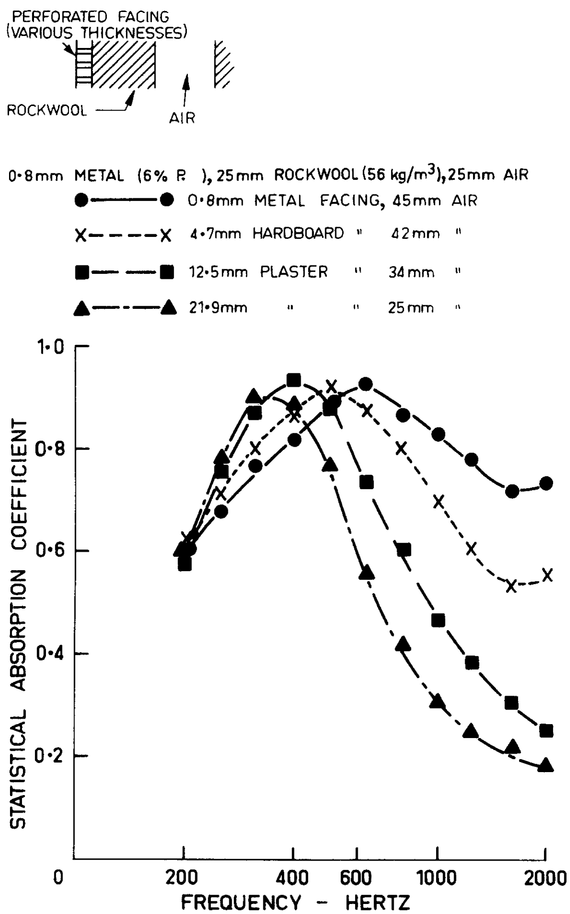
8.1.4. Thickness of the facing

The experimental study of W.A. Davern (1977) shows that thicker facings have a maximum absorption at a lower frequency than thinner facings. Besides that, thicker materials have a broader range of frequencies which they can absorb. The absorption coefficient does not change by changing the thickness of the plate.

It can be seen from this result that with a change in facing thickness a different percentage perforation will be needed to achieve a particular design criterion.



Fig,4. *Absorption coefficient versus frequency graphs for a perforated faced sound absorber system for different percentage perforations of the facing. (Davern 1977)*



Fig,5. *Absorption coefficient versus frequency graphs for a perforated faced sound absorber system using different thickness of facing and cavity with the same percentage perforations. (Davern 1977)*

8.1.5. Density of the porous backing material

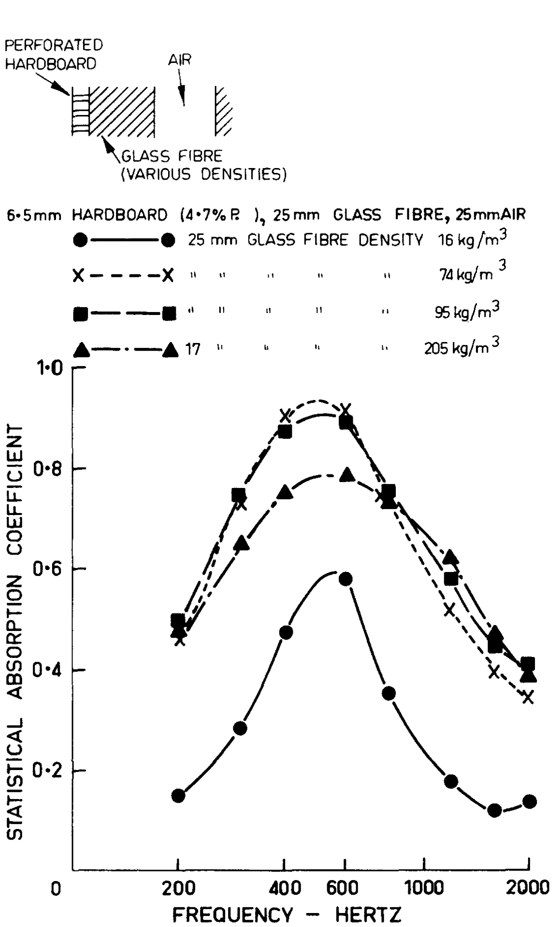
When the density of the porous backing material changes, the airflow resistance automatically changes too. Figure 6 shows the results of different densities. A change in density hardly affects the resonance frequency. In fact, a typical Helmholtz resonator situation applies where the addition of material has broadened the tuning of the system.

The 16 kg/m3 glass wool gives a highly tuned system and the higher densities give a more broadly tuned system.

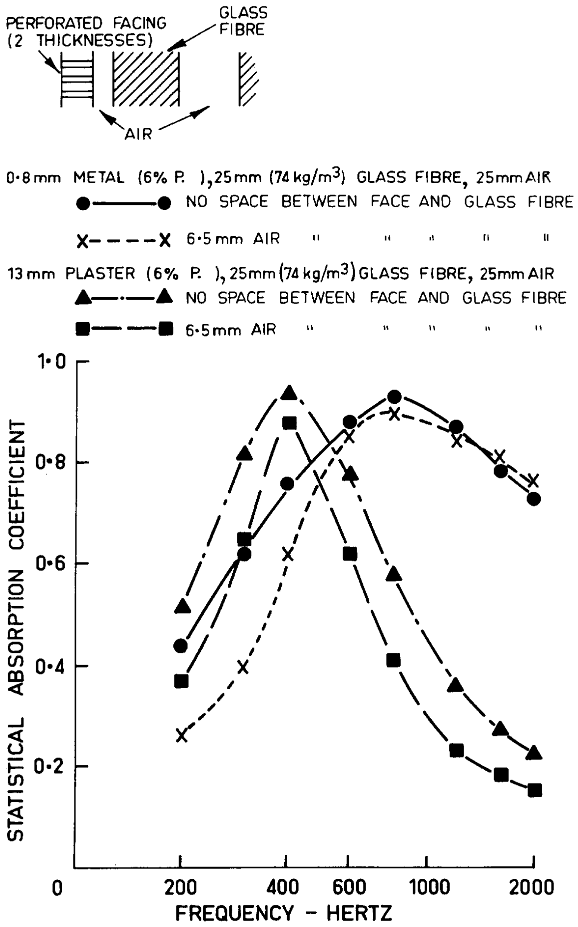
Apparently, an ideal density exists for the height of α. In figure 6, this is about 74 kg/m^3.

8.1.6. Air space between the facing and backing material

Figure 7 shows the change that occurs when a 6 mm air space is introduced between the perforated face and the backing material for two different thicknesses of the facing. The resonance frequency has remained unchanged. The absorption coefficient decreases when a layer of air is added.



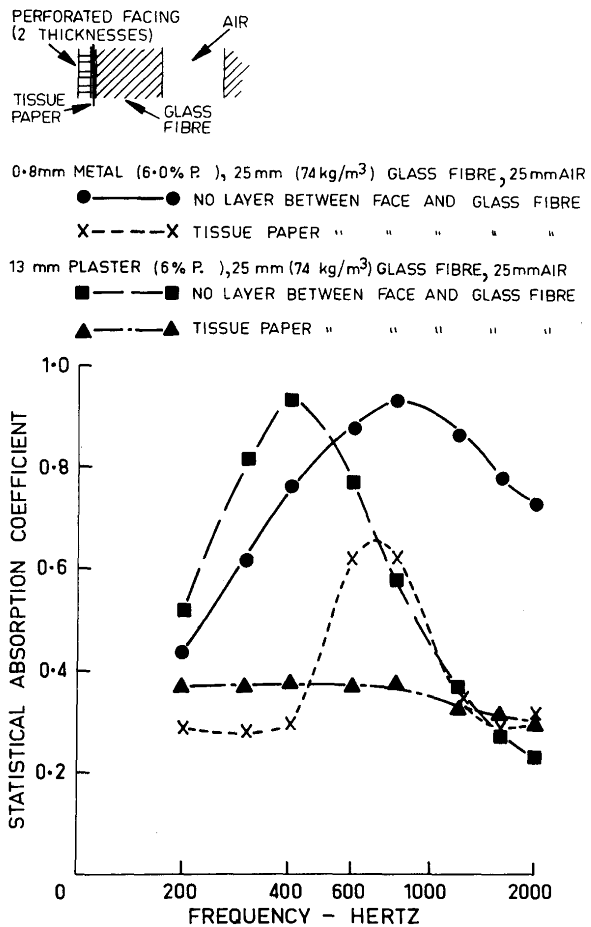
Fig,6. *Absorption coefficient versus frequency graphs for a perforated faced sound absorber system using porous backing materials of different densities. (Davern 1977)*



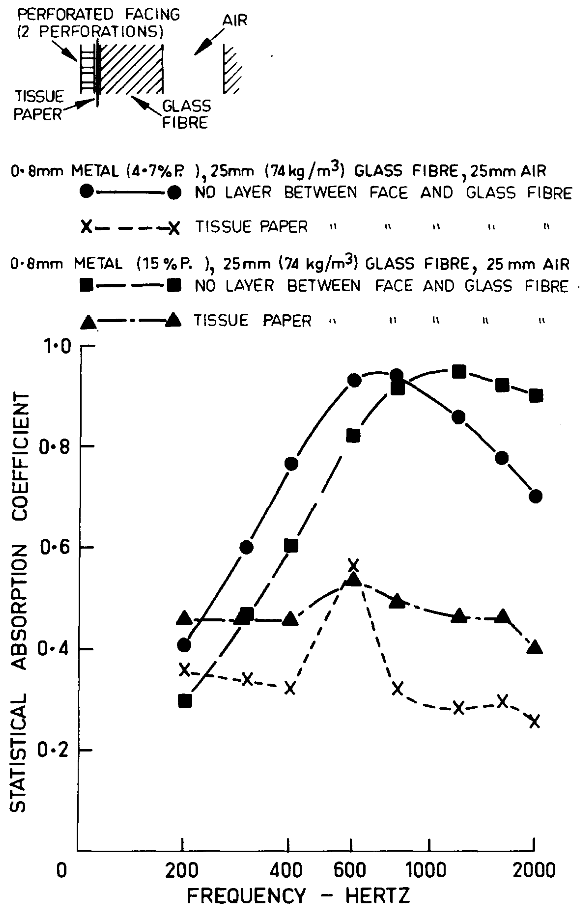
Fig,7. *Absorption coefficient versus frequency graphs for a perforated faced sound absorber system with and without an air space between facing and porous backing with the same facing perforations but different facing thickness. (Davern 1977)*

8.1.7. Impervious layer between the facing and backing material

In order to prevent small particles of mineral wool falling through the perforations of acoustic ceiling tiles an impervious layer, e.g. thin plastic film, is sometimes inserted between the facing and the backing material. Figure 8 shows the effect of adding such a layer. With such a modification major changes occur in the acoustic properties. The addition of tissue paper in the test situation has reduced the sound absorption properties considerably. The same test with different percentages of perforations in the metal sheet gives similar results. See figure 9.



Fig,8. *Absorption coefficient versus frequency graphs for a perforated faced sound absorber system with and without an impervious layer between facing and porous backing with the same facing perforations but different facing thickness. (Davern 1977)*



Fig,9. *As in figure 13. but with different facing perforations and the same facing thickness. (Davern 1977)*

III. Measurements

9. Scale model measurements 1

The research on the acoustical behaviour of perforated steel panels starts here. This chapter will describe the measurements done in a scale model of a sports hall. In this scale model, the effect of the shape of the roof on the absorbing properties of the roof is investigated. This chapter will start with the theory behind scale model tests. After that, the measurement method will be explained. Finally, the results and conclusions will be discussed.

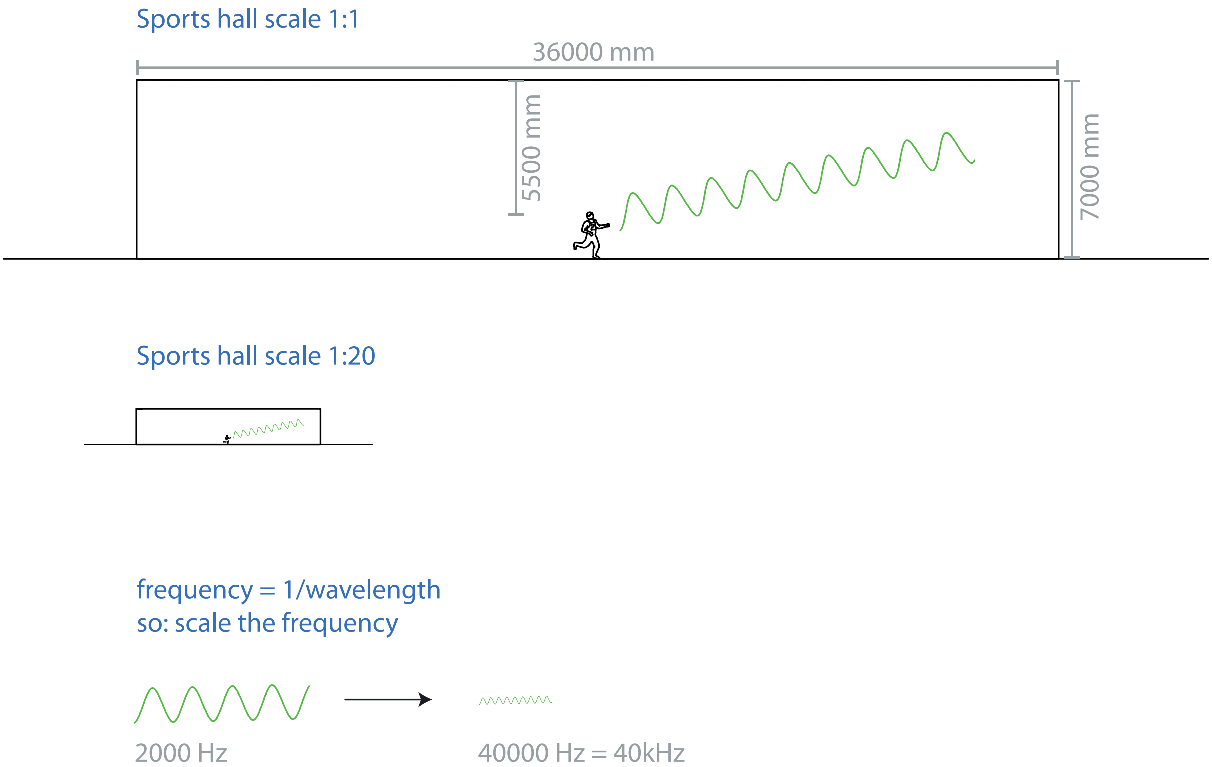
The scale model tests are divided into two parts:
1. Measurements in the 1:20 model of a sports hall with 2, 4 or 6 walls,
2. Measurements on the reflection coefficients of the different profiles.

This chapter will first focus on the measurements in the model. In the next chapter, the reflection coefficient measurements are discussed.

9.1. Theory behind scale model tests

Scale models can be used to study different things. Architects use scale models to check the shape, size, texture and other aspects of their design. Also in acoustical design, scale models can be useful. Those models make it possible to study sound propagation in built-up areas or give extra information about phenomena like scattering, diffraction and propagation over absorbing surfaces. The nice part of scale models is that the mentioned phenomena behave on scale. So, it does not matter on which scale you work, the behaviour will be the same, but scaled.

In order to maintain the ratios between the geometrical dimensions and the wavelength of the sound in reality and in the scale model, the frequencies of the original signal have to be multiplied by the scale factor. The upper limit of the scale factor is about 100, which is set by the microphone dimensions and by the sound absorption of air. (Nijs, 1977) The upper limit of the frequencies is about 180 kHz although there can occur problems when using those high frequencies.

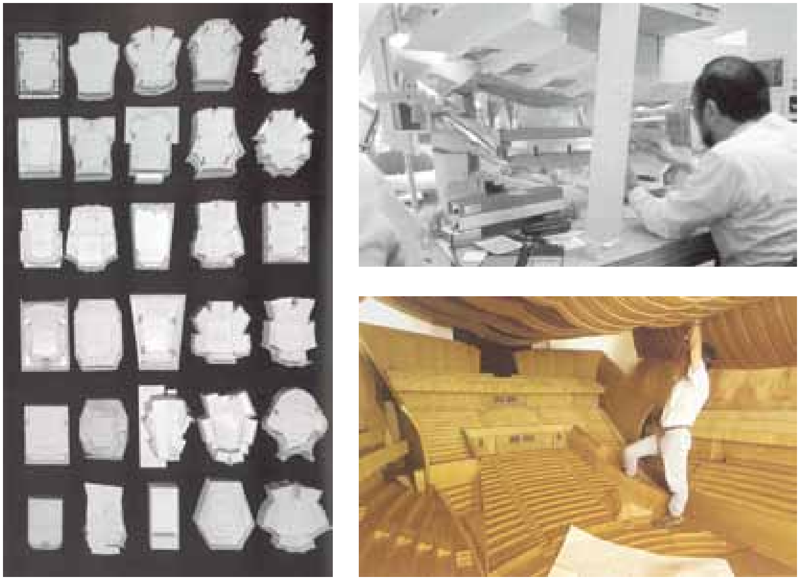


Fig,1. *Scaling models and frequency*

Other use of scale models

Scale models have been often used to design concert halls. The pictures show some scale models of Frank Gehry. ‘The problem with such models is the ‘translation’ of the acoustical results into a measure for the subjective ‘quality’ of the hall.’ (Nijs 1977)

Besides for concert halls, tests on scale models are very useful for research on traffic noise. The sound level is the only parameter that counts. The results of those measurements can be easily compared to standards.

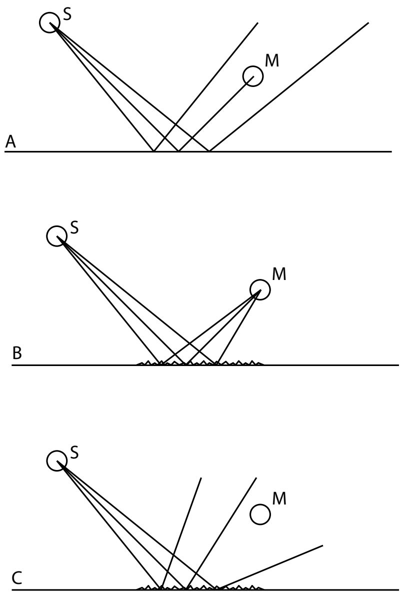


Fig,2. *Scale model tests, Frank Gehry.*
(Kolarevic, 2003)

9.1.1. Scattering effects

If a sound wave strikes a surface, it will be reflected. A ‘flat’ surface (A) will reflect the sound in one direction, corresponding to ‘the angle of reflection’ (specular reflection). On a rough surface, the sound will be reflected in multiple directions. This could lead to an increase of the sound pressure (figure 3): see B, or a decrease: see C. Normally, a decrease will occur.

It is not easy to calculate scattering effects, but a scale model is an excellent tool to measure these effects.



Fig,3. *Scattering effects*

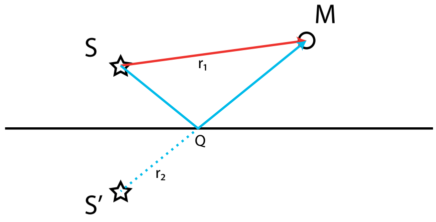
9.1.2. Ground effect

The ground effect is the absorption of sound incident on the ground surface. It consists of the absorption of the sound by the ground surface and the reduction of sound due to a phase shift between incident and reflected sound waves. (Quin 2009)

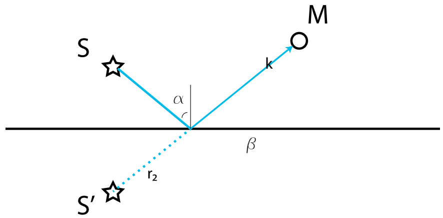
How materials absorb sound is explained in part II of this report. This section will focus on the phase shift principle since this is necessary to understand the scale model test results.

Interference pattern

A pulse, produced by sound source S is received by microphone M. The received sound consists of the direct sound (red) and the reflected sound (blue). The reflected sound can also be seen as a sound coming from imaginary source S'.



After an incident sound wave is reflected by the surface, it has often changed. A part of the sound is absorbed by the material. Besides that, the phase of the reflected sound wave might be different from the phase of the incident sound wave. This is represented by the 'reflection coefficient' Q of the surface, which is slightly different from the reflection coefficient r of the surface. Parameter Q influences the sound pressure of the reflected part of the sound wave.



Fig,4. *First reflection dimensions*

Factor Q depends on the following aspects (see diagram):
- the angle of incidence α,
- the wave number k,
- the distance r_2,
- the specific admittance β of the material.

The wave number k can be calculated from:

$$k = \frac{2\pi f}{c}$$

With:

$k =$ wavenumber

$f =$ frequency

$c =$ propagation speed = 343 m/s

The specific admittance β of the material is based on the impedance of the material (see also chapter 8.1).

The exact theory behind the Q value of a material is very complex. Since it is almost the same as a reflection coefficient, the next part of this explanation will be based on the more easy to understand reflection coefficient r_p. The reflection coefficient r_p can be calculated from:

$$r_p = \frac{\cos(\alpha) - Z}{\cos(\alpha) + Z}$$

With:

$\alpha =$ the angle of incidence

$Z =$ the acoustical impedance

$$Z = \frac{p}{v \cdot S}$$

With:

$p =$ pressure at a boundary surface

$v \cdot S =$ the sound flux (flow velocity times area)

When the sound strikes the surface, so the angle of incidence is 90 degrees, the reflection coefficient will be -1. This explains why the angle of incidence is important for the reflection properties of the material.

The sum of the sound pressures of the reflected and the direct part of the sound wave in point M can be calculated from:

$$\frac{p_{total}}{p_{direct}} = 1 + \frac{r_1}{r_2} \cdot r_p \cdot e^{-jk(r_2 - r_1)}$$

With:

$r_1 =$ distance of direct sound

$r_2 =$ distance of indirect sound

$r_p =$ pressure reflection coefficient of panel

$e =$ Naperian base

$k =$ wavenumber

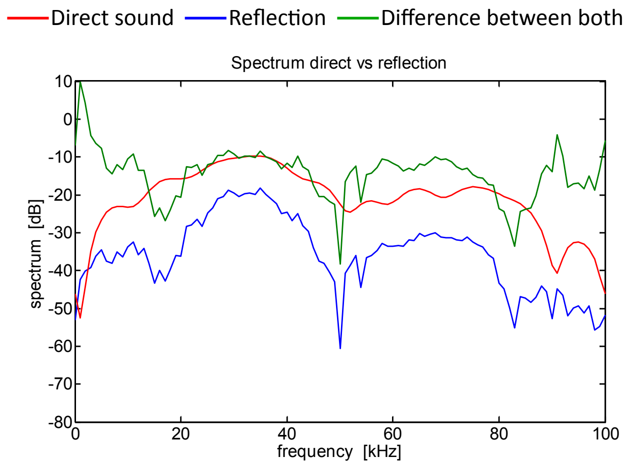
$\mathrm{j} =$ imaginary unit

$$p_{dir} = \frac{e^{-jkr_1}}{r_1}$$

$$p_{reflected} = r_p \frac{e^{-jkr_2}}{r_2}$$

$$p_{total} = \frac{e^{-jkr_1}}{r_1} + r_p \frac{e^{-jkr_2}}{r_2}$$

The graph on the right shows the spectrum of the direct and indirect sound. Valleys are clearly visible. These valleys occur in a rhythm. This is called the interference pattern. The red line of the direct sound does not show any interference valleys, the reflected sound (blue line) does. This is because of the just explained Q value of the surface. The green line, which shows the differences between the two is added to make the outcome independent from the type of sound. The scale model measurements are done with a spark as sound source. This spark is never the same. The green line in the graph can be used to compare different results from different spark sources.

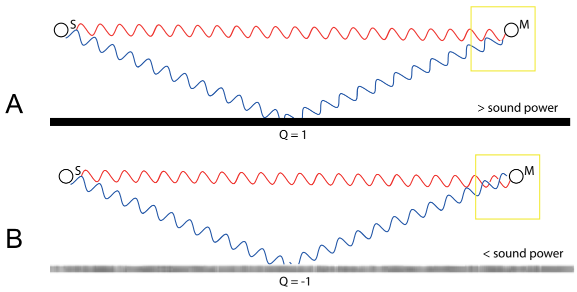


Fig,5. *Profile A, measurement 2, reflection t=6-7 ms.*

The phase shift effect mostly occurs for frequencies between 100 Hz and 1000 Hz because the chance that short sound waves interfere out of phase is smaller than that long waves do. In this research, the ground effect might also be called ‘the roof effect’ since the interference effect mostly takes place at the roof surface of a sports hall.

The influence of the type of material

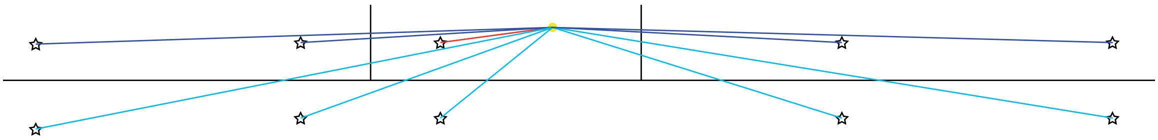
Hard surfaces with high reflection coefficients like the floor of a sports hall will reflect the incident sound in phase. That means that the phase is not shifted. When the reflection and the direct sound meet, the sound power level will increase (see A). Softer materials like mineral wool both absorb the sound and change the phase of the reflected sound wave. The changed phase of the reflection and the ‘normal’ phase of the direct sound will interfere with each other and it will lead to lower sound levels at the receiver (see B in figure 6). Parameter Q approximates the ground effect and varies between -1 and 1. Hard ground surfaces have a Q close to 1, soft surfaces go to Q = -1.



Fig,6. *Effect of different Q value*

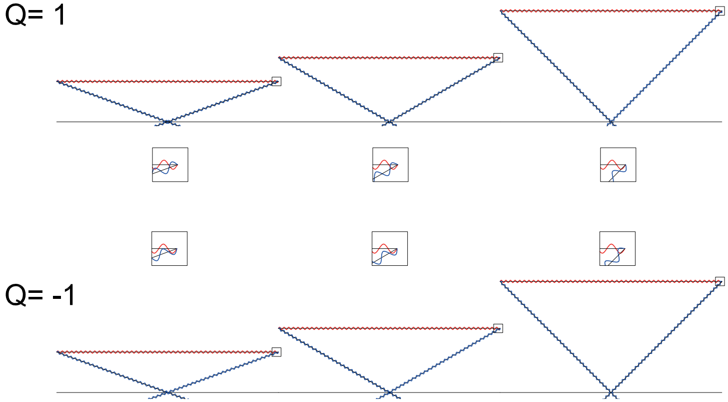
The influence of the angle of incidence

As discussed before, the angle of incidence is important when calculating the phase shift. In practice this means that the further away the ‘mirror sources’ are that reach the microphone by a reflection of a hard ground surface (light blue lines), the lower the Q value will be.



Fig,7. *Mirror sources*

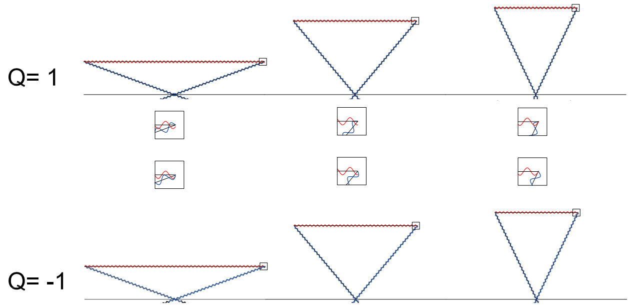
When the ground surface is hard (Q=1), a bigger angle of incidence will lead to a lower Q value: the sound waves will get out of phase.
When the ground surface is soft (Q=-1), a bigger angle of incidence will lead to a higher Q value: the sound waves will get in phase.
This effect does not change with the length of distance r2, the reflection distance. This is shown in figure 9.



Fig,8. *Effect of different angle of incidence on Q value*

The influence of the frequency

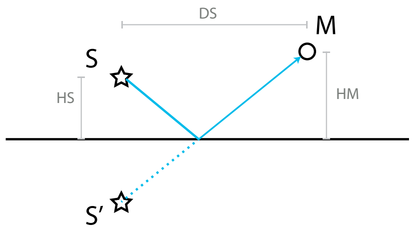
‘An examination of the 250 Hz and 500 Hz bands shows how the sound is much more highly attenuated in these bands - where the ground effect is greatest- than in the adjacent ones.’ (Quin 2009)



Fig,9. *Effect of different r2 distance on Q value.*

Interference frequencies

The graph showing the spectrum of the sound on the next page, shows a clear interference pattern. The interference frequencies (maxima and minima) can be calculated with the formula shown in figure 10. This formula is an approximation. It only applies when DS>>HM/HS.
Figure 11 shows the difference between interference maxima and minima. For a minimum, the sound will extinct, a maximum will amplificate the sound.



$$f = \frac{(2n-1)c}{2(r_2 - r_1)}$$

With:

$(r_2 - r_1) =$ Path length difference between direct and reflected sound waves

$$f = \frac{N \cdot c \cdot DS}{4 \cdot HS \cdot HM}$$

With:

N= 0, 2, 4, 6, 8... for maxima

N= 1, 3, 5, 7, 9... for minima

c = propagation speed = 343 m/s

HS, DS and HM: see diagram [m]

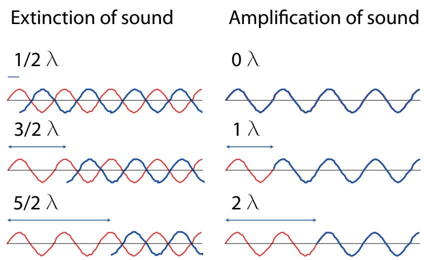
Fig,10. *Formula (top) and approximation (bottom) to calculate interference frequencies.*

Expected interference pattern

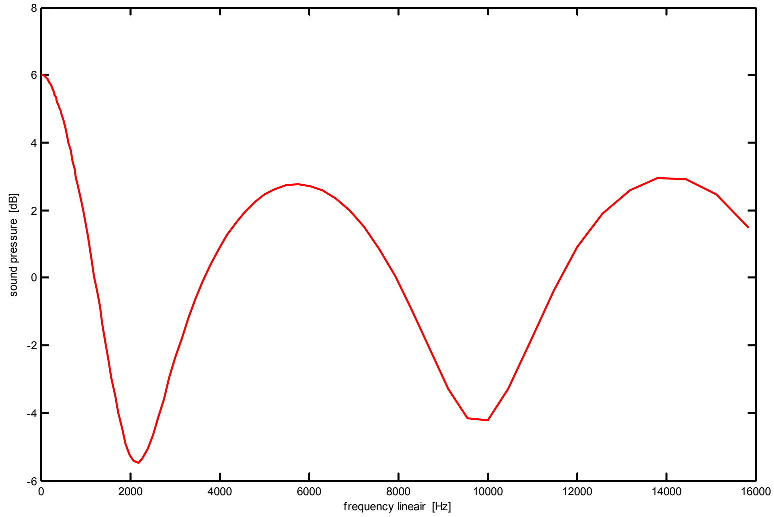
The interference frequencies as discussed before can be calculated for the measurement setup beforehand:
DS= 72 cm, HS= 9.7 cm, DS= 14.6 cm.
For the first reflection, a sound coming from source S', the interference frequency is 4.4 kHz. The dips in the spectrum graph of this reflection will occur at 4.4 kHz, 13.2 kHz, 22 kHz etc.

Also for later reflections, the interference frequencies can be calculated. An example:

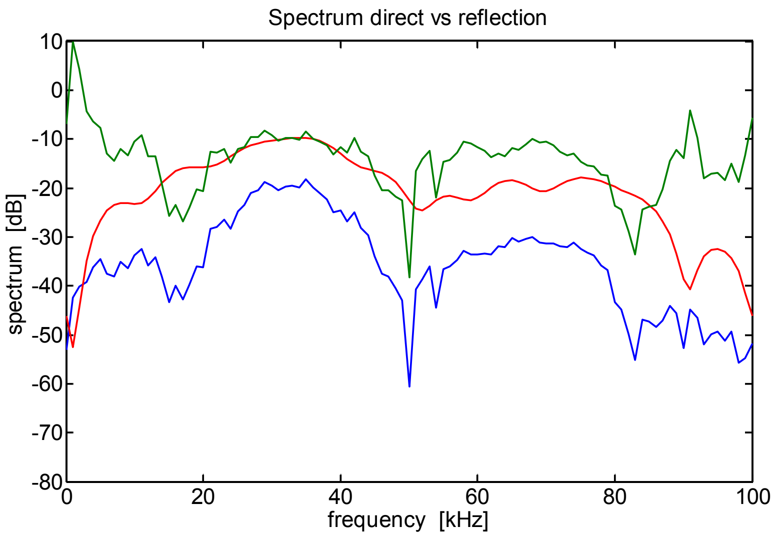
When the reflection takes place after 6.3 ms, distance DS will be: 0.72m + 0.0063sec * 343m/s= 2.88m.
The interference frequencies (minima) will be: 17 kHz (N=1), 52 kHz (N=3), 87 kHz (N=5). See figure 13 on next page.



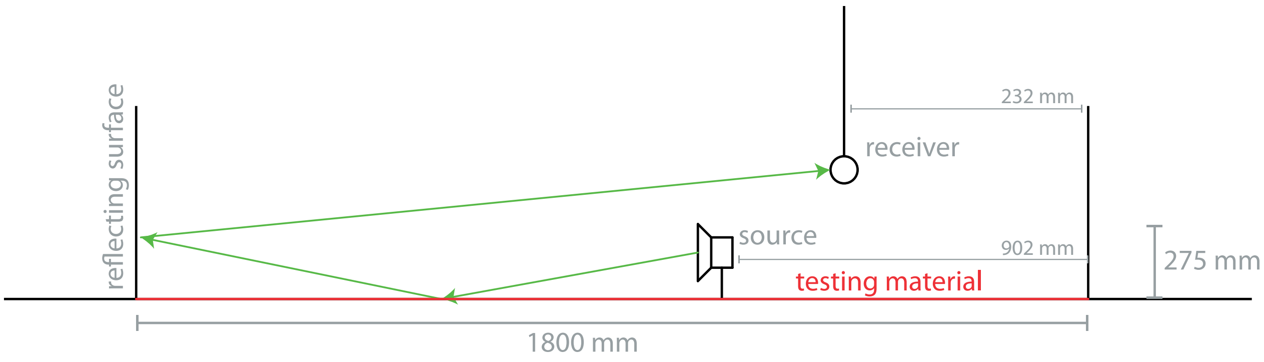
Fig,11. *Interference maxima and minima*



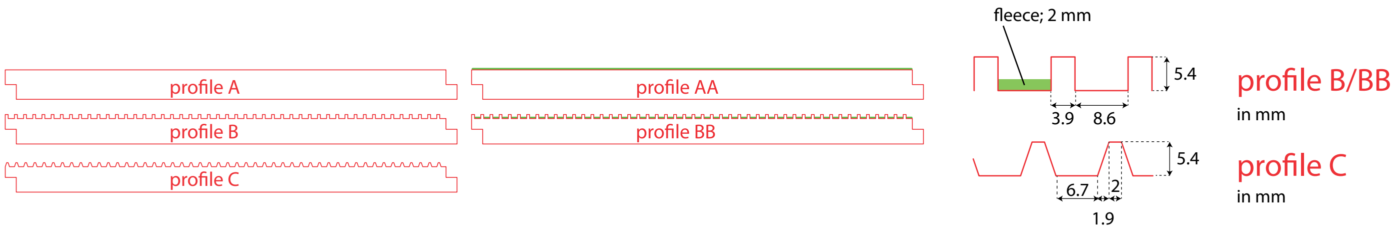
Fig,12. *Interference pattern for DS = 72 cm*



Fig,13. *Interference pattern in spectrum of reflection after 6.3 ms*



Fig,14. *Test method scale 1:20*



Fig,15. *Testing materials*

Fig,16. *Size of model*

Fig,17. *Two walls - configuration*

9.2. Measurement setup

9.2.1. Testing materials

The measurement setup is shown in figure 14. The red line refers to the testing material, which are profile A, AA, B, BB and C (figure 15). Profile A is a flat surface, created from one painted MDF plate. Profile B is a ribbed surface with perpendicular slots, created from a milled and painted foam board. Same for profile C, which has the shape of a 'real' roof structure. The slots are tapered, it is based on the size of SAB panel type 106+/750, which is also used in the laboratory measurements.

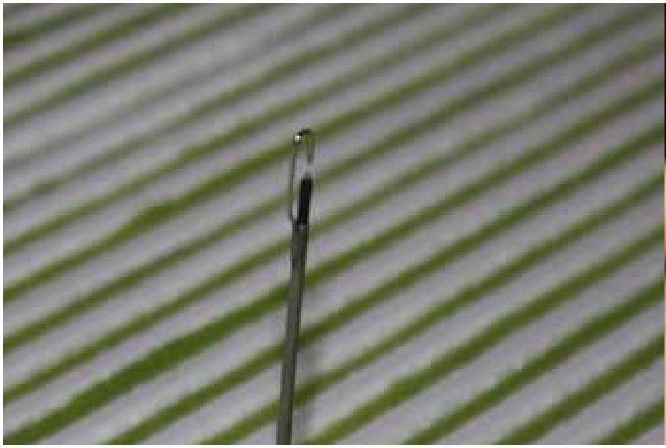
Profile AA and BB are like profile A and B, but some absorption is added. Therefore, a thin (2.5mm) layer of fleece is used. The fleece in profile BB, imitates the sound absorption in the fluting of perforated steel panels. The total surface of profiles B, BB, and C is created out of three of the same panels. The seams between the panels are small (2 mm). All profiles are painted with the same paint.

9.2.2. Scale

The used scale of the model is 1:20. This scale is chosen in a way that the dimensions of the slots in the panels are not too small and the frequency of the sound source is not too high. Although the material is milled, very small slots would give small differences, which are big differences on scale 1:1.
Very high frequencies are not easy to produce and might be absorbed by air. Besides that, scale 1:20 would give more clear results than a model in scale 1:50. Finally, the fleece strips would not fit in a smaller model.

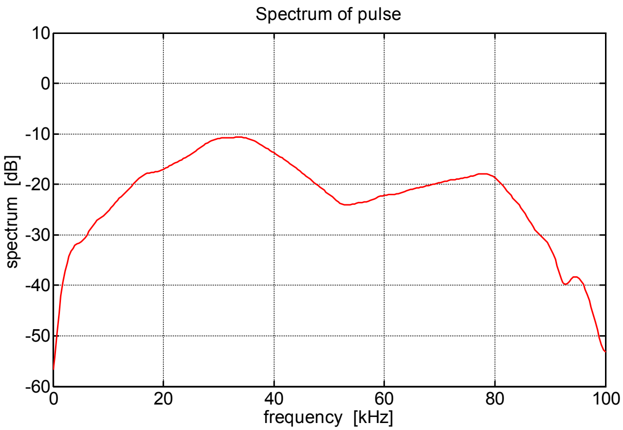
9.2.3. Size of the model

The (interior) model size is: 1200 x 1804 x 350 mm. This corresponds to a sports hall of 24 x 36 x 7 meter. Besides the tests on the profiles with two walls and with four walls, tests in a complete scale model of a hall are done too. The floor and walls are materialized by painted wooden panels without seams.
The ISA-Sport standard gives a maximum reverberation time of 1,5 seconds for type B3: 32 x 28m and 1,6 seconds for type C1: 24 x 44m. Those limits are comparable for the used scale model.

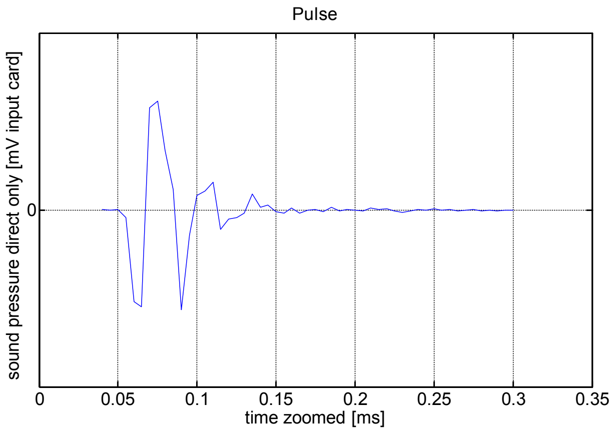


Fig,18. *The sound source*

Fig,19. *The microphone*



Fig,20. *Spectrum of spark*



Fig,21. *Impulse*

9.2.4. Sound source

The used sound source is a spark source, type GE 31001 C10 EN.

The produced sound is a pulse evacuating from an electrical spark. The direct sound and the various reflections are separated in time due to the differences in travel times. The time plot is converted to a frequency spectrum. The main advantage of a pulse source is the ability to cut out any part of the time plot that is necessary. Besides that, the pulse can be corrected for the sound absorption of sound in air.

Figure 21 shows the sound pressure level of an impulse. When we have a look at the spectrum of the impulse, we see that the microphone has some problems receiving the very high frequencies (90kHz and higher). This is visible in the graph. The spark source can not produce low frequencies well. This is also visible in the graph.

Different pulses are used for one measurement. During 6 seconds, about 5 pulses are registered. When the differences between the pulses are too big, the measurement is repeated. In this way, a comparable pulse is used for all measurements.

Since the scale of the model is 1:20 and the frequencies of interest are low, the used frequencies during the measurement differ from 0 Hz to 100 kHz. (Or 0 Hz to 5 kHz in real scale.)

9.2.5. Receiver

The used receiver is a 1/8’’ microphone, type 4136 592785, Bruel & Kjaer, Copenhagen. It changes the acoustical signal into an electrical signal that makes it possible to derive a spectrum of the direct sound and the indirect sound.

The placement of the sound source and receiver is very important. The difference in distance between the direct sound and the first reflection should be big enough in order to not have an overlap of the pulses. In this research, the difference is about 720 mm, which is enough to distinguish the direct from the indirect sound pulses.

‘The microphone and its preamplifier is the weakest link in the measuring chain. The microphone should meet a few contrary demands: it has to be small not to disturb the sound field, it has to be sensitive, it should register frequencies up to 200 kHz.’ (Nijs 1977)

The microphone works on battery power supply. Because of the conversion to an electrical signal, it produces a whistling sound of 90 kHz. This is visible in the reflection coefficient measurement results; see chapter 12.

9.2.6. Other equipment

Mathlab version R2011b,
Computer: Dell optiplex 790,
Amplifier ERIZ77 No. 81.690267, figure 23,
Amplifier Bruel & Kjaer, Copenhagen type 2804, figure 22,
Selective amplifier model 189 No. 81.750108, figure 24.

9.3. Measurement method

The measurements done for this graduation research focus on the differences between direct sound and a reflection of this sound in different configurations. Those differences are visualized with the use of graphs that show the spectrum of the sound.

Because of all reflecting surfaces of the configuration with four walls and the complete box, it is not possible anymore to select a single reflection for the comparison with the direct sound. Therefore, other analysis graphs are used: the histogram and the Schroeder curve. For an explanation of these graphs is referred to the website bk.nijsnet.com. From these graphs, reverberation times are calculated. This method comes very close to a 'normal' reverberation time measurement in a sports hall.

For the last results, a correction of the reverberation times is introduced. This correction takes the sound absorption of air into account. The input for relative humidity is 30%, the temperature is 22 degrees Celsius. The correction of reverberation times is important for the results of high frequency bands. This type of sound can be absorbed easily by air.

Fig,22. *Amplifier for microphone 1*

Fig,23. *Amplifier for microphone 2*

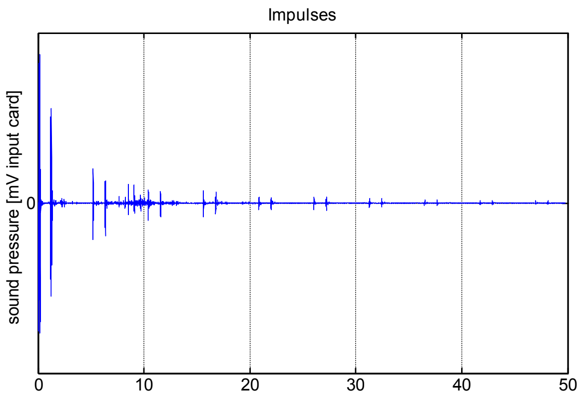
Fig,24. *Amplifier 3*

Fig,1. *Scale model*

10. Results and discussion

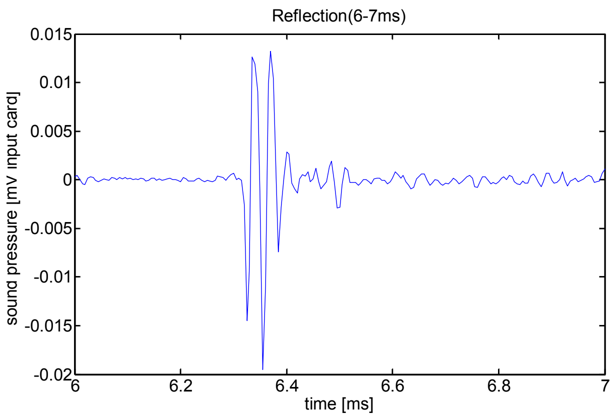
This chapter will give an overview of the results per profile and configuration. It will show the spectra of direct sound compared to two reflections, an early and a later one. Besides that, a graph showing all reflections will be displayed. Next, the results per configuration will be compared. Histograms, reverberation times and Schroeder curves are used. Per section, the results are shortly discussed. Conclusions can be read in chapter 10.14.

For all measurements and data, see Attachment D.

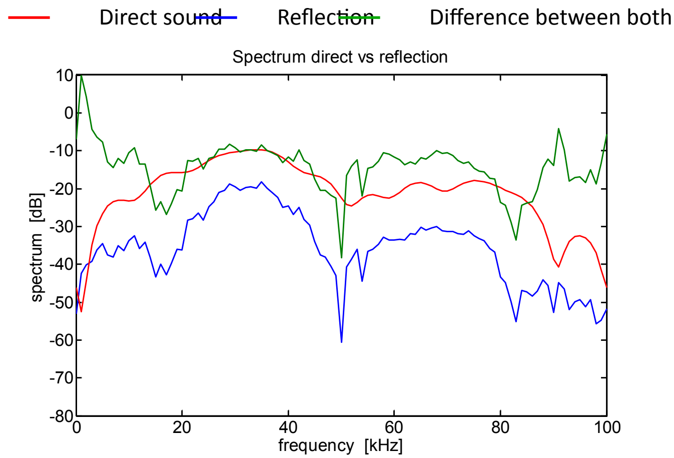


Fig,2. *Impulses*

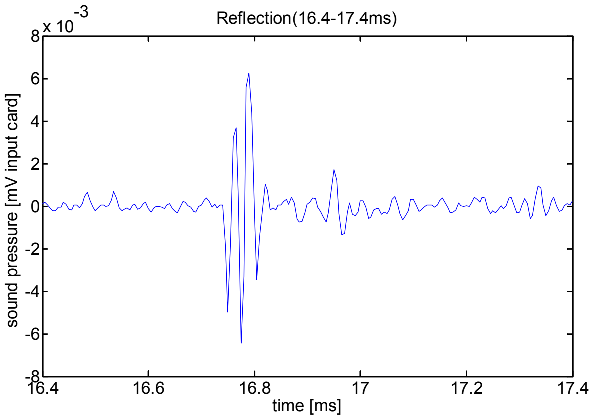
Fig,3. *Picture of measurement setup*



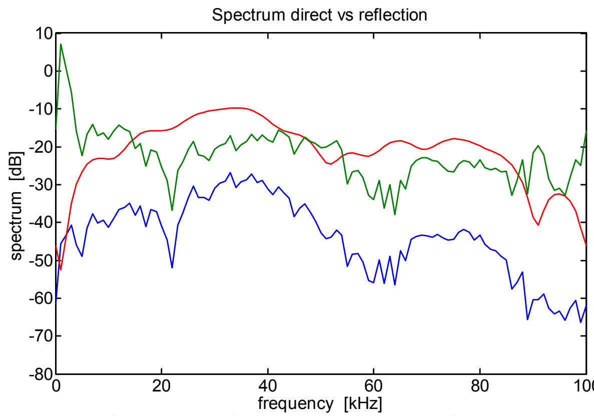
Fig,4. *Reflection 6-7 ms*



Fig,5. *Spectrum of direct sound and reflection 6-7 ms*



Fig,6. *Reflection 16.4-17.4 ms*



Fig,7. *Spectrum of direct sound and reflection 16.4-17.4 ms*

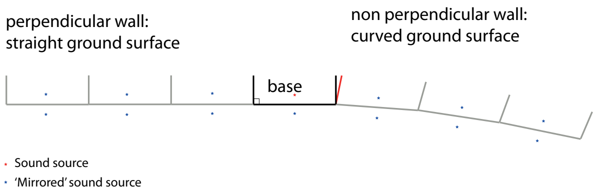
10.1. Profile A with two walls

Data used of measurement 2.

The graph 'Impulses' gives an overview of all reflections between the two walls. The energy of the reflections decreases in time. This decrease is caused by travelling distance, the absorption of sound by the profile, the air and the walls of the anechoic room.

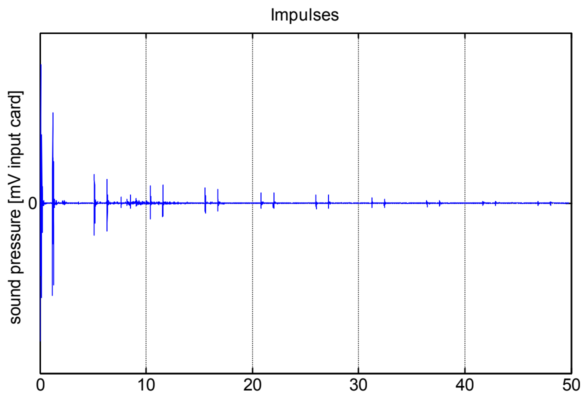
The comparison of the spectra shows clearly an interference pattern. The interference frequencies depend on the ground effect and the angle of incidence. The later reflection shows a bigger difference between two interference frequencies. For the calculation of the interference frequency (for the reflection at 6.3 ms), distance DS should be about: 0.72 m + 343 m/s*0.0063 s = 2.862 m. This leads to interference valleys at 17 kHz, 51 kHz, and 85 kHz. This corresponds to the graph. The interference frequencies of the later reflection should occur at: 39 kHz and 117 kHz. This does not correspond to the graph.

A possible reason for this is the position of the reflecting angled surfaces. If these are not placed exactly perpendicular to the ground (roof) surface, a 'curved' ground surface will occur for the mirrored sound sources. The curvature of the ground surface causes a difference in height HS for the height of the sound source. See figure 8. The use of mirrors as reflecting surfaces could help to prevent from this deviation.



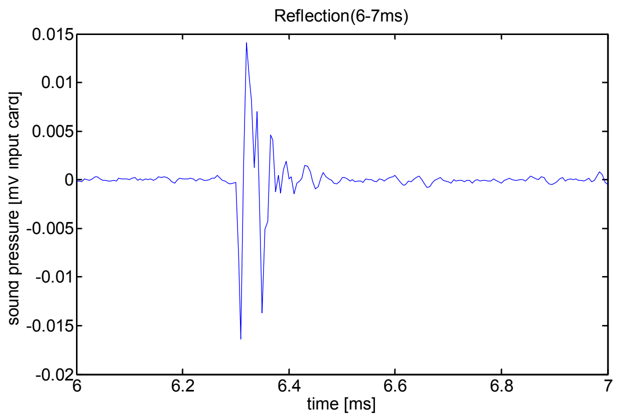
Fig,8. *Curved ground surface, change of heigth HS for imaginary mirrored sound sources.*

Besides that, the blue line of the later reflection (16.4-17.4 ms) is less straight than the blue line of the early reflection (6-7 ms). This shows the disturbing effect of the background noise, the more diffuse sound field and the effect of little deviations in the setup of the measurement. Of course the later reflection has a lower sound pressure level than the earlier one. (See also calculation in figure 16 of next section.)

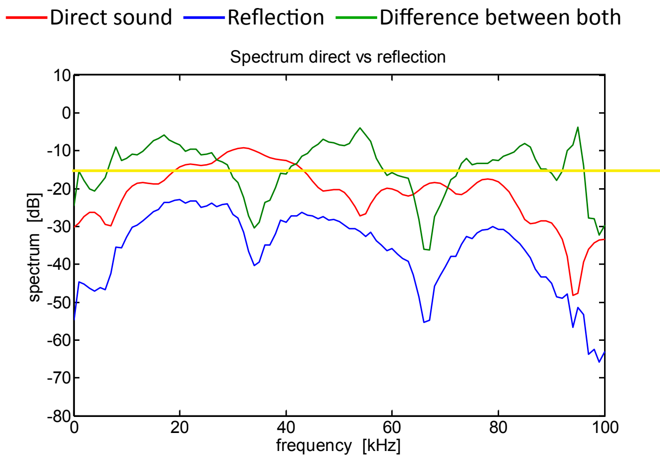


Fig,9. *Impulses*

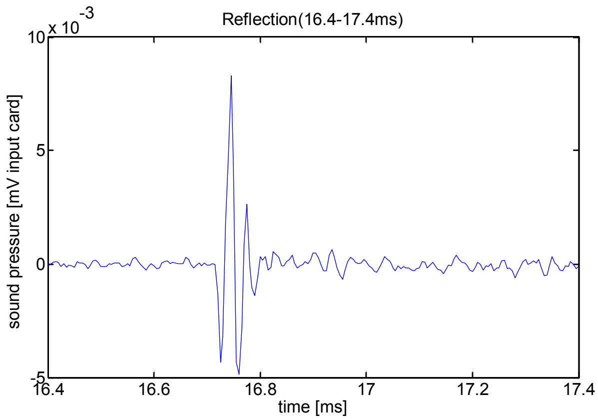
Fig,10. *Picture of measurement setup*



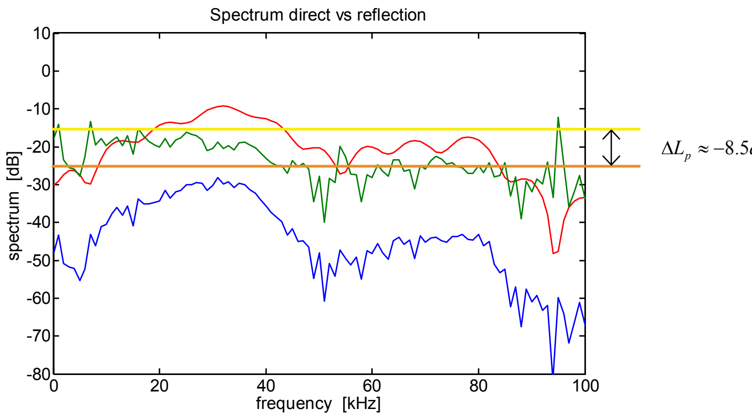
Fig,11. *Reflection 6-7 ms*



Fig,12. *Spectrum of direct sound and reflection 6-7 ms*



Fig,13. *Reflection 16.4-17.4 ms*



Fig,14. *Spectrum of direct sound and reflection 16.4-17.4 ms*

10.2. Profile AA with two walls

Data used of measurement 3.

The graph ‘Impulses’ shows the total of all reflections within a 50 ms time frame. It is clearly visible that the decrease of sound energy is much bigger than for profile A. This is caused by the sound absorbing layer of fleece. Because of this layer, the sound pressure level of the selected reflections is much lower than for profile A.

The spectrum of the late reflection shows clear valleys at the interference frequencies. These are, like they should be, different than for the early reflection (6-7 ms), because the distance DS is different.

It is remarkable that the interference frequencies are in between the interference frequencies of profile A. This is caused by the change of the Q value of the material. The hard surface of profile A has a Q value of about 1, the soft surface of profile AA has a Q value of about -1. This affects the interference pattern for minima:

$$f = \frac{(2N-2)c \cdot DS}{4 \cdot HS \cdot HM} = 0\text{kHz, 34 kHz, 68 kHz} \ldots$$

Fig,15. *Q=-1 interference minima frequencies*

Furthermore, the green line of the late reflection lies lower than the one of the early reflection. Here we can see the decrease of sound pressure (see also figure 14). The tortuous line is caused by the many times that the sound is reflected by a surface. Every time, the sound slightly changes.

dB

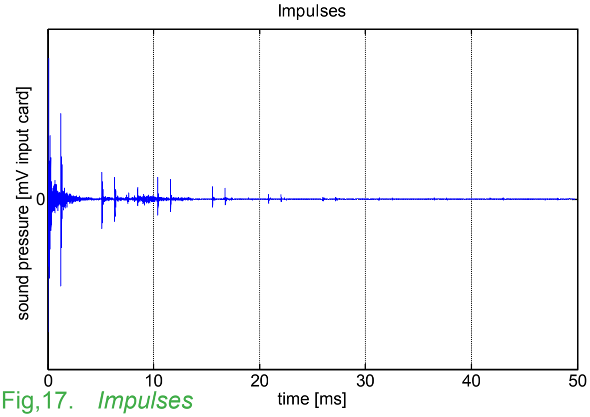
$$\Delta L_p = 10 \log \frac{\boxed{r_2^{\,2}}}{\boxed{r_1^{\,2}}}$$

$$r_2 = 16.7 \cdot 10^{-3} \cdot 343 = 5.73m$$

$$r_1 = 6.3 \cdot 10^{-3} \cdot 343 = 2.16m$$

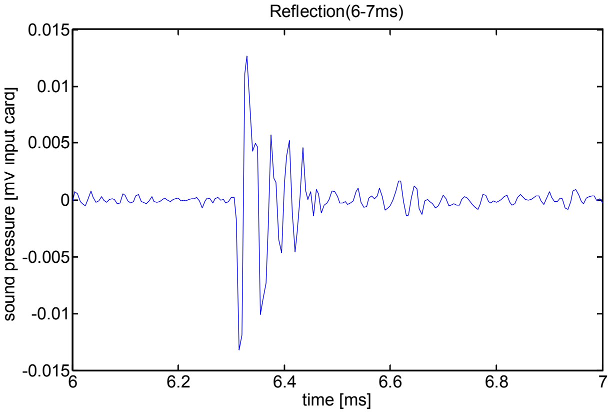
$$\Delta L_p = -8.5dB$$

Fig,16. Δ *L_p calculation*

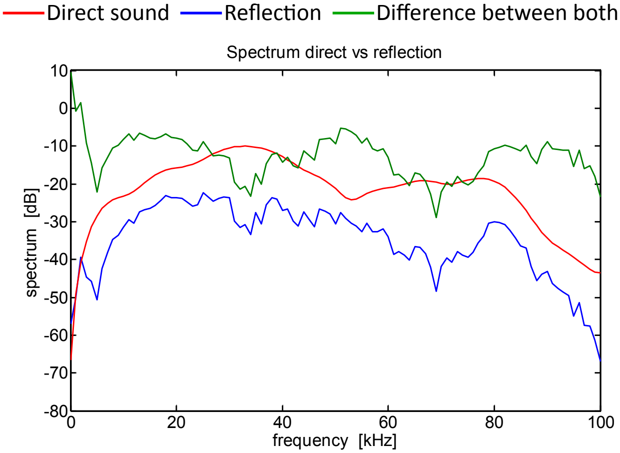


Fig,17. *Impulses*

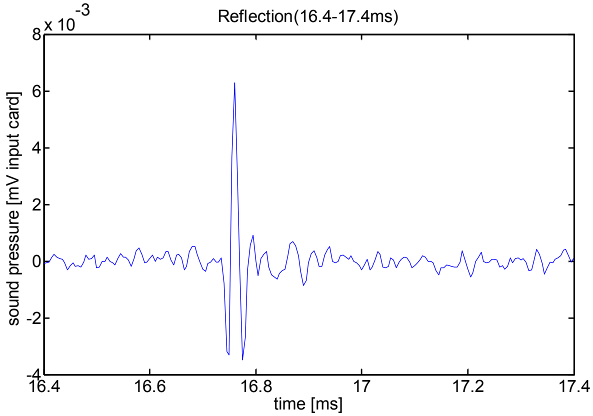
Fig,18. *Picture of measurement setup*



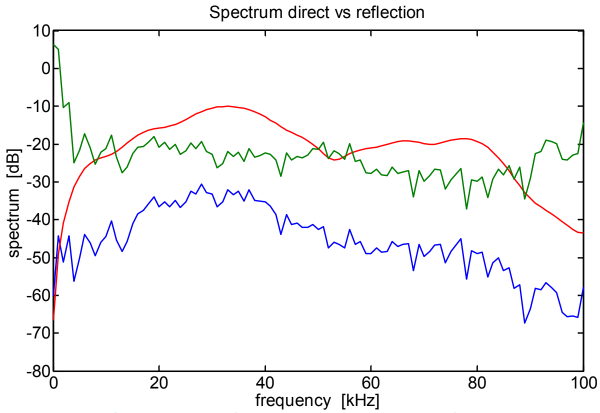
Fig,19. *Reflection 6-7 ms*



Fig,20. *Spectrum of direct sound and reflection 6-7 ms*



Fig,21. *Reflection 16.4-17.4 ms*



Fig,22. *Spectrum of direct sound and reflection 16.4-17.4 ms*

10.3. Profile B with two walls

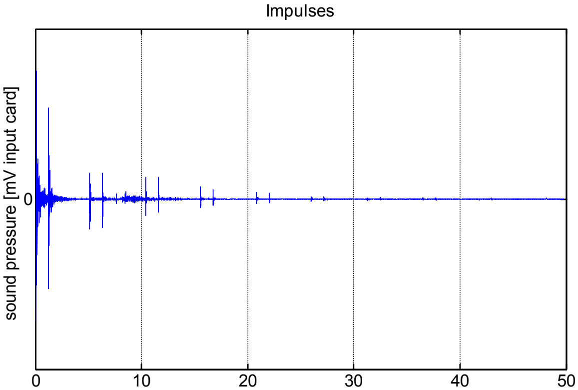
Data used of measurement 3.

The ‘Impulses’ graph of profile B is slightly different than the one of profile A and AA. It is shows more small reflections at 6-7 ms and 16.4-17.4 ms and it takes more time for the reflection of the impulse to ‘disappear’ in the background noise level. This is caused by the shape of the profile. A part of the sound (the bigger part) is reflected in a ‘normal’ way. A small part of the incident sound is reflected in other directions because of the slits in the panel. In this way, the sound field becomes more diffuse. This is clearly visible in the spectrum of the later reflection. Its interference effects have almost gone. Also in the early reflection, the effect of the structure can be seen.

Besides that, the effect of scattering on the sound pressure level is also visible in the graph. A decrease of about 3 dB can be seen when comparing the two (the green line).

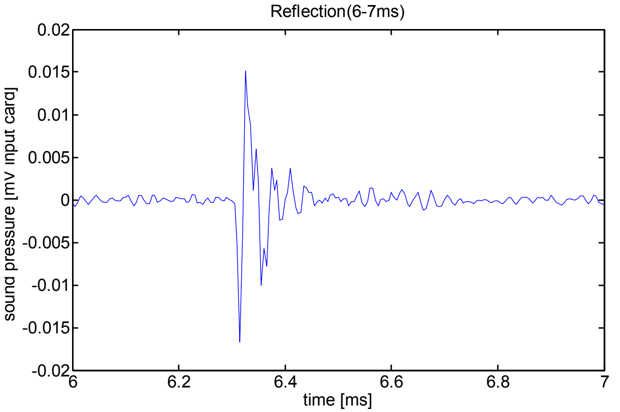
The early reflection (of 6-7 ms) shows an interference pattern. When we have a closer look at the interference frequencies, we notice that these are moved compared to profile A. They are in between two interference frequencies of profile A (for example 50 and 85 kHz, where profile B gives about 37 and 70 kHz).

This means that the Q value of the profile is about -1, comparable to profile AA. Profile AA, though, shows clear reflections from t=0 till 50 ms later (see figure 9). For profile B, the reflections are gone after about 30 ms (see figure 17). Diffusing the sound helps to shorten this time.

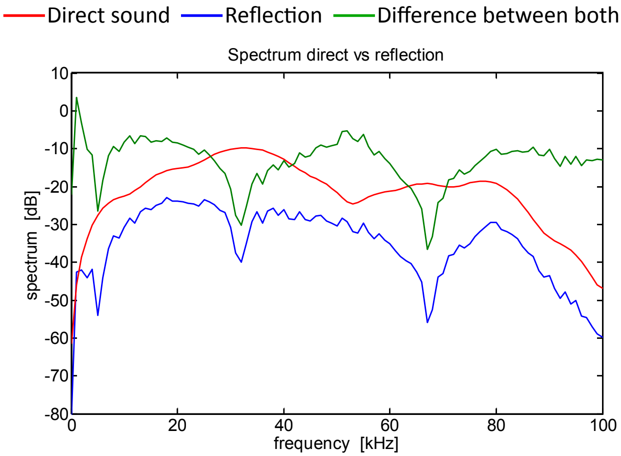


Fig,23. *Impulses*

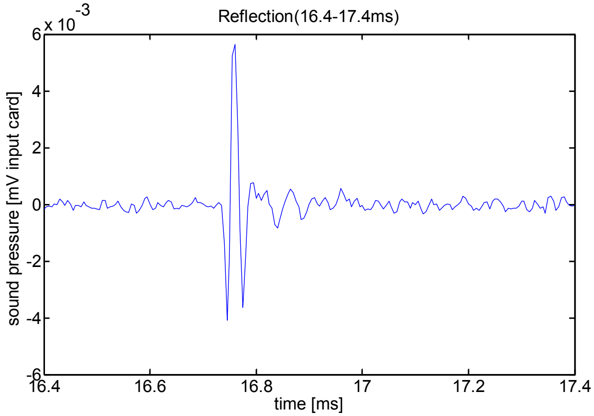
Fig,24. *Picture of measurement setup*



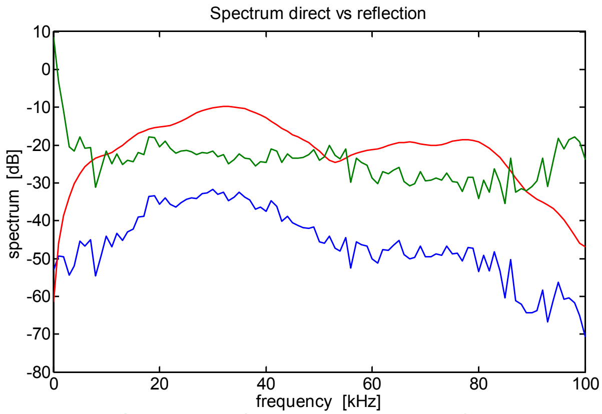
Fig,25. *Reflection 6-7 ms*



Fig,26. *Spectrum of direct sound and reflection 6-7 ms*



Fig,27. *Reflection 16.4-17.4 ms*



Fig,28. *Spectrum of direct sound and reflection 16.4-17.4 ms*

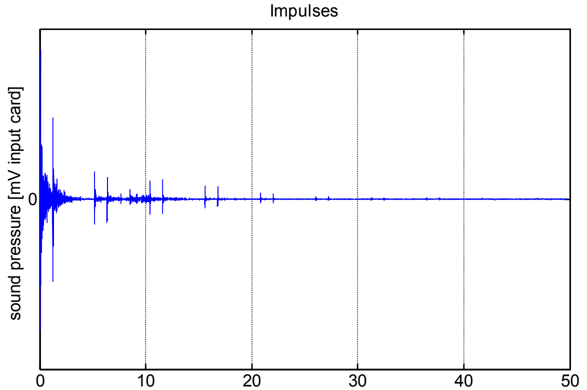
10.4. Profile BB with two walls

Data used of measurement 3.

Now, some sound absorbing material is added to profile B. The effect is visible in the graph ‘Impulses’. The decrease of the reflections is bigger than without fleece strips.

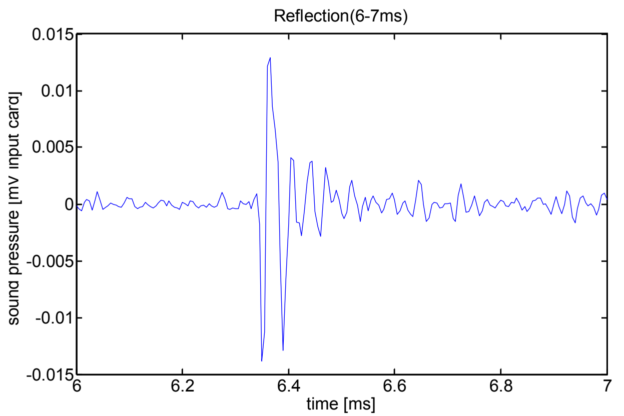
The interference frequencies are comparable to the ones of profile AA. This means that the Q value of the surface is about -1. This is remarkable because it is the same as for profile B. The addition of the fleece strips has little influence on the Q value of this profile.

The later reflection does not give a clear interference pattern. This is caused by the big decrease of sound pressure, the scattering effects of the profile and the many reflections that the sound made before reaching the microphone.

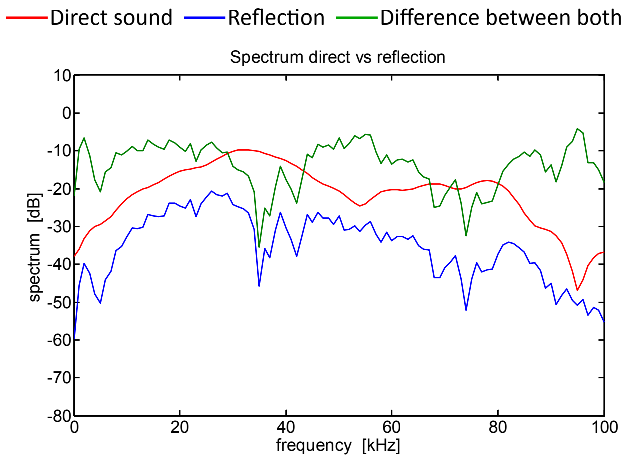


Fig,29. *Impulses*

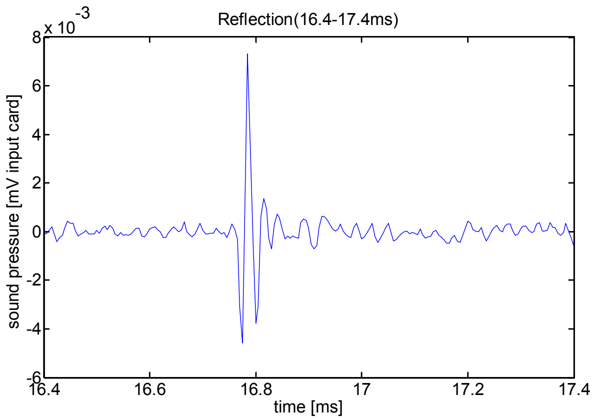
Fig,30. *Picture of measurement setup*



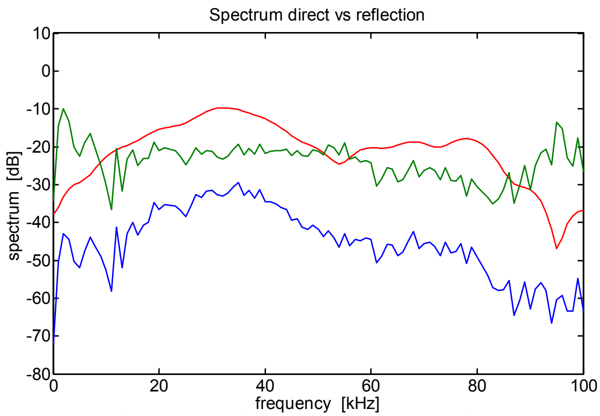
Fig,31. *Reflection 6-7 ms*



Fig,32. *Spectrum of direct sound and reflection 6-7 ms*



Fig,33. *Reflection 16.4-17.4 ms*



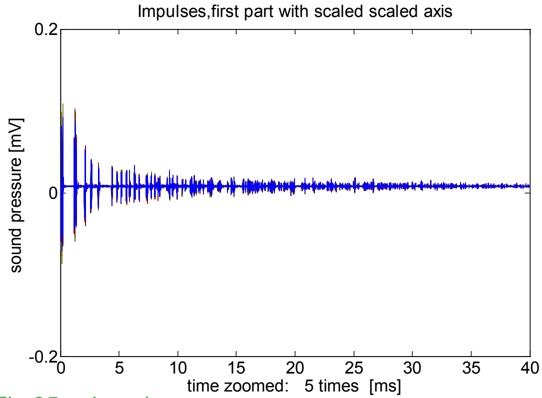
Fig,34. *Spectrum of direct sound and reflection 16.4-17.4 ms*

10.5. Profile C with two walls

Data used of measurement 2.

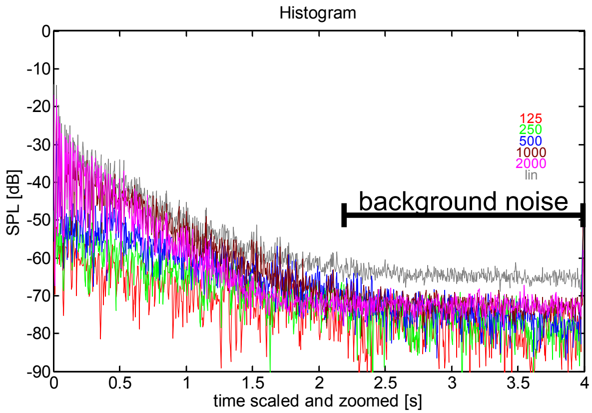
Profile C comes closest to a ‘real’ profile used as a roof structure in sports halls. It has tapered slits. The effect of this shape looks like the effect of profile B. Except for the slightly less ‘absorption of sound’ by the phenomena of diffusion but this will be explained later this chapter.

The interference pattern looks almost the same as for profile B. The effect of scattering and the more diffuse sound field on the later reflection is clearly visible.

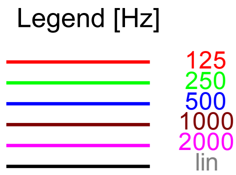


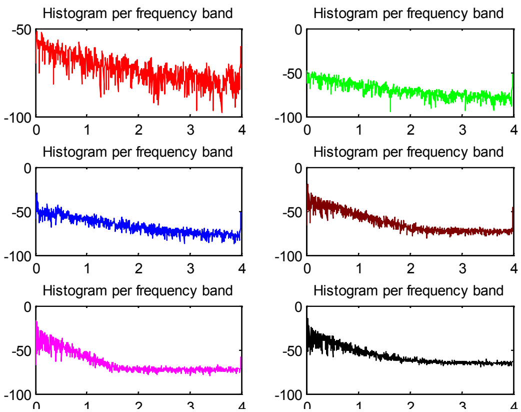
Fig,35. *Impulses*

Fig,36. *Picture of measurement setup*

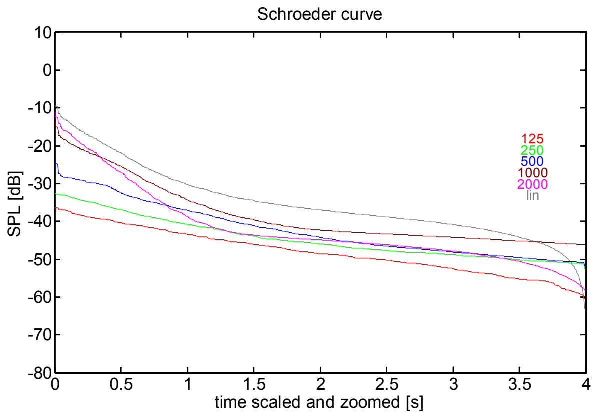


Fig,37. *Histogram*





Fig,38. *Histogram: single-curves only*



Fig,39. *Schroeder curve*

10.6. Profile A with four walls

Data used of measurement 1.

When the same measurements are done with four walls, the graph 'Impulses' changes a lot. The multiple reflections of all four walls and roof are now visible. Because these reflections are close to each other, it is almost impossible to select just one to compare to the direct sound.

The solution is a histogram and a corresponding Schroeder curve. These graphs make it possible to compare the sound per frequency on sound pressure level in time. The graphs are scaled back to scale 1:1. The frequencies are too.

The purple line (2000 Hz) shows a clear decrease, this is caused by the sound absorption of air which is most on high frequencies at high temperatures. Though, just the beginning of figure 37, showing the histogram of the measurement, gives useful information. The horizontal continuation of the curves shows the background noise level.

Reverberation times

	125	250	500	1000	2000	lin
L_p_0	-36.29	-32.50	-24.10	-13.81	-10.91	-8.11
EDT	9.66	7.64	5.27	3.09	1.86	2.23
T_5_15	12.82	11.40	6.08	3.67	2.23	2.93
T_5_25	12.59	16.28	9.59	3.73	2.30	3.62
T_5_35	12.59	16.28	10.94	10.01	4.99	7.94
T_5_45	12.59	16.28	10.94	10.01	8.13	8.21
T_tijd	5.41	6.58	5.77	2.64	1.93	2.14

times between 0.02 0.20 scaled seconds

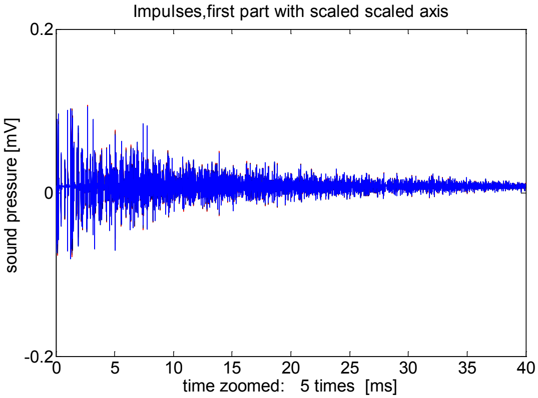
r_EDT	0.99	0.99	0.93	0.93	0.96	0.95
r_5_15	0.99	0.98	0.99	1.00	1.00	1.00
r_5_25	0.99	0.95	0.96	0.99	1.00	0.98
r_5_35	0.99	0.95	0.94	0.81	0.84	0.89
r_5_45	0.99	0.95	0.94	0.81	0.81	0.92
r_time	0.97	0.97	0.63	0.94	0.97	0.96

Fig,40. *Reverberation times*

The blue and red line are more straight in that way.
This lower frequent sound decreases slower than the purple and brown high frequencies at the start.

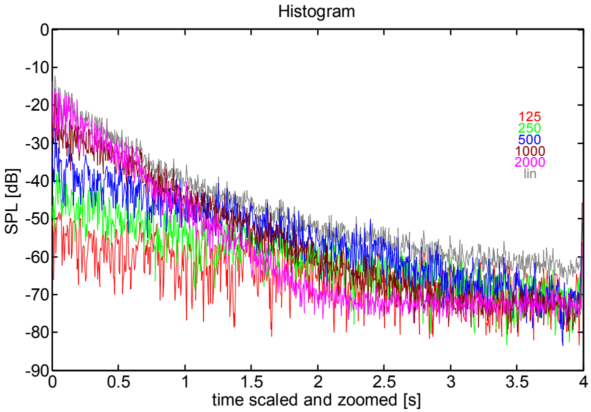
The reverberation times in figure 40 show some strange values. Most of these values are unreliable because of the r_ value. This is a correlation coefficient that gives the reliability of the RT value. When this is below 0.95, it should not be used (gray).

Since the measurements with four walls do not give more interesting results than the ones done in the complete box, I will continue with those results on the next page. All results can be found in Attachment D.

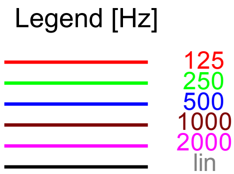


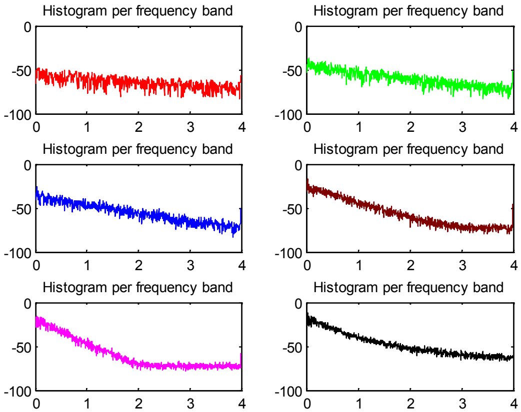
Fig,41. *Impulses*

Fig,42. *Picture of measurement setup*

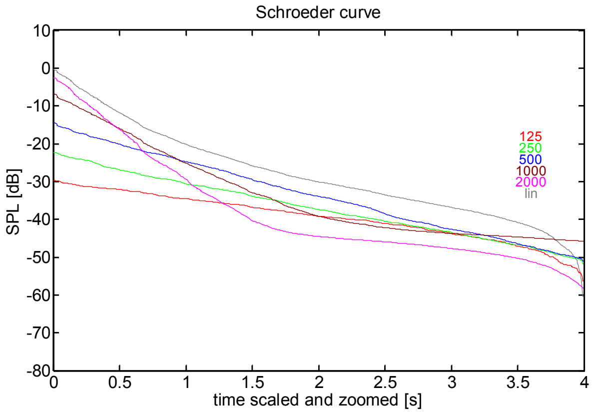


Fig,43. *Histogram*





Fig,44. *Histogram: single-curves only*



Fig,45. *Schroeder curve*

10.7. Profile A with four walls and floor

Measurements in a complete box come close to real life situations. All walls are constructed on scale 1:20. The roof of the model corresponds to the floor of a sports hall. The results are comparable to 1:1 situations.

Data used of measurement 3.

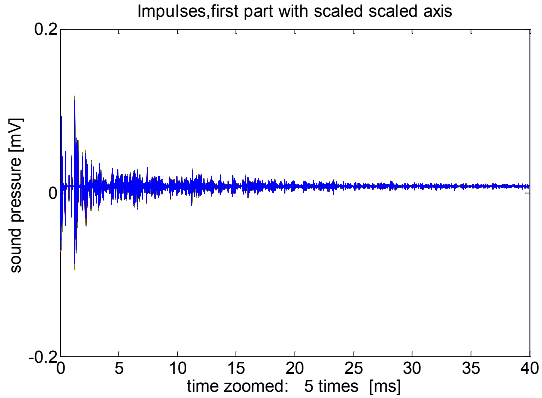
Since the model is a complete box, more reflections are received by the microphone. This can be seen in the graph 'Impulses'. The decrease of sound energy is comparable for most frequencies except for higher frequencies where the sound absorption of air plays an important role.

The reverberation times in the box below show that lower frequencies have a longer reverberation time. The high frequent sound is mostly absorbed by air. In common, high frequent sound can be absorbed more easily because of its short wavelength.

Fig,46. *Reverberation times*

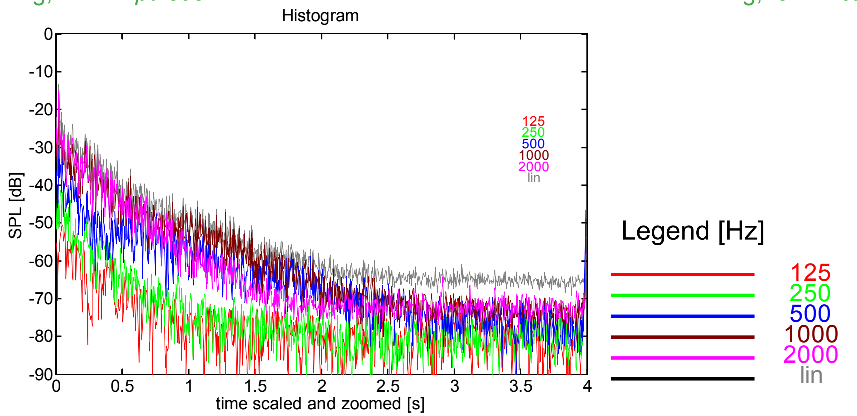
Reverberation times

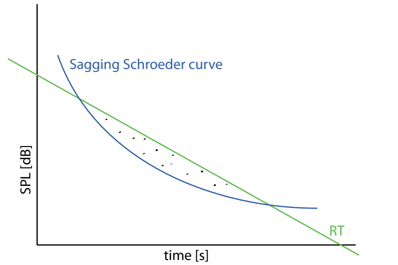
	125	250	500	1000	2000	lin
L_p_0	-29.64	-22.35	-14.43	-6.61	-2.45	-0.23
EDT	13.21	7.80	6.13	3.33	2.12	2.52
T_5_15	13.23	8.73	6.63	3.11	2.18	2.84
T_5_25	10.84	9.15	6.60	3.45	2.15	3.82
T_5_35	10.78	9.07	6.86	4.10	2.33	5.50
T_5_45	10.78	9.07	6.89	6.84	4.25	6.37
T_tijd	8.14	6.38	4.58	3.58	2.01	2.33
times between	0.02	0.20 scaled seconds				
r_EDT	1.00	0.99	0.99	1.00	1.00	1.00
r_5_15	1.00	1.00	1.00	1.00	1.00	1.00
r_5_25	0.96	1.00	1.00	1.00	1.00	0.98
r_5_35	0.96	1.00	1.00	0.98	1.00	0.96
r_5_45	0.96	1.00	1.00	0.88	0.88	0.96
r_time	0.98	0.97	0.97	0.98	0.98	0.99



Fig,47. *Impulses*

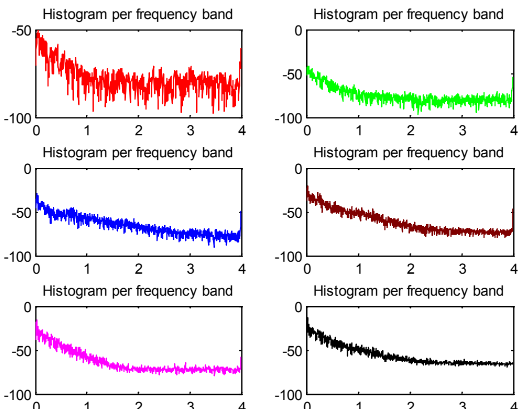
Fig,48. *Picture of measurement setup*



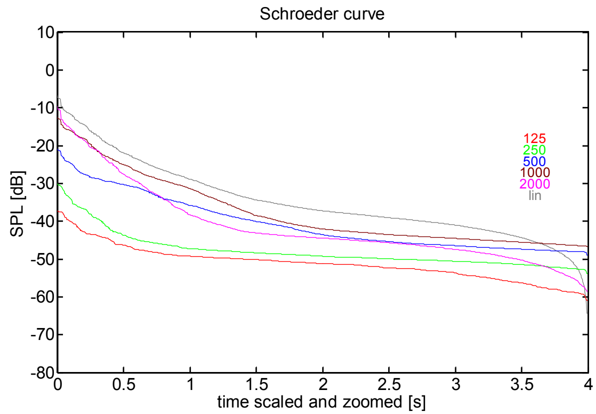


Fig,50. *Sagging cuve: uncorrelated RT.*

Fig,49. *Histogram*



Fig,51. *Histogram: single-curves only*



Fig,52. *Schroeder curve*

10.8. Profile AA with four walls and floor

Data used of measurement 3.

The start level (SPL) at t=0 is lower than of profile A. The decrease of sound pressure of profile AA is bigger than for profile A, this is caused by the sound absorbing layer of fleece. The decreasing, sagging Schroeder curves (for all frequencies) clearly show this effect. High frequencies are still absorbed by air, so the curves are steeper than low frequency Schroeder curves; at least at the start.

The sagging Schroeder curve is typical for measurements in sports halls. Its shape can be explained by early and late reflections. The early reflections, the ones from the roof (mostly vertical) are displayed in the first second of the curve. The late reflections, the ones from the walls (mostly horizontal) are displayed after about 1 second. Since the roof absorbs most of the sound, the curve decreases strongly. The walls reflect most of the sound, so the curve of the late reflections has a smaller slope than the first part. This results in a sagging curve.

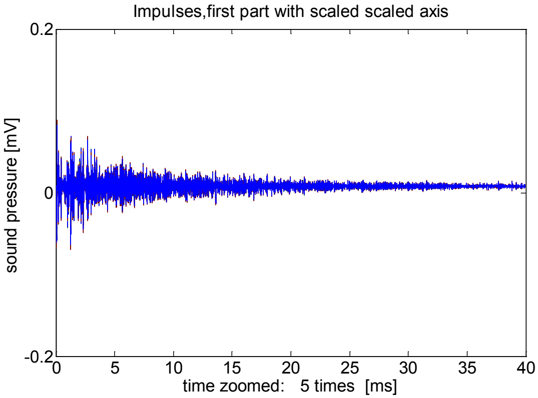
The steep part of the curves result in somewhat uncorrelated (see figure 50) reverberation times. See below: r_ values smaller than 0.95 have to be assessed as unreliable. The EDT values give the best results. Low frequent sound has a longer reverberation time than high frequent sound. The value for 500 Hz looks pretty long, but the r_ value explains why. Remarkable is the value for T5_15 of 500 Hz. It is a high value but its r_value is 0,99.

The question raises whether the EDT correlates more strongly to the subjective experience than T_{30}. In case of music or speech it is. In a sports hall, it depends on the time between the impulses.

Reverberation times

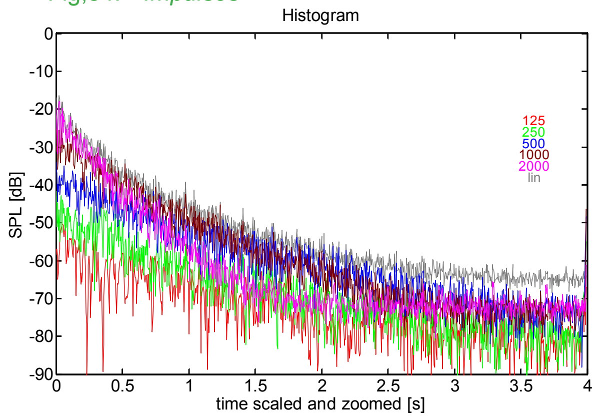
	125	250	500	1000	2000	lin
L_p_0	-37.21	-30.04	-20.91	-12.40	-9.62	-6.83
EDT	3.67	2.06	4.00	2.30	1.60	1.90
T_5_15	18.03	2.82	5.77	2.91	2.01	2.32
T_5_25	18.14	21.46	7.70	4.13	2.27	3.63
T_5_35	18.14	21.46	11.13	9.15	4.09	7.13
T_5_45	18.14	21.46	11.13	9.15	7.94	7.85
T_tijd	1.95	1.67	1.95	2.65	1.72	1.99
times between	0.02	0.20 scaled seconds				
r_EDT	0.95	0.97	0.88	0.98	0.94	0.96
r_5_15	0.84	0.98	0.99	0.98	0.99	1.00
r_5_25	0.94	0.76	0.97	0.98	0.99	0.97
r_5_35	0.94	0.76	0.90	0.85	0.89	0.90
r_5_45	0.94	0.76	0.90	0.85	0.82	0.92
r_time	0.99	0.96	0.97	0.96	0.96	0.96

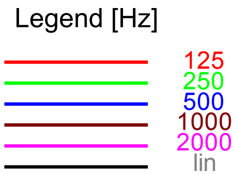
Fig,53. *Reverberation times*



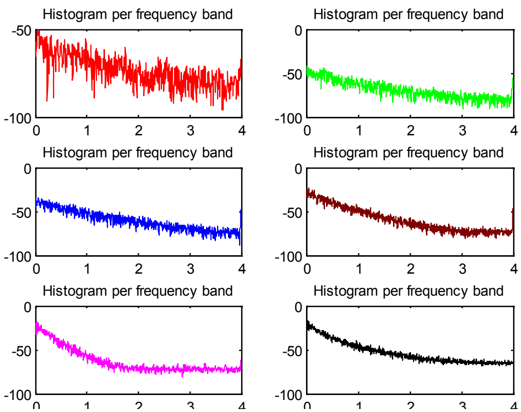
Fig,54. *Impulses*

Fig,55. *Picture of measurement setup*

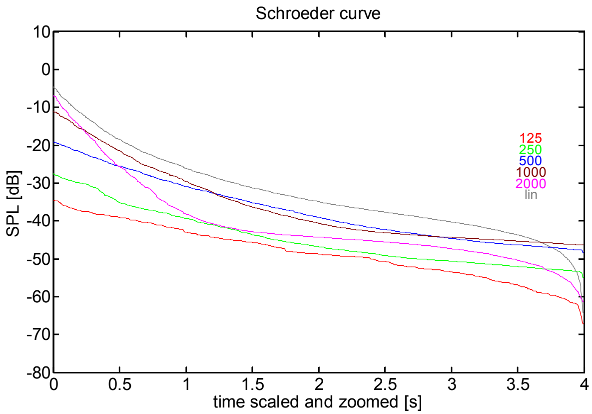




Fig,56. *Histogram*



Fig,57. *Histogram: single-curves only*



Fig,58. *Schroeder curve*

10.9. Profile B with four walls and floor

Data used of measurement 1.

Profile B causes a more diffuse sound field because of its shape. The 'Impulses' graph shows less clear peaks of reflections than for profile A.

The reverberation times below show a decrease of reverberation time for higher frequencies.
The T_5_15 for 250 Hz is 6.79 seconds. Where it is 8.73 seconds for profile A.

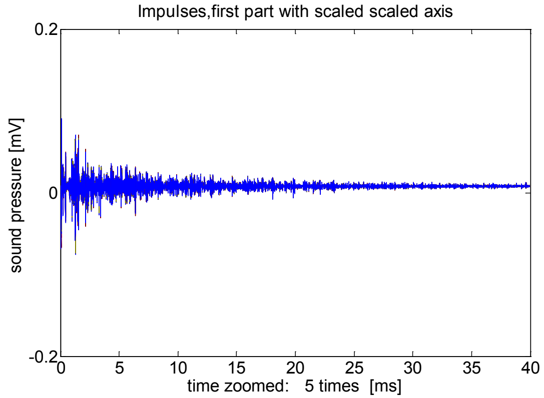
The sound pressure level of 125 Hz sound at t=0 is -35 dB.
The sound pressure level of 2000 Hz sound at t=0 is -7 dB.

For profile A, these values are -30 and -4 dB.

Reverberation times

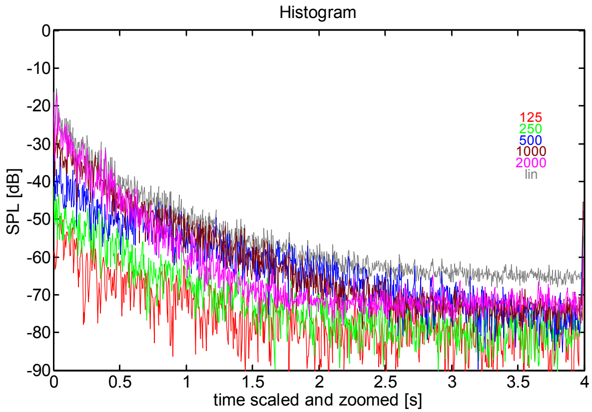
	125	250	500	1000	2000	lin
L_p_0	-34.58	-27.46	-18.98	-11.01	-6.73	-4.60
EDT	8.13	4.33	4.92	2.81	1.50	2.00
T_5_15	10.04	6.79	5.95	3.05	1.63	2.49
T_5_25	10.58	10.60	7.47	3.80	1.83	3.71
T_5_35	10.10	11.37	9.27	7.91	2.36	6.13
T_5_45	10.10	11.37	9.27	8.29	7.42	7.15
T_tijd	4.03	5.21	4.41	2.52	1.48	1.82
times between		0.02	0.20 scaled seconds			
r_EDT	0.99	0.98	1.00	1.00	0.99	0.99
r_5_15	0.98	0.99	1.00	1.00	1.00	1.00
r_5_25	0.99	0.95	0.99	0.99	0.99	0.97
r_5_35	0.98	0.94	0.96	0.87	0.97	0.94
r_5_45	0.98	0.94	0.96	0.87	0.77	0.95
r_time	0.96	0.98	0.99	0.99	0.99	1.00

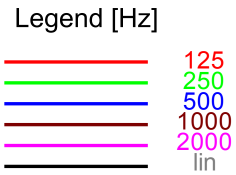
Fig,59. *Reverberation times*



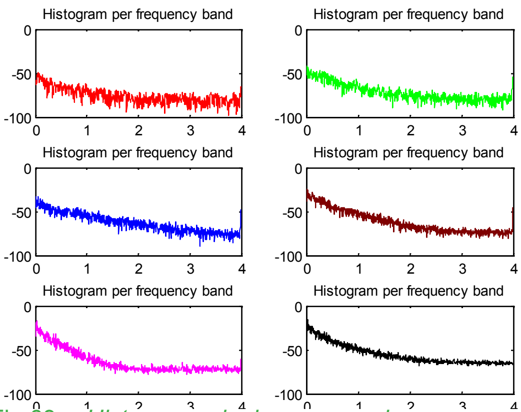
Fig,60. *Impulses*

Fig,61. *Picture of measurement setup*

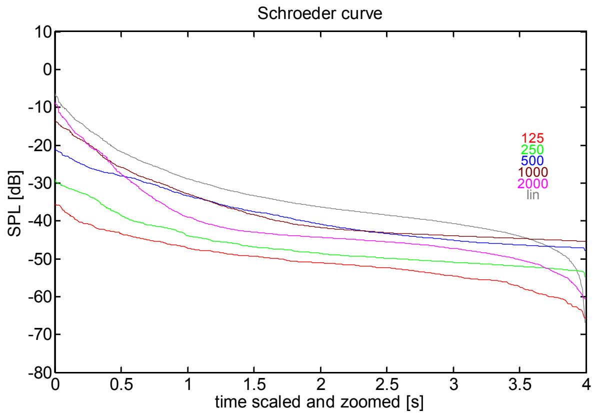




Fig,62. *Histogram*



Fig,63. *Histogram: single-curves only*



Fig,64. *Schroeder curve*

10.10. Profile BB with four walls and floor

Data used of measurement 3.

Profile BB is comparable to profile B. Though, the sound absorbing material causes a faster decrease of the sound pressure level. This faster decrease is visible in the Schroeder curves from t=0 till t=0.5. After that, the line becomes more flat. The sagging curve is also seen for profile AA. This effect is noticed for frequencies up to 1000 Hz, higher frequencies are more easily absorbed by air. In this way, these Shroeder curves are less influenced by the horizontal reflections in the hall.

The reverberation times below show a decrease of reverberation time for higher frequencies. An exception to this is the 250 Hz frequency band, which has a shorter reverberation time than the 500 Hz frequency band.

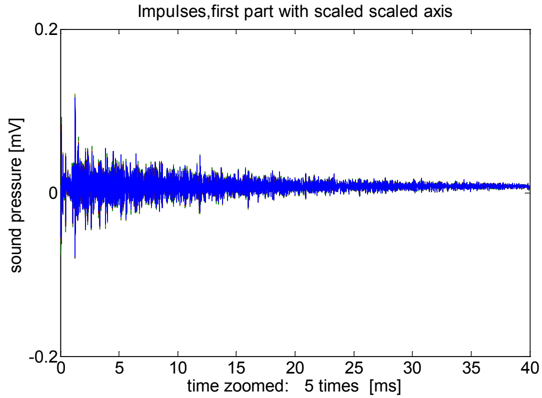
Reverberation times

	125	250	500	1000	2000	lin
L_p_0	-35.68	-29.17	-20.86	-13.33	-8.69	-6.47
EDT	5.32	3.34	5.31	2.41	1.39	1.77
T_5_15	9.94	5.02	6.01	2.72	1.98	2.33
T_5_25	13.43	15.20	8.90	4.14	1.96	3.65
T_5_35	12.80	15.20	10.57	10.22	3.29	7.27
T_5_45	12.80	15.20	10.57	10.22	8.10	7.89
T_tijd	2.17	4.24	3.11	2.35	1.42	1.69

times between 0.02 0.20 scaled seconds

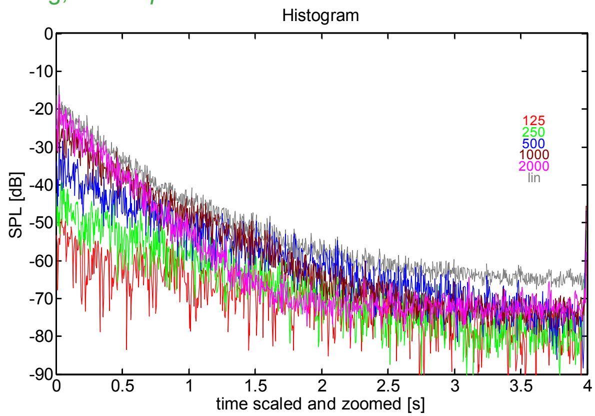
r_EDT	0.93	0.99	0.97	0.99	0.97	0.98
r_5_15	0.97	0.95	1.00	0.98	0.98	1.00
r_5_25	0.97	0.88	0.97	0.98	1.00	0.97
r_5_35	0.96	0.88	0.94	0.82	0.92	0.91
r_5_45	0.96	0.88	0.94	0.82	0.79	0.93
r_time	0.99	0.99	0.99	0.99	0.98	0.99

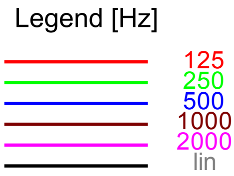
Fig,65. *Reverberation times*



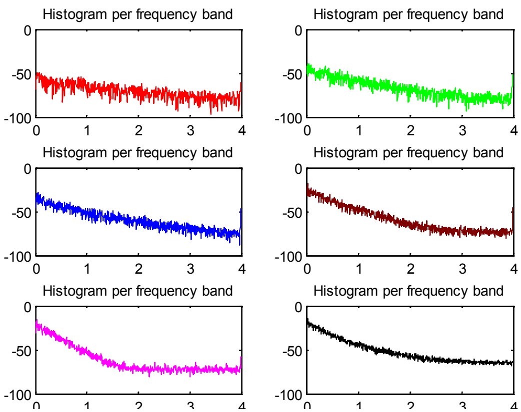
Fig,66. *Impulses*

Fig,67. *Picture of measurement setup*

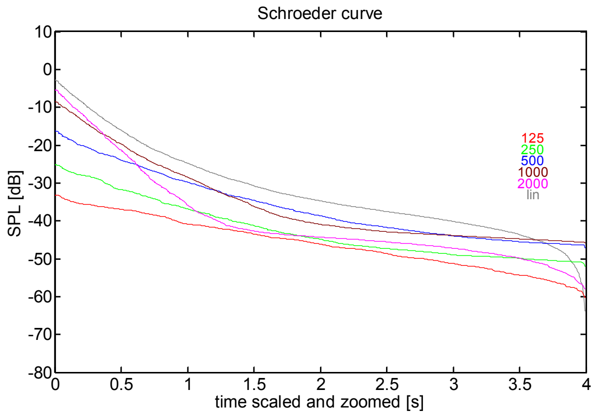




Fig,68. *Histogram*



Fig,69. *Histogram: single-curves only*



Fig,70. *Schroeder curve*

10.11. Profile C with four walls and floor

Data used of measurement 3.

The ‘Impulses’ graph of profile C shows more clear peaks than the graph for profile B. This is probably caused by the shape of the profile. The tapered slits project the sound back into the room.

The reverberation times below show a decrease of reverberation time for higher frequencies. The value for 125 Hz is abnormally long. This is caused by the small total decrease of this curve.

The sound pressure level of 125 Hz sound at t=0 and 2000 Hz sound at t=0 per profile is given below:

	125 Hz	2000Hz
A	-30	-4
AA	-37	-10
B	-35	-7
BB	-36	-9
C	-34	-5

Profile A, the roof with the most absorption works best. It gives the lowest sound pressure level. Profile BB, B and C follow with a small difference.

Reverberation times

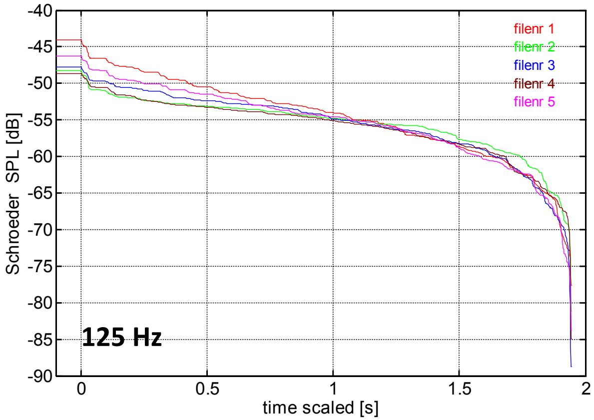
	125	250	500	1000	2000	lin
L_p_0	-33.09	-25.04	-16.16	-8.34	-5.19	-2.66
EDT	9.12	5.03	4.36	2.53	1.86	2.12
T_5_15	10.59	5.93	5.05	2.86	1.80	2.42
T_5_25	10.78	9.73	6.26	3.38	1.84	3.20
T_5_35	10.67	10.89	8.95	5.01	2.10	5.06
T_5_45	10.67	10.89	8.95	7.88	6.53	6.68
T_tijd	4.62	3.96	3.03	2.50	1.83	2.05

times between 0.02 0.20 scaled seconds

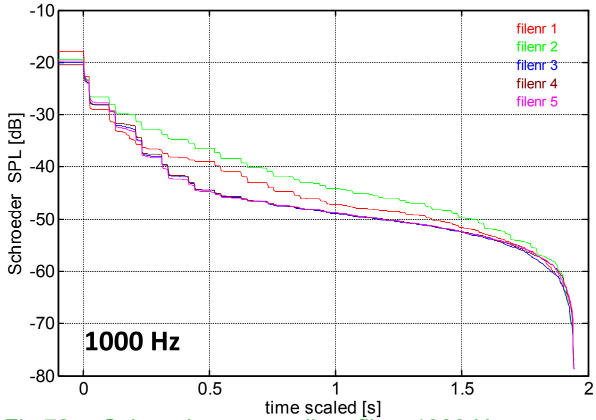
r_EDT	0.99	0.99	0.99	1.00	1.00	1.00
r_5_15	0.99	1.00	1.00	1.00	1.00	1.00
r_5_25	1.00	0.94	0.99	0.99	1.00	0.98
r_5_35	0.99	0.92	0.94	0.94	0.99	0.94
r_5_45	0.99	0.92	0.94	0.84	0.77	0.94
r_time	0.97	0.97	0.99	1.00	1.00	1.00

Fig,71. *Reverberation times*

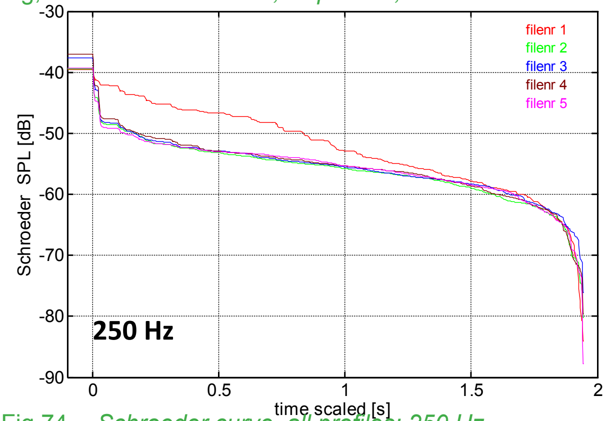
filenr 1 = Profile A   filenr 2 = Profile AA   filenr 3 = Profile B   filenr 4 = Profile BB   filenr 5 = Profile C



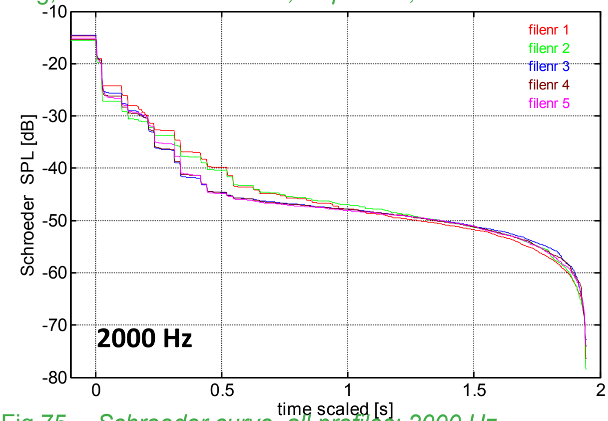
Fig,72. *Schroeder curve, all profiles; 125 Hz*



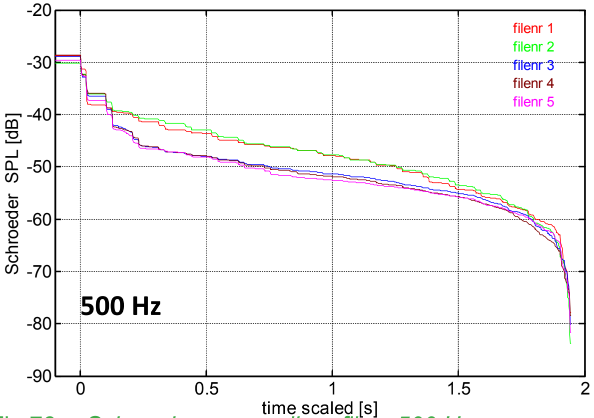
Fig,73. *Schroeder curve, all profiles; 1000 Hz*



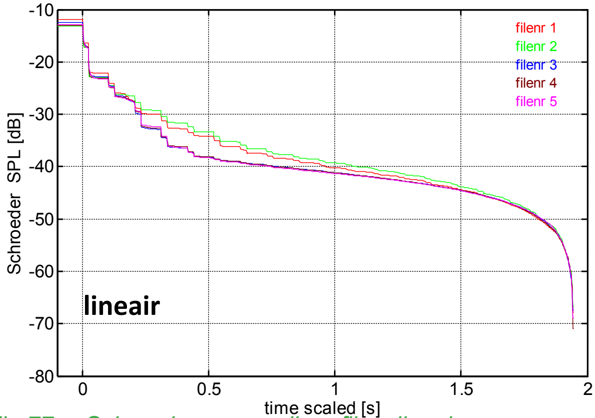
Fig,74. *Schroeder curve, all profiles; 250 Hz*



Fig,75. *Schroeder curve, all profiles; 2000 Hz*

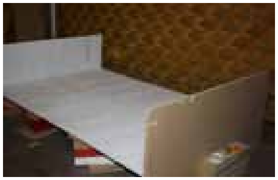


Fig,76. *Schroeder curve, all profiles; 500 Hz*



Fig,77. *Schroeder curve, all profiles; lineair*

10.12. All profiles compared: two walls



When all five tested profiles are shown in one graph, it is easy to compare them. On the left page, the Schroeder curves are shown per frequency band. There are a few things that are striking:

1. The start-level of the panels with absorbing material or profiled facings are much lower for low frequent sound than for high frequent sound. The effect of sound absorption is strongly found for low frequent sound. This is also visible in the sagging shape: especially for 250 and 500 Hz sound, the curves are sagging. This should indicate to strong sound absorption of these frequencies by the roof structure.

2. The order of the lines is the same for most of the graphs. Which means that profile A absorbs least of the sound and profile BB and profile AA absorbs most of the sound. This is caused by the amount of sound absorbing material added to the profile and the shape of the profile.
Besides that, the order shows that profile C absorbs less sound than profile B.

3. The effect of scattering is almost as good as the effect of adding a fleece layer to a flat profile. Sometimes it is even better. When we compare profile AA with profile B, we see a crossing of the lines in the graph of 125 Hz at t = 1.2 seconds. This means that from that moment profile B, without any sound absorbing materials, absorbs more of the sound than profile AA which is completely covered by sound absorbing material.

4. The differences in sound absorbing behaviour of the profiles are bigger for low frequencies than for high frequencies.

5. All curves are curved, just the 125 Hz frequency graph shows a kind of straight lines. Curved lines cause problems when calculating reverberation times. The tangent is different for every interval. This is exactly the problem of using reverberation times in sports halls. For more information, see chapter 3.5.

Profile A

	125	250	500	1000	2000	lin
EDT	13.21	7.80	6.13	3.33	2.12	2.52
T_5_15	13.23	8.73	6.63	3.11	2.18	2.84
T_5_25	10.84	9.15	6.60	3.45	2.15	3.82
r_EDT	1.00	0.99	0.99	1.00	1.00	1.00
r_5_15	1.00	1.00	1.00	1.00	1.00	1.00
r_5_25	0.96	1.00	1.00	1.00	1.00	0.98
Reverberation times corrected						
EDT	14.23	9.16	10.18	7.99	7.33	2.52
T_5_15	14.26	10.46	11.62	6.85	8.08	2.84
T_5_25	11.51	11.08	11.54	8.69	7.70	3.82

Fig,78. *Reverberation times, complete box; profile A*

Profile AA

	125	250	500	1000	2000	lin
EDT	3.67	2.06	4.00	2.30	1.60	1.90
T_5_15	18.03	2.82	5.77	2.91	2.01	2.32
T_5_25	18.14	21.46	7.70	4.13	2.27	3.63
r_EDT	0.95	0.97	0.88	0.98	0.94	0.96
r_5_15	0.84	0.98	0.99	0.98	0.99	1.00
r_5_25	0.94	0.76	0.97	0.98	0.99	0.97
Reverberation times corrected						
EDT	3.74	2.14	5.41	3.84	3.46	1.90
T_5_15	19.99	2.98	9.24	5.95	6.22	2.32
T_5_25	20.13	36.25	15.38	14.95	9.51	3.63

Fig,79. *Reverberation times, complete box; profile AA*

Fig,80. *Reverberation times, complete box; profile B*

Profile B

	125	250	500	1000	2000	lin
EDT	8.13	4.33	4.92	2.81	1.50	2.00
T_5_15	10.04	6.79	5.95	3.05	1.63	2.49
T_5_25	10.58	10.60	7.47	3.80	1.83	3.71
r_EDT	0.99	0.98	1.00	1.00	0.99	0.99
r_5_15	0.98	0.99	1.00	1.00	1.00	1.00
r_5_25	0.99	0.95	0.99	0.99	0.99	0.97
Reverberation times corrected						
EDT	8.51	4.72	7.23	5.55	3.01	2.00
T_5_15	10.62	7.80	9.71	6.52	3.59	2.49
T_5_25	11.23	13.27	14.49	11.32	4.78	3.71

Fig,81. *Reverberation times, complete box; profile BB*

Profile BB

	125	250	500	1000	2000	lin
EDT	5.32	3.34	5.31	2.41	1.39	1.77
T_5_15	9.94	5.02	6.01	2.72	1.98	2.33
T_5_25	13.43	15.20	8.90	4.14	1.96	3.65
r_EDT	0.93	0.99	0.97	0.99	0.97	0.98
r_5_15	0.97	0.95	1.00	0.98	0.98	1.00
r_5_25	0.97	0.88	0.97	0.98	1.00	0.97
Reverberation times corrected						
EDT	5.48	3.57	8.11	4.18	2.60	1.77
T_5_15	10.51	5.55	9.84	5.19	5.90	2.33
T_5_25	14.49	21.38	21.06	14.99	5.71	3.65

10.13. All profiles compared: complete box

Because the complete box configuration comes closest to a real life situation and disturbing background noise is minimal, the comparison of all profiles gives very interesting results. For all results of the measurements: see Attachment A.The complete box configuration gives the opportunity to calculate reverberation times as in 'real life'. The results are shown on the left page. A correction for sound absorption of air is added.

1. The reverberation times become shorter when the frequency increases. This behaves as expected. Though, the 250 Hz frequency band gives a longer RT (T_5_25) for all profiles but most of these results are not reliable because of too low r_ values and strongly sagging curves.

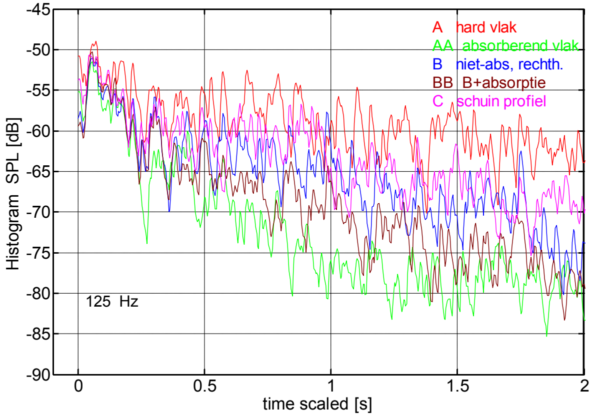
2. Profile B and C give shorter reverberation times (T10 and EDT) than profile A just because of their shape. This is of course a very interesting result.

3. The expected shorter reverberation time (T20, as used in 'real-life' measurements) for profile B, BB and C is not found. For sports halls constructed with such shaped roofs, nothing 'strange' will be found from reverberation times. A look at the Schroeder curves or histograms will give more information. These are shown on the next pages. They show a lower sound pressure level for profiles BB, B and C than for A at all frequencies.

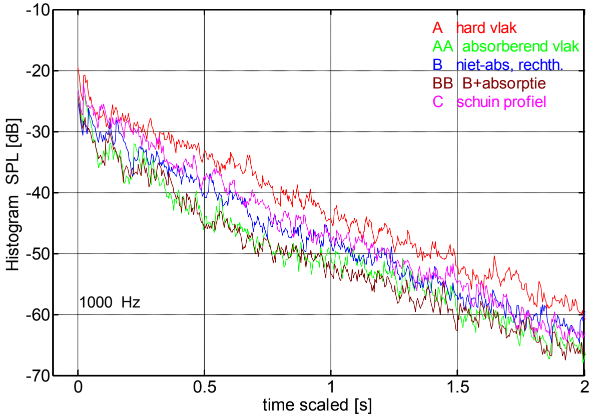
Profile C

	125	250	500	1000	2000	lin
EDT	9.12	5.03	4.36	2.53	1.86	2.12
T_5_15	10.59	5.93	5.05	2.86	1.80	2.42
T_5_25	10.78	9.73	6.26	3.38	1.84	3.20
r_EDT	0.99	0.99	0.99	1.00	1.00	1.00
r_5_15	0.99	1.00	1.00	1.00	1.00	1.00
r_5_25	1.00	0.94	0.99	0.99	1.00	0.98
	Reverberation times corrected					
EDT	9.59	5.56	6.08	4.53	4.98	2.12
T_5_15	11.24	6.68	7.52	5.74	4.57	2.42
T_5_25	11.45	11.94	10.54	8.30	4.78	3.20

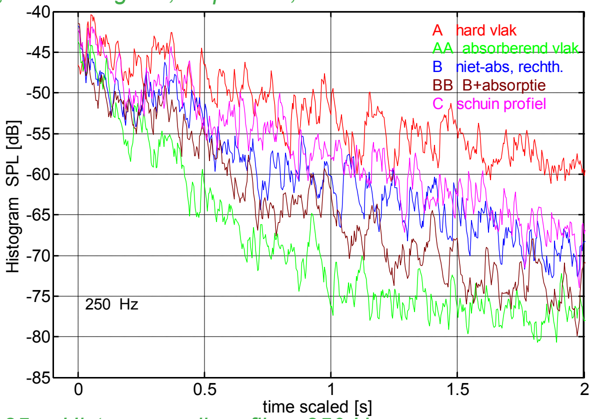
Fig,82. *Reverberation times, complete box; profile C*



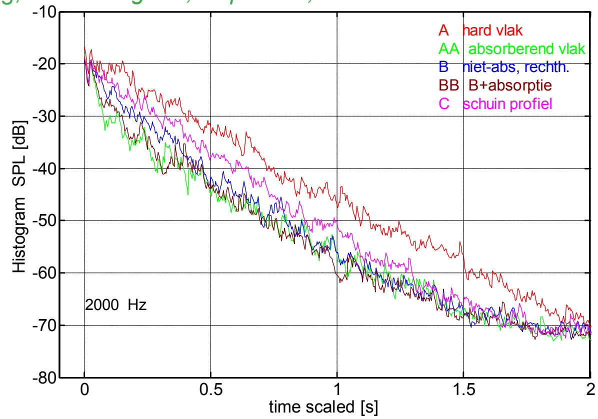
Fig,83. *Histogram, all profiles; 125 Hz*



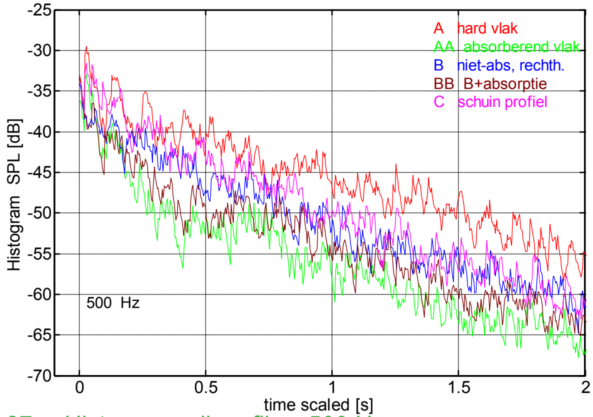
Fig,84. *Histogram, all profiles; 1000 Hz*



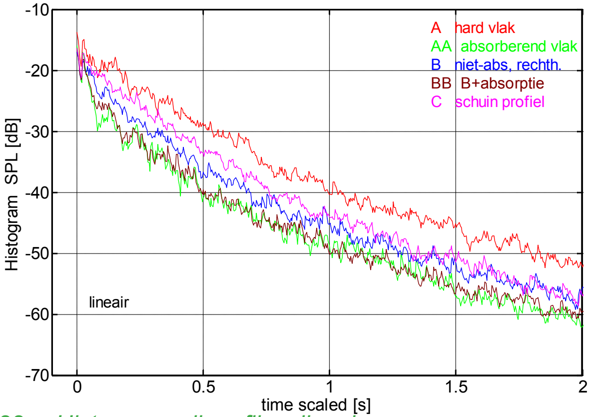
Fig,85. *Histogram, all profiles; 250 Hz*



Fig,86. *Histogram, all profiles; 2000 Hz*

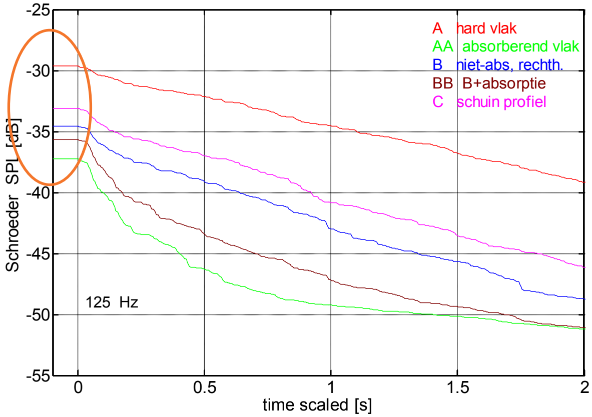


Fig,87. *Histogram, all profiles; 500 Hz*

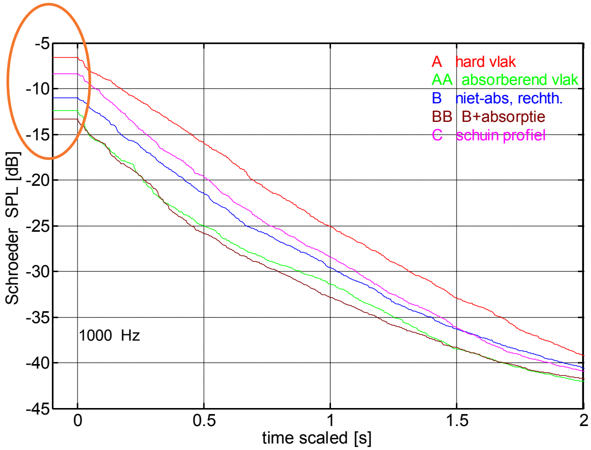


Fig,88. *Histogram, all profiles; lineair*

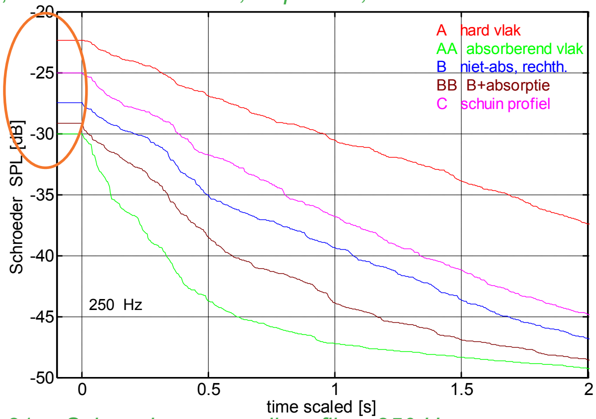
Profile A Profile AA Profile B Profile BB Profile C



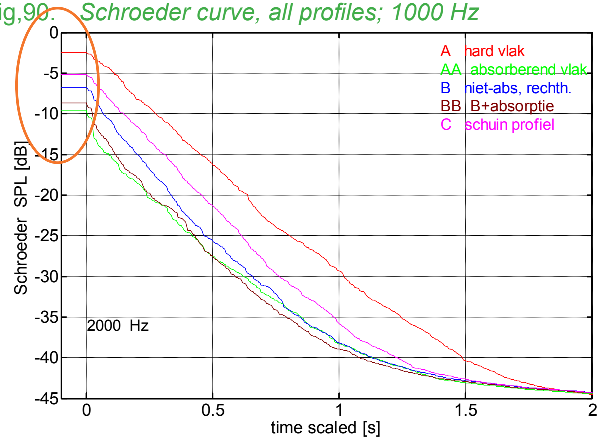
Fig,89. *Schroeder curve, all profiles; 125 Hz*



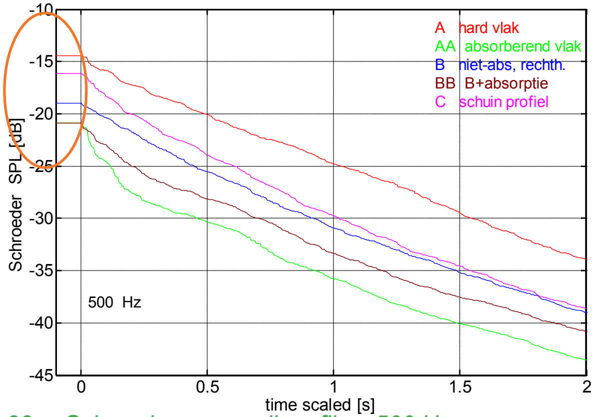
Fig,90. *Schroeder curve, all profiles; 1000 Hz*



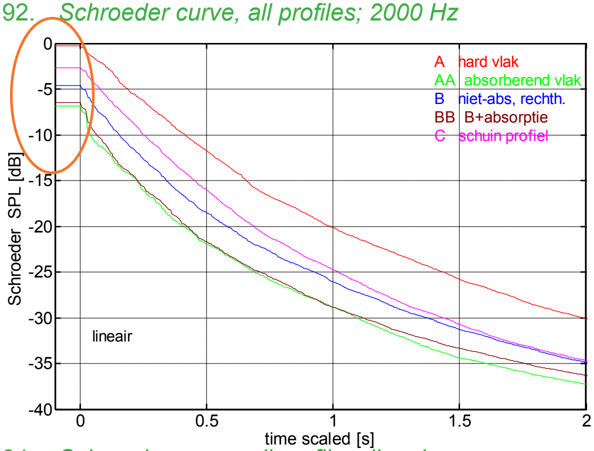
Fig,91. *Schroeder curve, all profiles; 250 Hz*



Fig,92. *Schroeder curve, all profiles; 2000 Hz*



Fig,93. *Schroeder curve, all profiles; 500 Hz*



Fig,94. *Schroeder curve, all profiles; lineair*

10.14. Conclusions

The main questions to answer by the scale model measurements are: ‘Does the sound absorbing behaviour of a roof structure depend on its shape?’ and ‘Can the found high sound absorption values of a perforated panel roof structure be explained by its shape?’. The first question can be answered with ‘yes’. The second question is a more complicated to answer. The answer to this question will be given after the conclusions of the measurements.

The hanging Schroeder curves seen for measurements in the complete box explain the problem of the ‘good’ and ‘bad’ sports halls very clearly. Because of this hanging shape, the reverberation times EDT, T15 and T25 are very different. The reverberation time can be decreased by adding more sound absorbing materials to the walls. Then, the horizontal sound field is less predominant in the shape of the Schroeder curve and so on the reverberation time calculation.

Low frequent sound can be absorbed by the shape of a roof structure. The slits in profile B and C cause strong scattering and phase shifting effects. Therefore, the sound pressure of the reflection decreases. This ‘absorption without any sound absorbing materials’ can absorb more low frequent sound than a flat sound absorbing panel after about 1.2 seconds (which means that the reflection has travelled about 11 times the length of a sports hall).

The interference frequencies strongly depend on the Q value of the surface. This ‘ground effect’ of profile B, BB and C is comparable to the ground effect of profile AA. This means that the effect of scattering is comparable to an additional layer of sound absorbing material.

Profile BB has half of the amount of sound absorbing material compared to profile AA. Profile BB might preform better than profile AA with the same amount of sound absorbing material.

A large angle of incidence (striking sound) on a hard surface can cause a better sound absorption of low frequent sound. This information can be useful during the design of a sports hall. When the produced sound in the hall hits a wall and the reflection is projected to the roof structure, the angle of incidence of this projection can make a big difference in sound absorption of this reflection depending on the properties of the roof structure.

The profile has large influence on the time it takes before reflections are gone. Diffusing the sound gives better results than adding a sound absorbing material.

Sports halls constructed with perforated steel panels seem to behave differently than sports halls constructed with stones. This can not be seen just from reverberation times (T20) only. (For T10 and EDT, it can.) This conclusion can be found by comparing the histogram and Schroeder curve of both sports halls. These graphs give more important information than just reverberation times. When trying to improve the acoustics of a 'bad' sports hall, this information should be taken into account.

A scale model is a good tool to investigate the effects of different roof structures on the sound field. Though, a note has to be made. The sound source, the pulse, cannot easily produce low frequent sound. This is a disadvantage because of the special interest in this type of sound. Another sound source would not be so small as a pulse, therefore it is still a very good option for this research.

Large low frequent absorption of perforated panel caused by shape?

The high sound absorption values of a perforated panel as a roof structure can be partly explained by its shape. Like seen in the results of this scale model test, the diffusing properties of the corrugated roof can cause the same effect as sound absorption. This effect has large influence and therefore can be the main reason for the extraordinary good absorption of low frequent sound by perforated steel panels.

11. Scale model measurements 2

This chapter continues on chapters 9 and 10 where the scale model tests in a 1:20 model of a sports hall are discussed. This part of the report focuses on theory behind the reflection coefficients of the in previous chapters used profiles.

11.1. Theory behind reflection coefficient measurements

The reflection coefficient is the part of the sound that is reflected by the surface. A very basic formula for this is: a+r+t=1. Where the absorbed part (a), the reflected part (r) and the transmitted part (t) of the sound together are one.

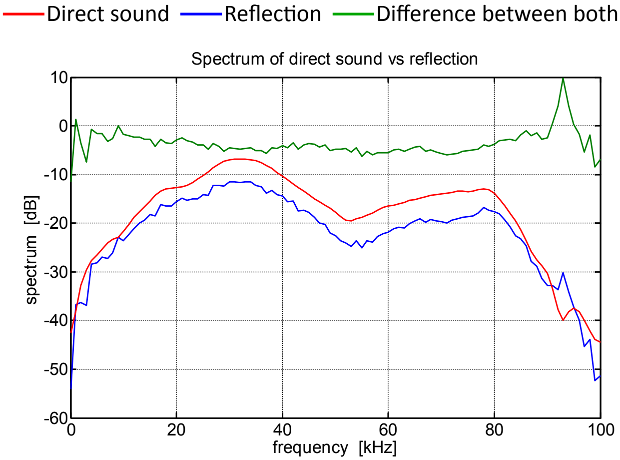
A more precise way to calculate the reflection coefficient of a surface is to use the formula of figure 2. It is based on the measured sound pressure of the pulses. When we know the Rp value of the panel, the absorption coefficient can be calculated by: α = 1-R.

An example. ΔL is found in figure 1 (green line).

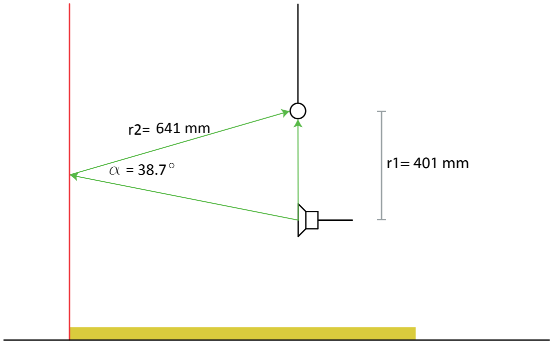
$$R_p = \frac{641^2}{401^2} \cdot 10^{-4.5/10} = 0.9$$

This value comes close to the found reflection coefficient of profile A discussed in next chapter.

The physically correct way to calculate the reflection coefficient is based on the impedance of the material, this goes too far to discuss here.



Fig,1. *Profile A, reflection coefficient measurement.*



Fig,3. *Measurement setup.*

$$\Delta L = L_{p1} - L_{p2} = 10\log(\frac{r_1^2}{r_2^2} \cdot R_p)$$

With:

ΔL= The difference in sound pressure of two pulses (the green line in the graph)

L_{p1} = The sound pressure of pulse 1 (direct sound, also the red line in the graph)

L_{p2} = The sound pressure of pulse 2 (direct sound, also the blue line in the graph)

r_1 = Travelling distance of sound from source to receiver

r_2 = Travelling distance of sound from source to receiver via surface

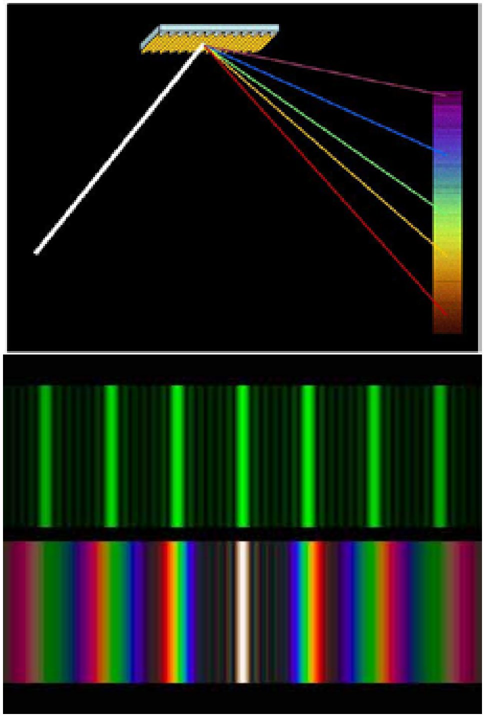
R_p = The sound pressure of the reflected sound

$$R_p = \frac{r_1^2}{r_2^2} \cdot 10^{\Delta L/10}$$

Fig,2. *Formula to calculate the sound pressure of the refleciton.*

11.2. Diffraction grating

'In optics, a diffraction grating is an optical component with a periodic structure, which splits and diffracts light into several beams travelling in different directions. The directions of these beams depend on the spacing of the grating and the wavelength of the light so that the grating acts as the dispersive element.' (Wikipedia, 2012)



Fig,4. *Diffraction grating in optics. (http://oco.jpl.nasa.gov) (http://h2physics.org/?cat=49)*

For sound, this phenomenon works the same. Sound waves can be splitted and defracted by a 'diffraction grating'. For this research, the slits in profile B, BB and C can be seen as a diffraction grating. The relationship between the grating spacing and the angles of the incident and diffracted beams of light is known as the grating equation. The formula rewritten for sound is:

$$f = \frac{nc}{a \sin \alpha}$$

With:

f = frequency of interference maximum

n = an integer representing the propagation-mode of interest

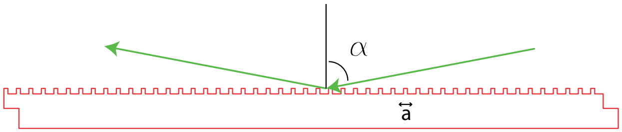
c = the velocity of sound in air

a = distance from the center of one slit to the center of the adjacent slit

α = angle of incidence

See also figure 5 on next page.

The frequency at which these peaks occur depends on the distance from the source on which the panel is placed. This is probably slightly different per measurement.



Fig,5. *Diffraction grating*

An example:

Profile B:

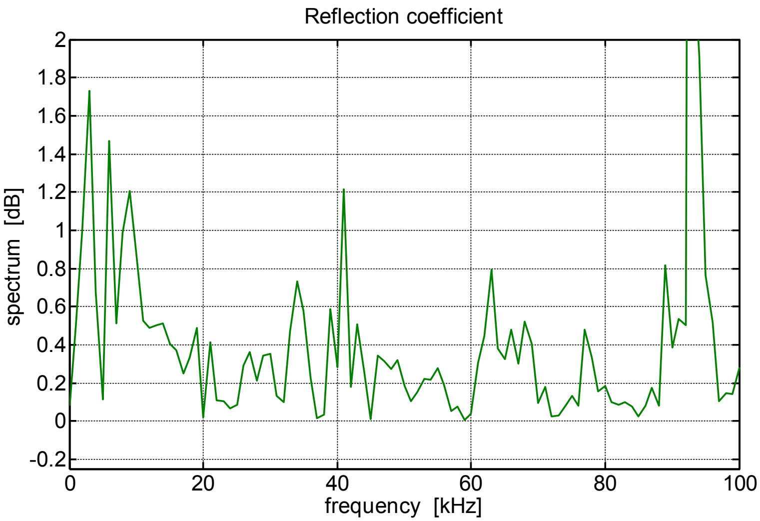
a= 12.5 mm

alpha= 38.7 degrees

Maximal reflection will take place for: f=(n*343)/(0.0125 sin 38.7)= 43.9 kHz, 87.8 kHz, 131.7 kHz ...

This is visible in the graph of the reflection coefficient. More results will be discussed in the next chapter.

The large peak at 90 kHz is probably caused by the flute-tone produced by the microphone.



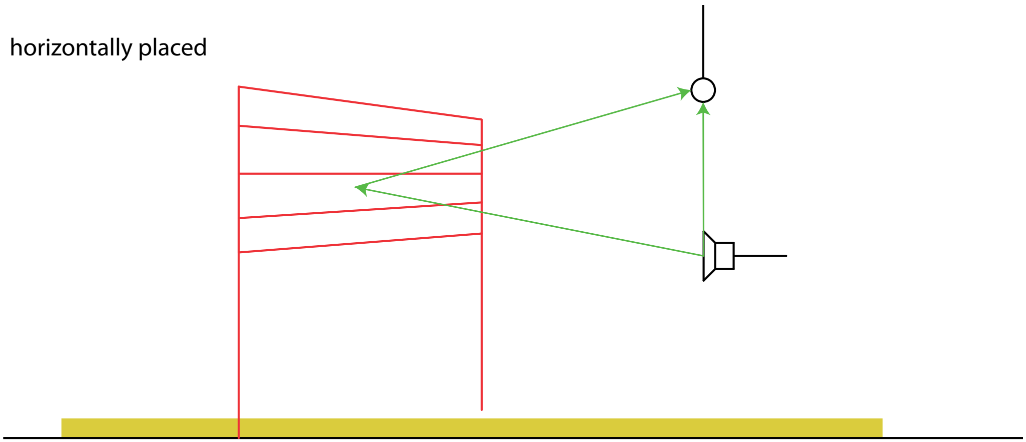
11.3. Measurement method

The reflection coefficient can also be measured. Therefore, the spectrum of the direct sound and the reflection are compared. The higher the sound pressure level of the reflection, the more sound is reflected, the higher the reflection coefficient. For a flat surface, the reflection coefficient cannot be greater than 1 because of the law of conservation of energy.

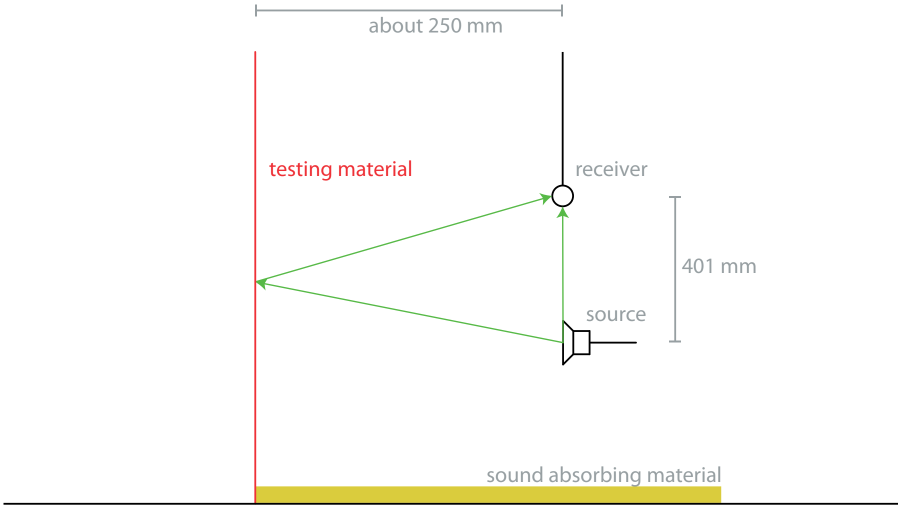
It is important to have just two pulses to calculate with: a direct and a reflection. When unwanted reflections arrive at the same time as the wanted reflection, the measurement is unreliable. Therefore, the ground is covered with a layer of sound absorbing material.

The setup for the measurement is shown in figure 7. The red line represents the tested profiles that are schematized in figure 8.

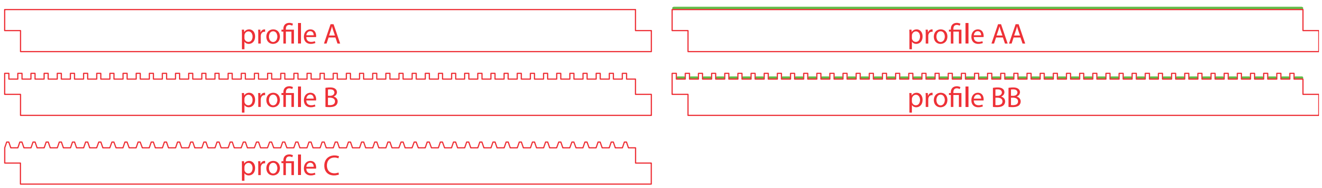
The reflection coefficients of the profiles with slits are measured in two orientations: vertically and horizontally placed. Figure 6 shows the direction of the slits for a horizontally placed profile.



Fig,6. *Orientation of profile*



Fig,7. *Measurement setup for reflection coefficient measurements*

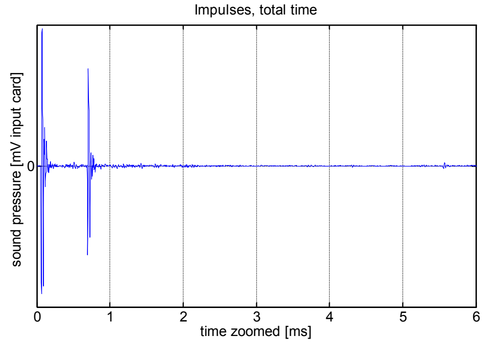


Fig,8. *Tested profiles*

12. Results and discussion

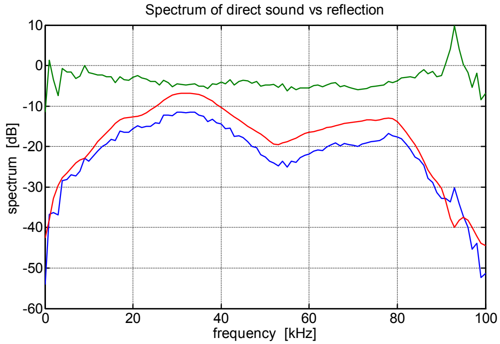
12.1. Profile A

The reflection coefficient of profile A is calculated from the difference between the direct sound and the reflection. The graphs show the two pulses, the spectrum of both and the reflection coefficient per frequency.



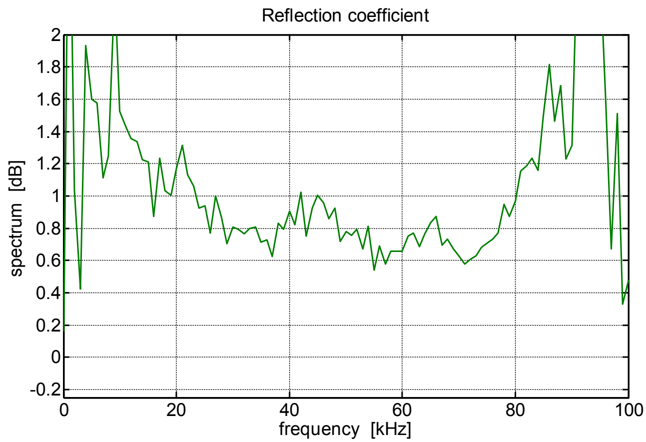
Fig,1. *Impulses profile A*

The reflection coefficient is about 0,8. It varies per frequency band. It is almost 1, which means that it reflects almost all of the sound. The peaks at high (90 kHz) and low (10 kHz) frequencies could be caused by the sensibility of the microphone. It has some problems to receive very high frequencies. Besides that, the peaks could be caused by little irregularities of the painted surface. The microphone could receive more than one reflection when the surface is not exactly flat. This leads to reflection coefficients that are bigger than 1.



Fig,2. *Spectra direct sound and profile A*

On the whole, high frequencies are slightly better absorbed (or transmitted) by the panel than low frequencies. The reflection coefficient is therefore larger for low frequent than for high frequent sound.



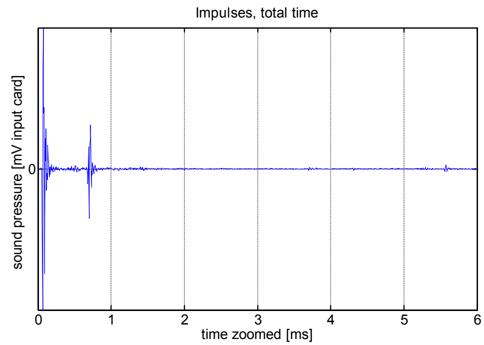
Fig,3. *Reflection coefficient profile A*

12.2. Profile AA

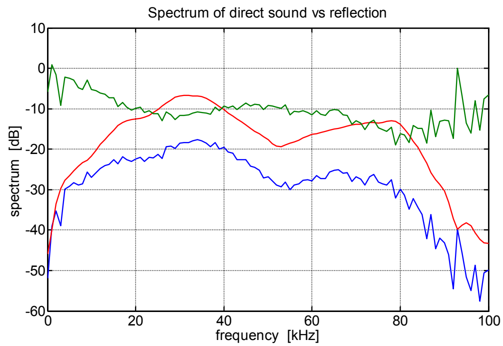
Profile AA consist of a flat surface with a top layer of fleece. Fleece is a sound absorbing material; the graph shows a much smaller reflection than for profile A. Also the spectrum of the reflection shows that a bigger part of the sound is absorbed. The thin layer can easily absorb high frequent sound, so the reflection coefficient of the profile is a little bigger for low frequent than for high frequent sound.

The large fluctuations of the reflection coefficient of low frequent sound could be explained by the size of the tested panel.

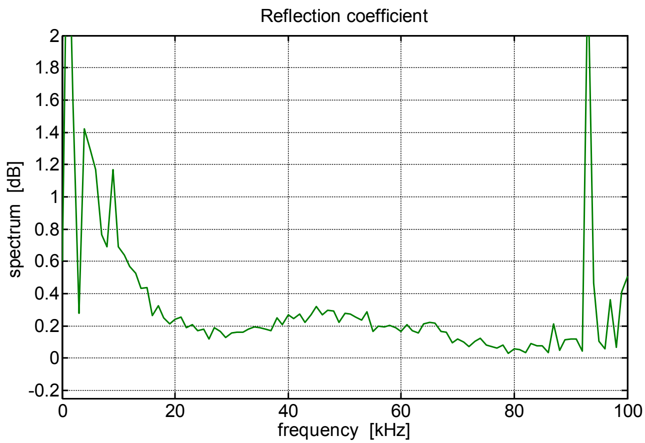
The reflection coefficient of profile AA is about 0,2.



Fig,4. *Impulses profile AA*



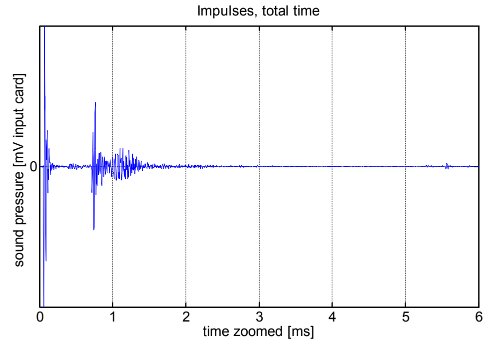
Fig,5. *Spectra direct sound and profile AA*



Fig,6. *Reflection coefficient profile AA*

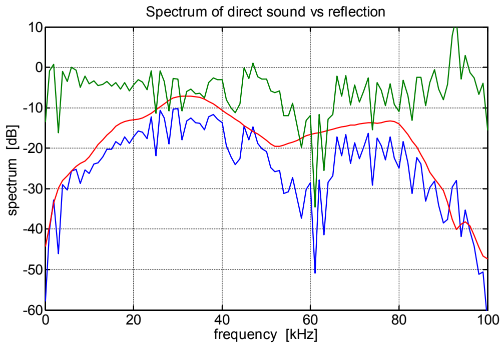
12.3. Profile B, vertically placed

The reflection coefficient of profile B is much harder to find in the graph. Because of the slits in the surface, different reflections reach the microphone. This leads to high reflection coefficients. On the other hand is the scattering of sound an important variable on the reflection coefficient. When most of the sound is reflected to different directions, the reflection coefficient will be low because just a part of the sound reaches the microphone.



Fig,7. *Impulses profile B*

The panel is placed vertically for this measurement. It is not sure whether the reflection took place on top or in between two of the slits. This could have effect on the reflection coefficient.

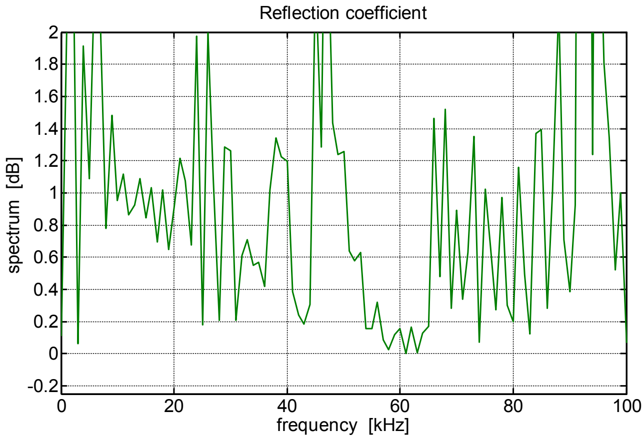


Fig,8. *Spectra direct sound and profile B*

The profiled panels are milled out of foam. The structure of the surface is not very smooth. The irregularities could cause more scattering than there would be on a 1 to 1 scale.

The average reflection coefficient of profile B is about 0,7.

According to the diffraction grating principle, peaks in the reflection coefficient graph should occur about 43,9 kHz and 87,8 kHz. The first one is clearly visible, the second one not.



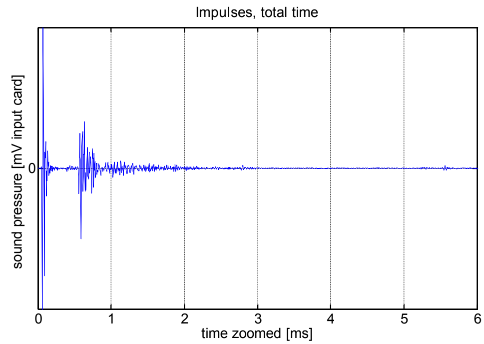
Fig,9. *Reflection coefficient profile B*

Figure 7 shows a clear stoppage of reflections after about 1,5 ms. This is caused by the size of the tested panel. For the horizontal placement, this stoppage is not visible later, because the panel has a rectangular shape.

12.4. Profile B, horizontally placed

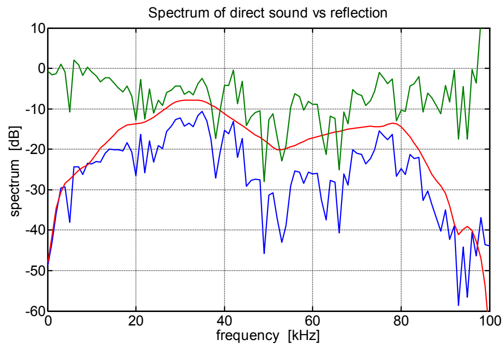
The reflection coefficient of profile B is also measured with the slits in horizontal direction because the direction of the panels might have influence on the absorbing behaviour.

The reflection in the graph 'Impulses, total time' is clearly influenced by the shape of the panel. This can also be seen for the vertical measurement. The direct sound is a clear peak, the reflection is not. The measurement in vertical direction shows a stronger peak. This is probably caused by the orientation of the panel.



Fig,10. *Impulses profile B*

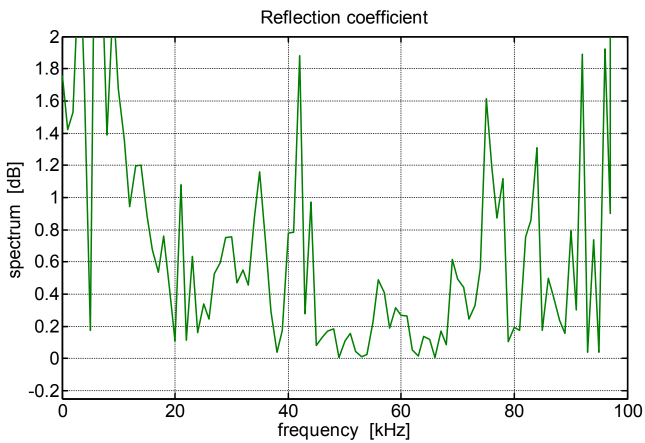
The spectrum of the reflection is, like the previous one, a jerky line. The effect of interference is visible: some frequencies are 'lost' (50 kHz) and some are doubled (42 kHz).



Fig,11. *Spectra direct sound and profile B*

The reflection coefficient of panel B, horizontally placed, is about 0,4.

According to the diffraction grating principle, peaks in the reflection coefficient graph should occur for about 43,9 kHz and 87,8 kHz. Both are visible. The orientation of the panel is important for the visibility of these peaks.



Fig,12. *Reflection coefficient profile B*

12.5. Profile BB, vertically placed

Profile BB has fleece strips in the slits and is for this measurement vertically placed.

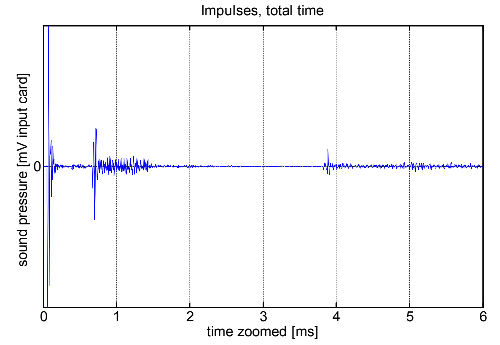
The decrease in sound pressure because of the sound absorbing material is clearly visible in the 'Impulses' graph.

The reflection coefficient is about 0,4.

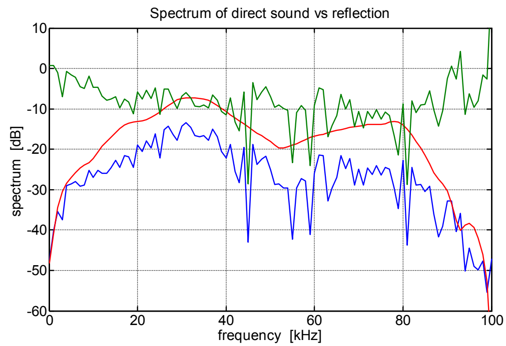
The extra reflection in the 'Impulses' graph is caused by a chair in the anechoic room that was not removed during the measurement. It does not influence the results.

According to the diffraction grating principle, peaks in the reflection coefficient graph should occur for about 43,9 kHz and 87,8 kHz. Both are visible.

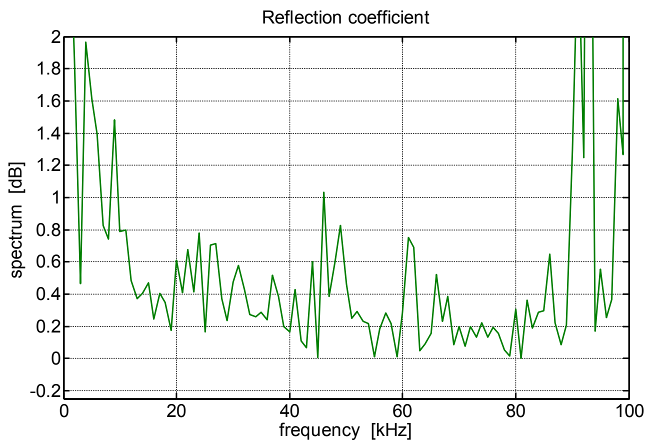
Again, the end of the tested panel is visible by a stoppage of received reflections. See figure 13.



Fig,13. *Impulses profile BB*



Fig,14. *Spectra direct sound and profile BB*



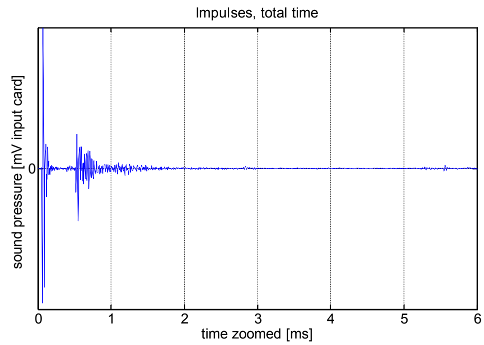
Fig,15. *Reflection coefficient profile BB*

12.6. Profile BB, horizontally placed

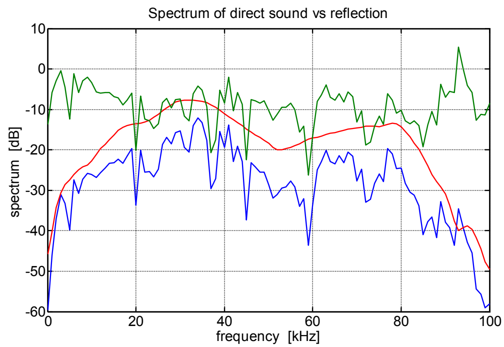
The horizontally placed profile BB shows a less strong reflection than the vertically placed version. The scattering of sound is again visible. The spectrum of the reflections shows some more clear dips than for the vertically placed version. This is probably caused by the orientation of the panel. The influence of the slits on the reflection is bigger when horizontally placed than vertically placed.

The reflection coefficient is about 0,3.

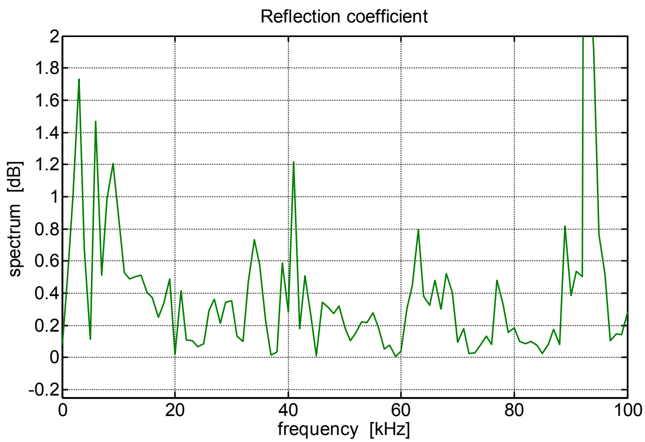
The peaks of the diffraction grating are again visible. Since the sound pressure of the reflected sound is much lower than for profile B because of the sound absorbing material, the peaks are smaller too.



Fig,16. *Impulses profile BB*



Fig,17. *Spectra direct sound and profile BB*



Fig,18. *Reflection coefficient profile BB*

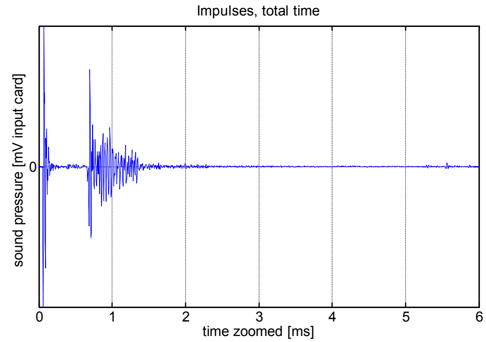
12.7. Profile C, vertically placed

Profile C has tapered shaped slits. This shape causes a stronger reflection than for profile B.

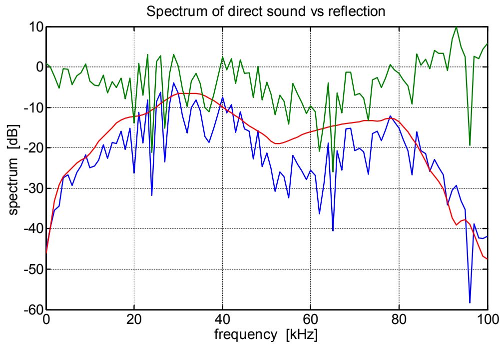
The reflection coefficient graph is very unclear. This probably has to do with the surface of the milled profile. The milling machine worked in the direction perpendicular to the direction of the slits. This caused an uneven surface. Profile B does not show this problem so clearly because the milling machine worked in the direction of the slits for that profile. The layer of paint was not thick enough to make the surface equal.

The reflection coefficient is very hard to read from the graph.

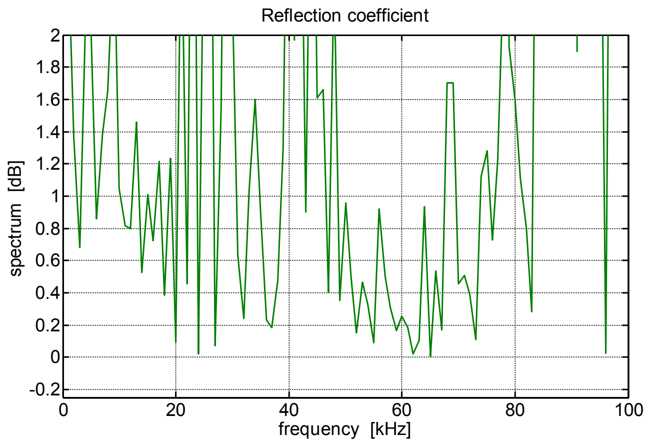
According to the diffraction grating principle, peaks in the reflection coefficient graph should occur. This is not visible for this measurement. The reflection coefficient graph is not understandable.



Fig,19. *Impulses profile C*



Fig,20. *Spectra direct sound and profile C*



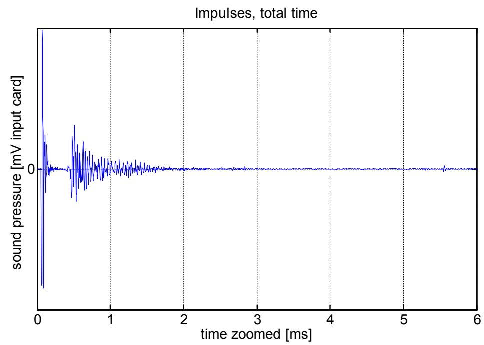
Fig,21. *Reflection coefficient profile C*

12.8. Profile C, horizontally placed

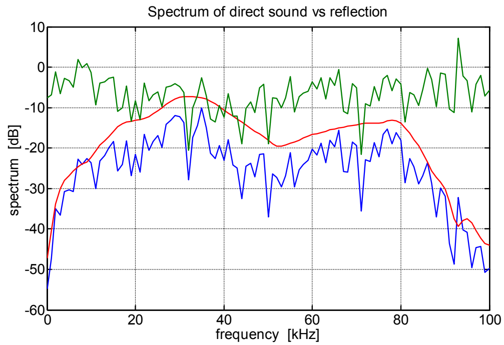
The reflection of the sound is less strong than for the vertically placed panel. This is again caused by the orientation of the panel because the influence of the shape is bigger when the panel is horizontally placed.

The reflection coefficient graph shows a lot of peaks and valleys so that it is hard to read. Though, the average reflection coefficient is about 0,7.

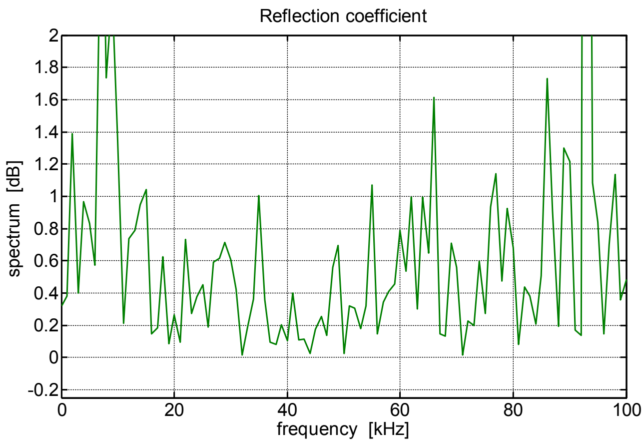
The effect of diffraction grating is not clearly visible in the graph. This is probably because the slits are tapered and the surface is not very smooth.



Fig,22. *Impulses profile C*



Fig,23. *Spectra direct sound and profile C*



Fig,24. *Reflection coefficient profile C*

12.9. Conclusions

The reflection coefficient of profile A is 0,9 which means that it is not a perfectly reflecting surface.

The layer of fleece absorbs more high frequent sound than low frequent sound.

Interference occurs for profiles with slits. This is caused by the scattering of sound. It also explains why the reflection coefficient is sometimes bigger than 1. The microphone receives at that moment two reflections at the same time that are in phase (or close to in phase).

Profile C reflects a bigger part of the sound than profile B. So, the tapered shape works less diffusing.

Profile BB reflects less sound than profile AA for some frequencies because of interference valleys in the reflection coefficient graph. This means that diffusing properties of a profile can lead to very good absorbing properties (for some frequencies). Although, the frequency bands for which this is the case are very narrow. The average sound absorption of a profiled surface will be less than the sound absorption of a flat sound absorbing profile.

The reflection coefficient of profiles with slits are almost equal for both orientations, the graphs of the reflections show stronger reflections when the panel is placed vertically. The graph for forizontally placed profiles is 'quieter'. The orientation has influence on the sound field. The value for the reflection coefficient is not strongly influenced by the orientation.

The phenomenon of diffraction grating occurs for profile B and BB. The straight shape of the slits causes strong reflections of high frequent sound. When the angle of incidence is increased, the interference frequency goes to 27.4 kHz (for 90 degrees, striking sound). This shape of the profile does not have any influence on low frequent sound. Therefore, the distance from the centre of one slits to the centre of the adjacent slit should increase.

For example a structure with a centre to centre distance of 4 meters and striking sound, should strongly reflect 86 Hz sound. This means that structural components with a repetitive distance of 4 meter in a sports hall could cause strong reflections of 86 Hz sound. Although this research focuses on absorption of low frequent sound, it is recommended to have further research on this topic to test this hypothesis and to prevent from unwanted reflections in a new to be built sports hall with repetitive elements.

13. Laboratory measurements

Laboratory measurements are done in order to find out how different parts of a roof structure influence the sound absorbing behaviour of the structure. These measurements give information on the absorbing behaviour of SAB roof structures. Special interest has the absorbing behaviour of the structure to low frequent sound.

The measurements are done in a reverberation room according to ISO standard: ISO 354_2003. It describes how to measure sound absorption in a reverberation room. ISO 9613-1_1993 gives a formula to calculate sound absorption of air. This standard is also summarized below.

13.1. ISO 354_2003

This section is based on the international standard ISO 354:2003(E). It gives the method of measuring absorption coefficients in a reverberant room. The selection of information is based on the used method used during this graduation process.

Sound absorption of test specimen
The average reverberation time in the reverberation room is measured with and without the test specimen mounted. From these reverberation times, the equivalent sound absorption area of the test specimen, AT, is calculated by using Sabine's equation.

Frequencies
Measurements shall be made in one-third-octave bands with the following centre frequencies, in hertz. Especially at low frequencies (below 100 Hz), it could be very difficult to obtain accurate measurement results due to the low modal density of the reverberation room.

Reverberation room and mounting
The volume of the reverberation room is 216 m^3. The test specimen has an area of 10,5 m^2.
The test specimen is installed in a specified mounting. The used mounting is explained in section 13.4.8. The

measurement of the reverberation time of the empty room is made in the absence of the frame of the test specimen.

Temperature and relative humidity

Changes in temperature and relative humidity during the course of a measurement can have a large effect on the measured reverberation time, especially at high frequencies and at low relative humidities. The changes are described quantitatively in ISO 9613-1.
Direrctly after every measurement, the relative humidity, temperature and air pressure is measured with a handheld meter.

Measurement of reverberation time

Two methods of measuring decay curves are described in this International Standard: the interrupted noise method and the integrated impulse response method. For this research, the interrupted noise method is used. The decay curve measured with the interrupted noise method is the result of a statistical process, and averaging several decay curves or reverberation times measured at one microphone/loudspeaker position is mandatory in order to obtain a suitable repeatability.

Microphones and microphone positions

The measurements are made with different microphone positions which are 1,5 m apart, about 3 m from the sound source and 1.5 m from any room surface and the test specimen.

Source positions

The sound in the reverberation room is generated by a sound source with an omnidirectional radiation pattern.

Number of microphone and loudspeaker positions

The number of microphone positions times the number of sound source positions is 12.

Excitation of the room

The loudspeaker source produces white noise. The excitation signal is sufficiently long to produce a steady-state sound pressure level in all frequency bands of interest before it is switched off.

Method of calculation; Calculation of reverberation times T_1 and T_2

The reverberation time of the room in each frequency band is expressed by the arithmetic mean of the total number of reverberation time measurements made in that frequency band. The mean reverberation times of the room in each frequency band without and with the test specimen, T_1 and T_2 respectively, are calculated and expressed using at least two decimal places.

Calculation of A1, A2 and AT

The equivalent sound absorption area of the empty reverberation room, A_1, in square metres, is calculated using the formula:

c is the propagation speed of sound in air, in metres per second;

T_1 is the reverberation time, in seconds, of the empty reverberation room;

m_1 is the power attenuation coefficient, in reciprocal metres, calculated according to ISO 9613-1 using the climatic conditions that have been present in the empty reverberation room during the measurement. The value of m can be calculated from the attenuation coefficient, α, which is used in ISO 9613-1 according to the formula

$$m = \frac{\alpha}{10\lg(\mathrm{e})}$$

NOTE For temperatures in the range of 15 °C to 30 °C, c can be calculated from the formula

$$c = (331 + 0{,}6t/°\mathrm{C})\ \mathrm{m/s} \qquad (6)$$

where t is the air temperature, in degrees Celsius.

Source: ISO 354_2003

The equivalent sound absorption area of the reverberation room containing a test specimen, A_2, in square metres, are calculated using the formula:

$$A_2 = \frac{55{,}3\,V}{c\,T_2} - 4V\,m_2 \qquad (7)$$

where

c and V have the same meanings as in 8.1.2.1;

T_2 is the reverberation time, in seconds, of the reverberation room after the test specimen has been introduced;

m_2 is the power attenuation coefficient, in reciprocal metres, calculated according to ISO 9613-1 using the climatic conditions that have been present in the empty reverberation room during the measurement. The value of m can be calculated from the attenuation coefficient, α, which is used in ISO 9613-1 according to the formula

$$m = \frac{\alpha}{10\,\lg(\mathrm{e})}$$

Source: ISO 354_2003

The equivalent sound absorption area of the test specimen, $A_{T,}$ in square metres, are calculated using the formula:

$$A_T = A_2 - A_1 = 55{,}3V\left(\frac{1}{c_2T_2} - \frac{1}{c_1T_1}\right) - 4V(m_2 - m_1) \qquad (8)$$

where

c_1 is the propagation speed of sound in air at the temperature t_1;

c_2 is the propagation speed of sound in air at the temperature t_2;

A_1, V, T_1 and m_1 have the same meanings as in 8.1.2.1;

A_2, T_2 and m_2 have the same meanings as in 8.1.2.2.

The sound absorption coefficient α_s of a plane absorber or a specified array of test objects is calculated using the formula:

$$\alpha_s = \frac{A_T}{S} \tag{9}$$

where

A_T is the equivalent sound absorption area of the test specimen, in square metres, calculated in accordance with 8.1.2.3;

S is the area, in square metres, covered by the test specimen (see 3.8).

Source: ISO 354_2003

Type A mounting

The sound-absorption properties of the material depend on how that material is mounted during a test. The used frame is solid and has no air space between the test specimen and the frame and between the room surface and the frame. A frame of 32 mm MDF is used.

13.2. ISO 9613-1 : 1993

The formula to calculate sound absorption of air is given below. The α depends on the frequency [Hz], temperature [Kelvin], relative humidity [%] and air pressure [kPa].

a = 8.686 * f ^2 * ((1.84 * 10^-11 * (Pa / Pr)^-1 * (T / To)^(1/2)) + y) [dB/m]
y = (T / To)^(-5/2) * (0.01275 * exp(-2239.1 / T) * (frO + f ^2 / frO)^-1 + z)
z = 0.1068 * exp(-3352 / T) * (frN + f ^2 / frN)^-1
frO = (Pa / Pr) * (24 + 4.04 * 10^4 * h * ((0.02 + h) / (0.391 + h)))
frN = (Pa / Pr)*(T / To)^(-1/2) * (9 + 280 * h * exp(-4.170 * ((T / To)^(-1/3)-1)))
h = hr * ((Psat / Pr) / (Pa / Pr)) = hr * (Psat / Pa)
Psat = Pr * 10^(-6.8346 * (To1/T)^1.261 + 4.6151)

a .... pure-tone sound attenuation coefficient, in dB/m, for atmospheric absorption
s .... distance in m through which the sounds propagates
Pi .... initial sound pressure amplitude, in Pa
Pa ...... ambient atmospheric pressure in kPa
Pr .... reference ambient atmospheric pressure: 101.325 kPa
Psat .. saturation vapor pressure ca equals: International Meteorological Tables WMO-No.188 TP94, World Meteorological Organization - Geneva Switzerland
T .... ambient atmospheric temperature in K (Kelvin). K = 273.15 + Temperature in °C
To ...... reference temperature in K: 293.15 K (20 °C)
To1..... triple-point isotherm temp: 273.16 K = 273.15 + 0.01 K (0.01 °C)
h .... molar concentration of water vapor, as a percentage
hr........ relative humidity as a percentage
f .... frequency
frO ..... oxygen relaxation frequency
frN ..... nitrogen relaxation frequency.
y .... Just a help factor to shorten formula .

13.3. Measurement equipment

13.3.1. The reverberation room

The used reverberation room is located in the acoustical laboratory of LBP|SIGHT in The Hague. Its volume is 216 m^3.

13.3.2. The microphone and sound source

The microphone (B&K) and sound source (Bose) are properties of LBP|SIGHT. The microphone is connected to a rotating rod. In this way, the microphone positions can be changed without entering the room. The sound source produces interrupted white noise.

13.3.3. The real-time frequency analyzer

The used real-time frequency analyser is shown in figure 2. Bruel & Kjaer type 2123.

13.3.4. The amplifier

The amplifier used for the sound source is shown in figure 4.

13.3.5. The rod system

The microphone position can be changed by a rod rotation system, shown in figure 3.

13.3.6. Temperature and relative humidity measurer

A handheld temperature and relative humidity measurer is used of type Lutron LM-81HT.

Fig,1. *Empty reverberation room*

Fig,2. *Real-time frequency analyzer*

Fig,3. *Rod rotation system*

Fig,4. *Amplifier sound source*

13.4. Measurement setup

13.4.1. The panels

Perforated panels used in sports halls can be applied as roof panels and wall profiles. Since the roof panels are the most common profiles used in sports halls, this research will focus on these panels.

13.4.2.2. Selected panels

The selected panel for the laboratory measurements are SAB panels.

- ***Warmdak profielplaat type SAB 106R+/750 P3L-B***
- ***Warmdak profielplaat type SAB 106R+/750 P4L-B***

The perforation degrees are: 23,4% for perfo 3 and 11,7% for perfo 4.

13.4.3.3. Material size

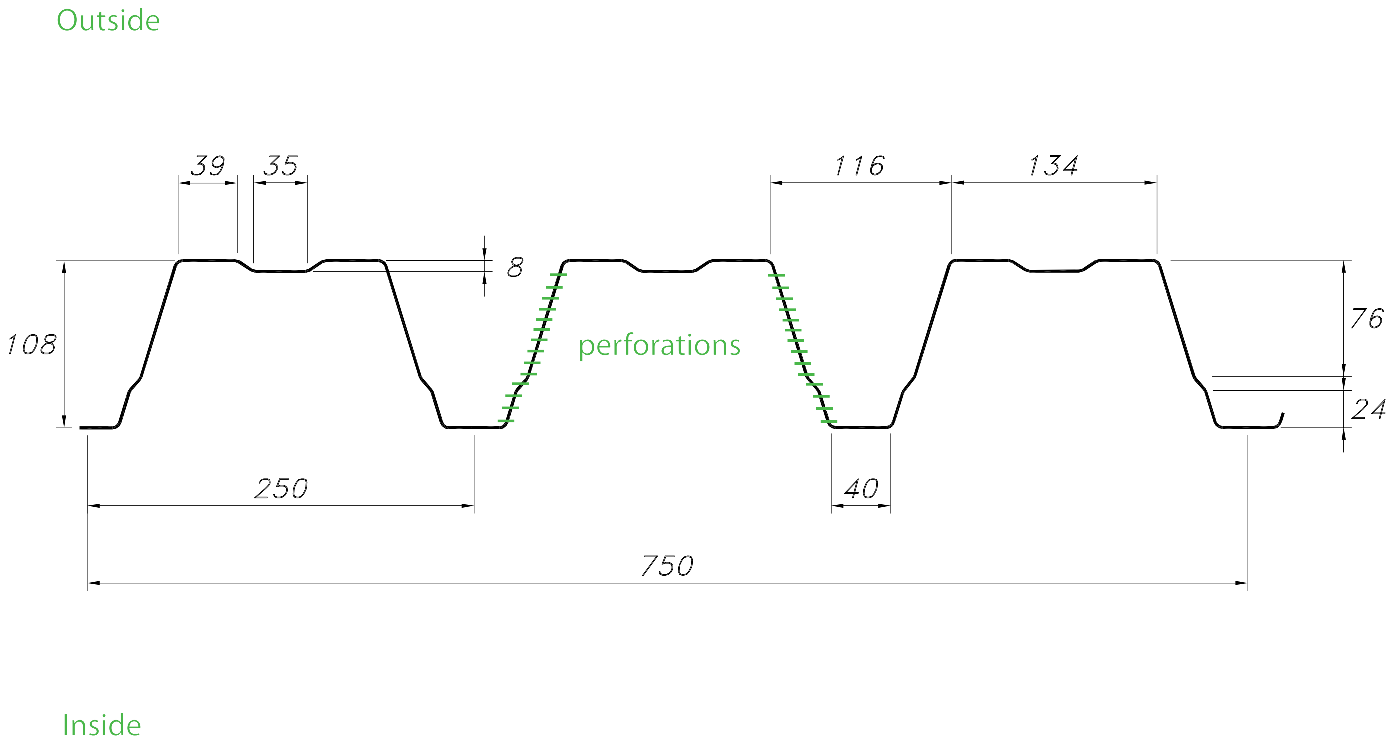
The panels have a steel thickness of 0.75 mm, this is the most used thickness. The length of the panels is 3.5 m.

13.4.4. Roof structure

The roof structure tested in the laboratory consists of different layers:

- ***Profiled steel panel: SAB 106+/750***
- ***'Cannelurevulling' : Rockwool***
- ***Vapour barrier***
- ***Thermal insulation: Rockwool Rhinox***

In the reverberation room, the roof structure is placed upside-down on the concrete floor.



Fig,5. *SAB 106R+/750 profile. (www.sab-profiel.nl)*

13.4.5. Cannelurevulling

'Cannelurevulling', the sound absorbing strips of rockwool that fit exactly in the profiles of the corrugated roof structure. The filling is sealed with a vapour barrier. Density: 30 kg/m^3.

13.4.6. The vapour barrier

A plastic foil is used as a vapour barrier. The foil weights 73 gr/m^2. It has a thickness of 75μm.

13.4.7. Thermal insulation

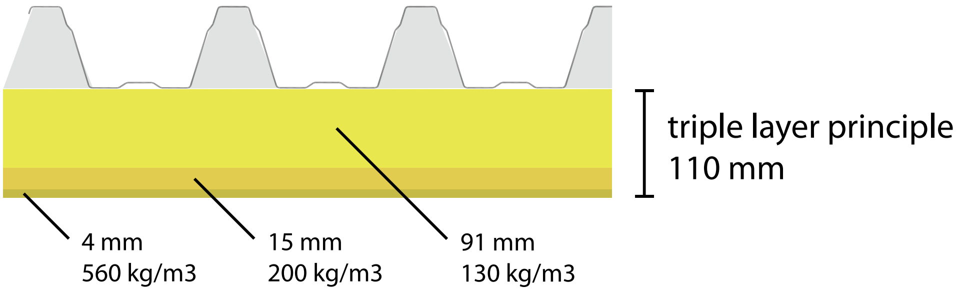
The thermal insulation is Rhinox insulation, 2000x600 mm. Rock wool.
Thickness: 110mm. Density according to firma Rockwool: 155 kg/m^3. Calculated at home: 166 kg/m^3.
More information in Attachment E.

Fig,6. *Thermal insulation*

Fig,7. *Vapour barrier*

Fig,8. *'Cannelurevulling'*

Fig,9. *Roof structure*



Fig,10. *Triple layer principle in 'Rhinox' thermal insulation panel.*

13.4.8. The mounting

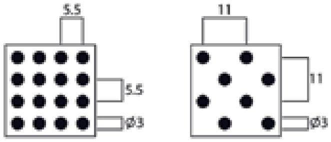
The standard describes the way a test sample should be framed by a mounting. The used mounting (type A) is constructed from two layers of MDF, with a total thickness of 32 mm. For sample A, B, C and D a height of 225 mm is used. The thinner samples are framed by a lower mounting; 110mm.

The corners of the mounting are constructed with use of pre fabricated corner elements to make sure no sound could enter the sample via the corner seams.

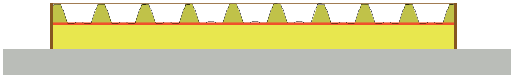
Fig,11. *Mounting A*

Fig,12. *Prefab corner*

Fig,13. *Building a sample*

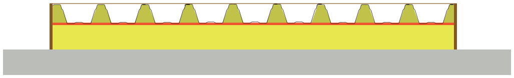


Perfo 3 Perfo 4



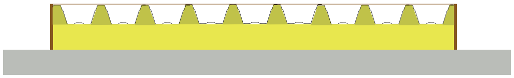
A3

Perfo 3 profile; 'cannelurevulling'; vapour barrier; thermal insulation.



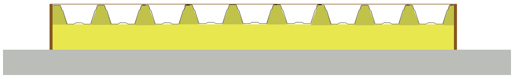
A4

Perfo 4 profile; 'cannelurevulling'; vapour barrier; thermal insulation.



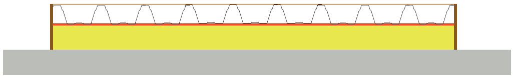
B3

Perfo 3 profile; 'cannelurevulling'; thermal insulation.



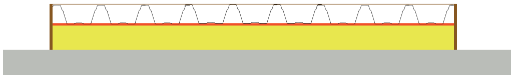
B4

Perfo 4 profile; 'cannelurevulling'; thermal insulation.



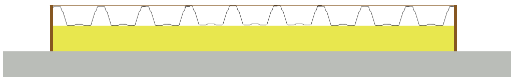
C3

Perfo 3 profile; vapour barrier; thermal insulation.



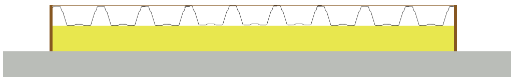
C4

Perfo 4 profile; vapour barrier; thermal insulation.



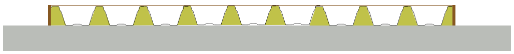
D3

Perfo 3 profile; thermal insulation.



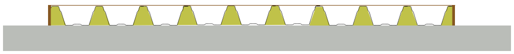
D4

Perfo 4 profile; thermal insulation.



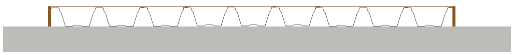
E3

Perfo 3 profile; 'cannelurevulling'.



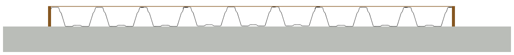
E4

Perfo 4 profile; 'cannelurevulling'.



F3

Perfo 3 profile.



F4

Perfo 4 profile.

G

Fig,14. *Variations of test samples measured in reverberation room*

13.5. Measurement method

The measurements on absorption and diffusion are done according to the standard. Different configurations are measured concerning absorption. The variations are shown in figure 14.

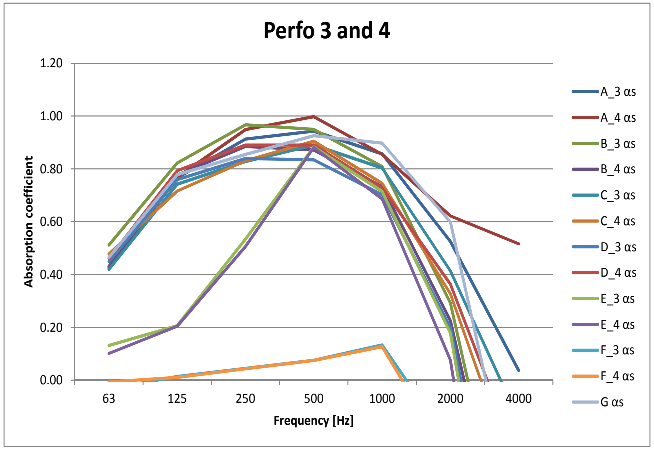
Directly after every measurement, the relative humidity, temperature and air pressure were measured with a handheld meter shown in figure 16.

After the first session of measurements, there seemed to be one ‘cannelurevulling’ forgotten. Every slit was filled with three pieces of cannelurevulling, except for one slit. All measurements with cannelurevulling have one missing piece, in this way the results are comparable. The measured values for sound absorption might be little lower than expected because of this. Though, the influence on the results is expected to be small.

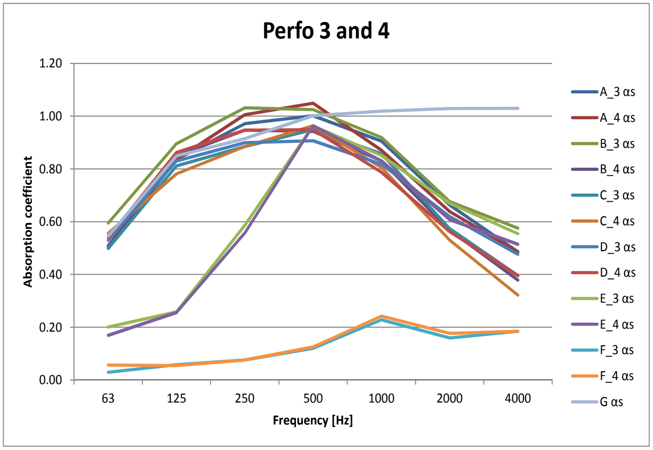
The mounting is adjusted (by changing the length of the sides) per sample to make sure it fits as good as possible.

Fig,15. *Missing cannelurevulling*

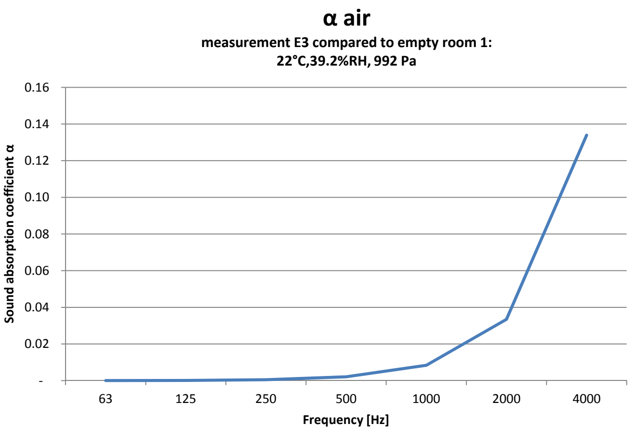
Fig,16. *Handheld meter*



Fig,1. *Sound absorption coefficient of 'empty 1'.*



Fig,2. *Sound absorption coefficient of 'empty 1' without air absorption.*



Fig,3. *Sound absorption of air for sample E3.*

14. Laboratory measurement results and discussion

This chapter presents the results of sound absorption coefficient measurements. Differences between the roof panels with perfo 3 and perfo 4 are discussed. Besides that, the influence of the vapour barrier, the thermal insulation and cannelurevulling are displayed.

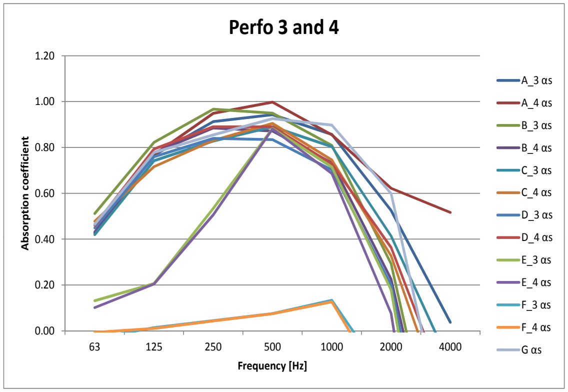
14.1. Influence of sound absorption of air

The results strongly depend on the sound absorption of air; relative humidity, temperature and air pressure. This section will explain the influence of this parameter.

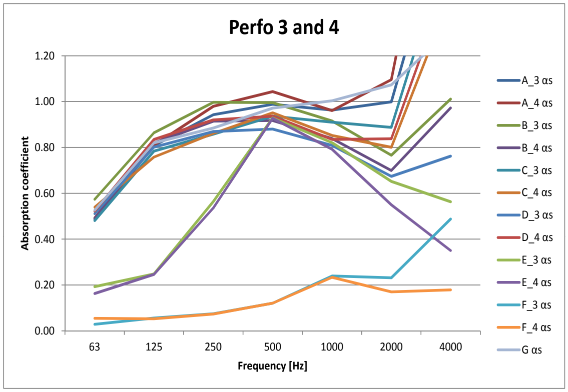
The graphs in figure 1 and 2 show the sound absorption coefficient of different samples tested in the reverberation room. The sound absorption coefficient is calculated by comparing the sound absorbing properties of the empty room *without* sample to the room *with* a sample.

Most of the differences in the graphs are found for high frequencies. This is logically because the sound absorption of air increases for higher frequencies (see figure 3). Also, the temperatures, relative humidity and the air pressure play a role. Of these parameters, the relative humidity is the most unstable factor. It directly changes when somebody enters the room.

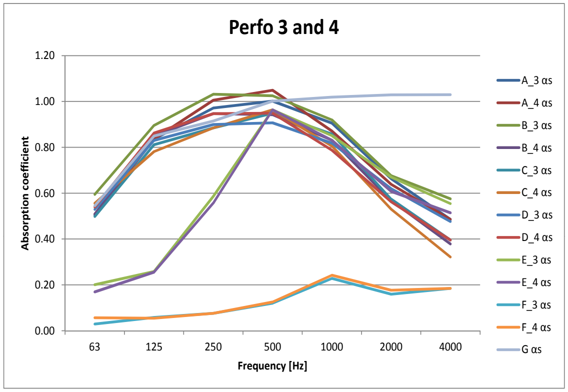
According to the graphs, the influence of the sound absorption of air is very important to the results. Because the measurements of air properties are done with a handheld meter, the measurement accuracy is not high enough to give reliable results for high frequencies.



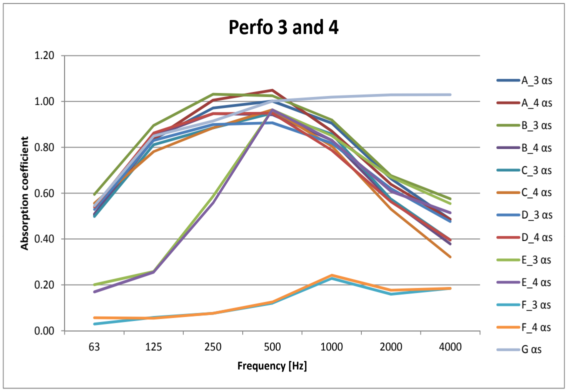
Fig,4. *Sound absorption coefficient of 'empty 1'.*



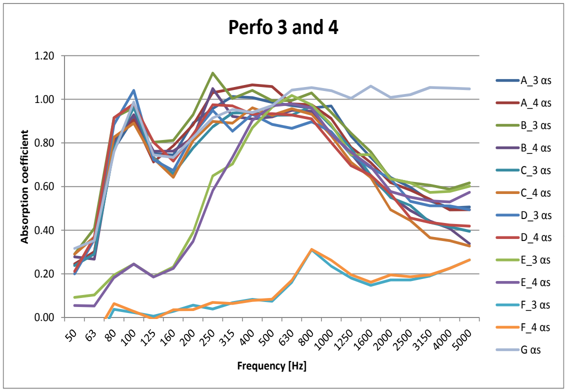
Fig,6. *Sound absorption coefficient of 'empty 2'.*



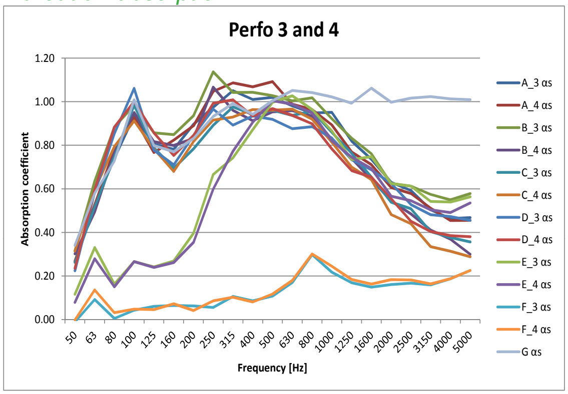
Fig,5. *Sound absorption coefficient of 'empty 1' without air absorption.*



Fig,7. *Sound absorption coefficient of 'empty 2' without air absorption.*



Fig,9. *Sound absorption coefficient of 'empty 1' without air absorption in terts bands.*



Fig,8. *Sound absorption coefficient of 'empty 2' without air absorption in terts bands.*

14.1.1. Influence of empty room measurement

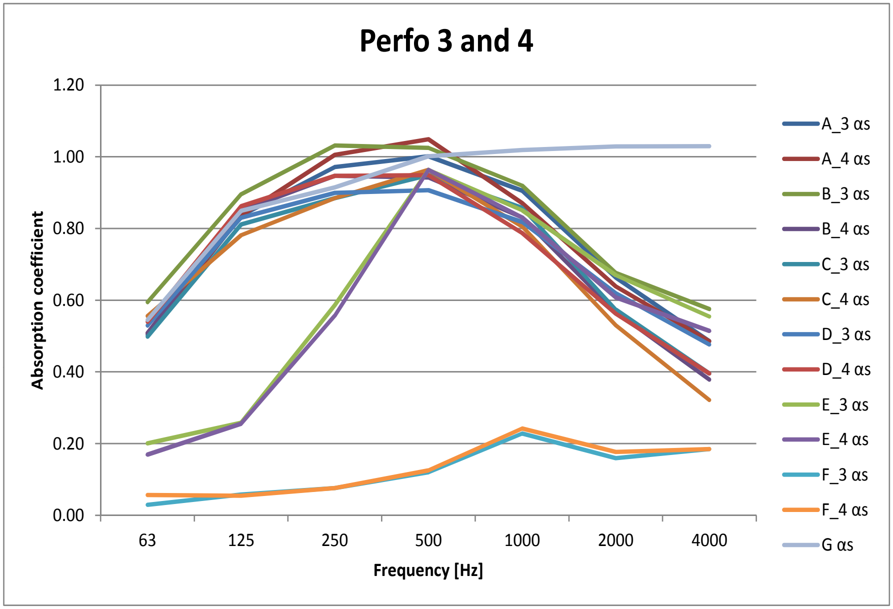
Reverberation times of the empty room have influence on the results. This section will discuss the influence of this parameter.

The reverberation time of the empty reverberation room is measured twice, once before and once after the series of measurements. The first one is called ‘empty 1’, the second one is called ‘empty 2’.

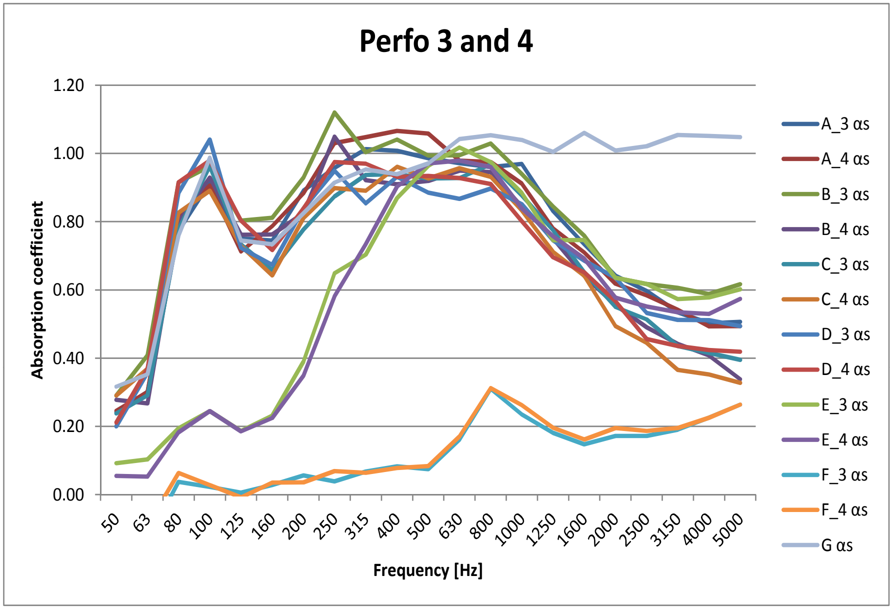
When comparing empty 1 and 2 (figure 6 and 8), a big difference in results is found. When the sound absorption of air is not taken into account, the graphs for empty 1 and 2 (figure 7 and 9) are the same. The sound absorption per terts band shows small differences between the two. Especially low frequent sound absorption is different. This is probably caused by the diffusion of the reverberation room. Creating a good diffuse sound field for low frequencies is much harder than for high frequencies because of the length of the sound waves.

The properties of the empty room did not change during the measurement sessions. The big difference for high frequencies is caused by the measured properties of air and so the sound absorption of air. The smaller differences for low frequencies are probably caused by the diffusity of the room.

The reverberation room has a good diffuse sound field. Together with stable measurement equipment, it gives reliable results.



Fig,10. *Results of measurements per octave band compared to 'empty room 1' without sound absorption of air correction.*

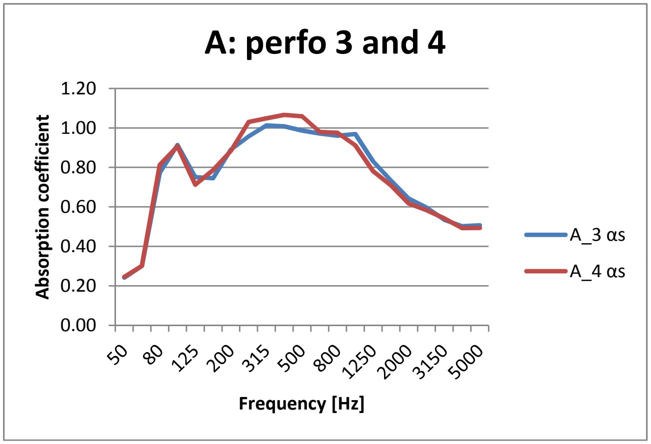


Fig,11. *Results of measurements per terts band compared to 'empty room 1' without sound absorption of air correction.*

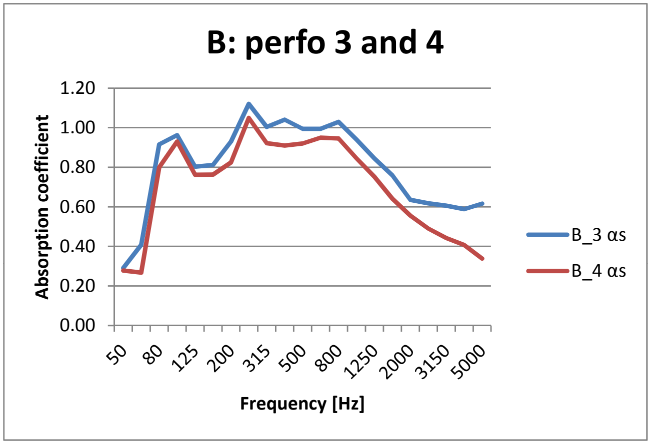
14.2. Octave band or 1/3rd octave band

Normally, results of sound absorption measurements are presented per octave band in a graph. This is as expected, according to the standard. Octave band results though, give less information about the absorbing behaviour than terts band results. This is shown in figure 10 and 11. The results are summarized in an easily readable graph.

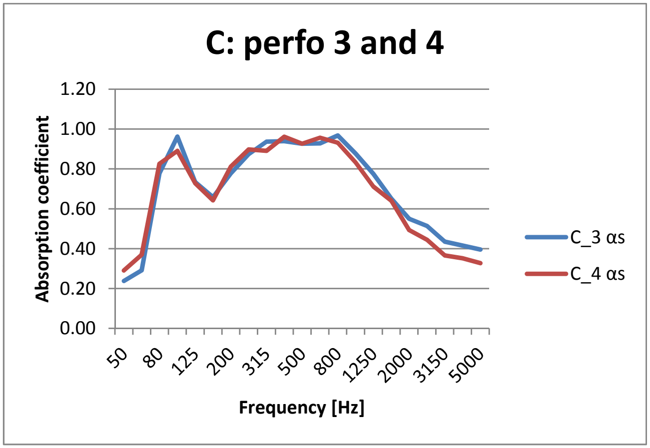
When using terts band results, small measurement errors or diffusity problems are visible. This means that a small deviation should be taken into account.



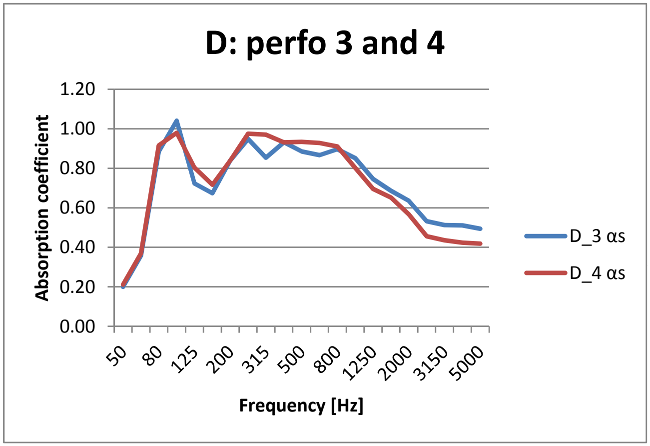
Fig,12. *Absorption coefficient; test sample A*



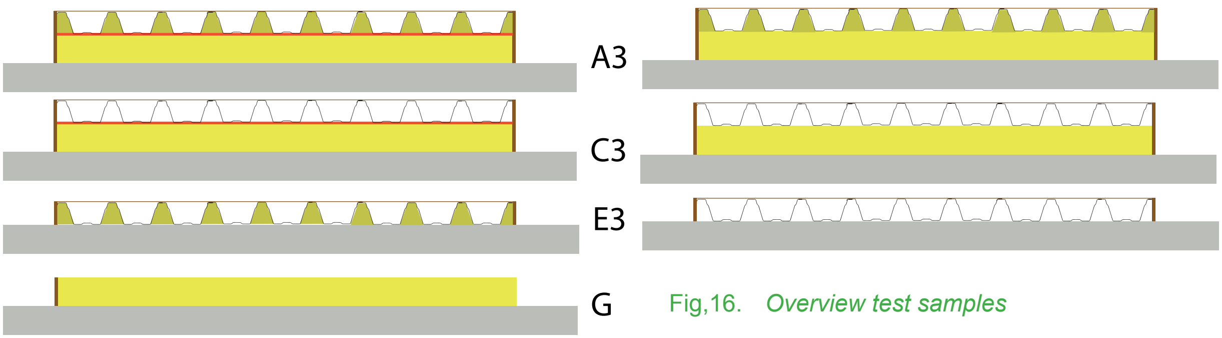
Fig,13. *Absorption coefficient; test sample B*



Fig,14. *Absorption coefficient; test sample C*



Fig,15. *Absorption coefficient; test sample D*



Fig,16. *Overview test samples*

14.3. Results and discussion

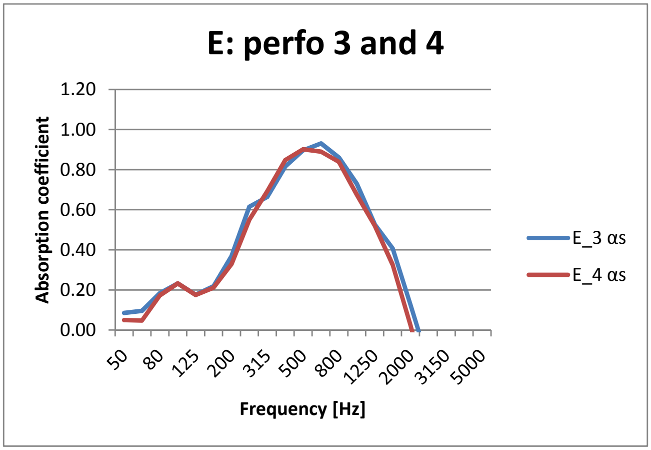
Since this graduation research focuses on the absorbing properties of the panels for low frequent sound, the high frequencies are less important. To decrease the deviation caused by inaccurate reading of the air properties equipment, the results used for this analysis are based on measurement ‘empty 1’ without a sound absorption of air correction.

14.3.1. Test sample A and B

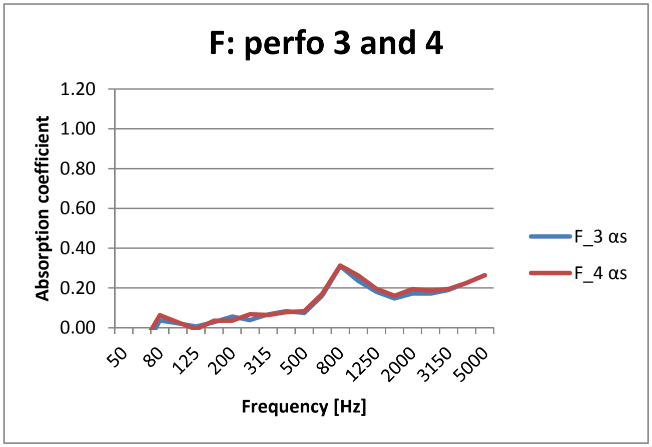
The sound absorption graphs show an arc-like curve. This shape is typically for a Helmholtz resonator as described in chapter 8.1. The shape of the graph of a porous material can also be seen for sample G (see next page), where just the porous rockwool is tested.

Expected from the theory in chapter 8.1, a higher perforation degree should lead to a peak in the sound absorption graph at a higher frequency. However, the blue line (perfo 3) shows a peak at about 315 Hz, where this is 400 Hz for the red line (perfo 4) (figure 12). The difference with the theory is probably caused by the backing insulation and vapour barrier. The graph of sample E shows the expected (but small) difference between perfo 3 and perfo 4 panels.

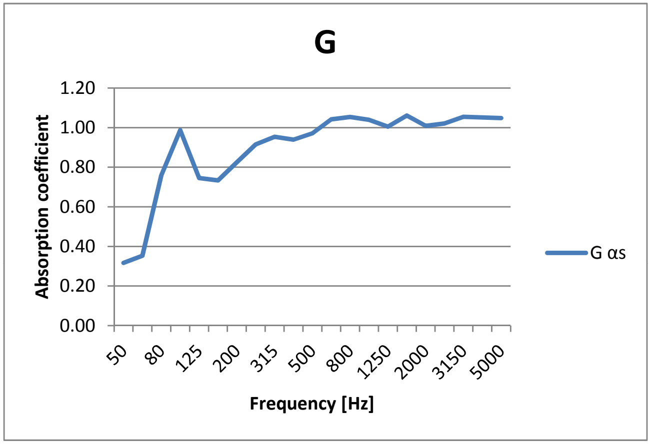
Another striking fact is the peak (α=0.9) of both lines at 100 Hz. This peak could be caused by something in the material or something in the measurement equipment or the reverberation room. Since the power frequency of the electricity grid in The Netherlands is 50 Hz, the peak could be caused by the measurement equipment that transforms power into physical movement. Although, the peak is not found for sample E and F. This means that it is probably not caused by the room or the equipment. It is a property of the tested sample. The peak is caused by the sound absorbing properties of the thermal insulation. This material, shown in graph G, shows an even higher peak (α=0.95) at 100 Hz.
The stiffness of the insulation panels possibly cause the ‘extra’ sound absorption because the panels can act like a panel absorber. Section 14.3.4 gives more information on the causes of the 100 Hz peak.



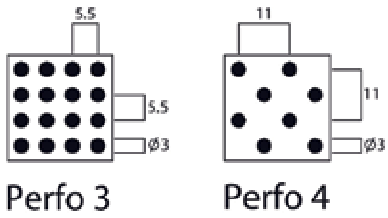
Fig,17. *Absorption coefficient; test sample E*



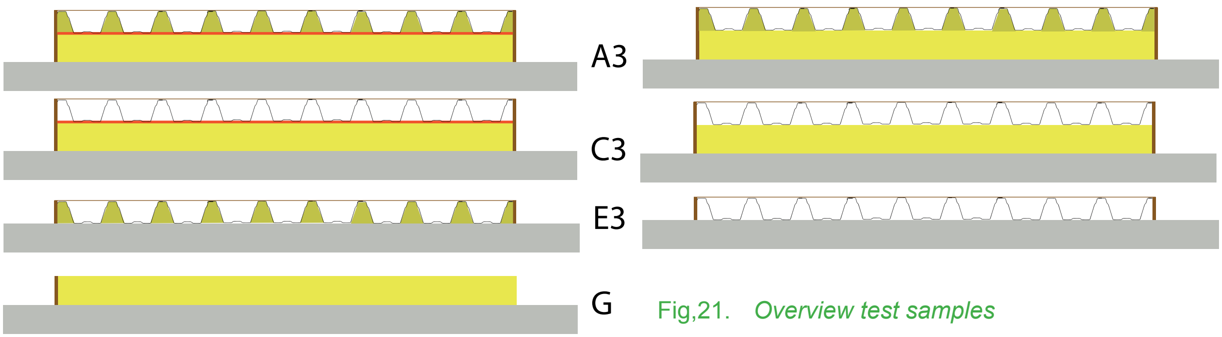
Fig,18. *Absorption coefficient; test sample F*



Fig,19. *Absorption coefficient; test sample G*



Fig,20. *Differences in perforation degree for perfo 3 and 4*



Fig,21. *Overview test samples*

What happens for sample B, without vapour barrier?
The higher absorption values for perfo 3 are caused by the higher perforation degree. The material covers less of the insulation than perfo 4. In this way, the sound absorbing properties of the insulation get more important in the results. The difference in the high frequencies is caused by the foil. Perfo 3 shows more of the foil than perfo 4, so that the thin layer can absorb more high frequent sound. The same effect can be seen for sample C and D.

The second peak at 250 Hz is also visible when comparing samples C and D. The samples without vapour barrier (B and D) show the peak.

14.3.2. Test sample C and D

Like the difference between sample A and B, the difference of the vapour barrier between sample C and D causes an increase and a decrease in sound absorption. Low frequency sound is better absorbed by the samples without foil, high frequency sound by the samples with foil.

Again, theoretical expectations from chapter 8.1. are not found. Adding an impervious layer would cause a strong decrease in sound absorption. Of course, the layer is not the same, but the effect is not found.

The difference between the samples with and without 'cannelurevulling' is big for frequencies of 160 Hz and higher. The more open panel perfo 3 shows a bigger decrease in absorption without cannelurevulling than perfo 4. This of course, has to do with the more open structure.

Low frequent sound absorption does not change a lot by adding or removing the 'cannelurevulling'. This is not the case though, for sample E and F. The backing insulation (and foil) cause therefore most of the low frequent sound absorption.

14.3.3. Test sample E and F

The difference in perforation degree is not visible for sample F, though it is for sample E. The expected shift in peak frequency occurs as expected although the difference is small.

The difference between the graphs of E and F is probably almost exactly the sound absorbing behaviour of the cannelurevulling.

14.3.4. Test sample G

The results of sample G are different from the expectations. The peak at 100 Hz is not expected. This peak is probably caused by the stiffness and large weight of the insulation panels.

Hypothese 1: Panel resonator; different densities in one panel

The whole panel possibly acts like a panel resonator. The mass-spring theory is now applicable to one panel. A part of the panel acts like the mass, another part as the spring. The different densities and thicknesses of the layers (see figure 22) are important for the resonance frequency which can be calculated by the formula shown on the right.

$$f_r = \frac{1}{2\pi}\sqrt{\frac{1.4p_0}{m_1 D}}$$

With:

m_1 = mass in kg/m^2

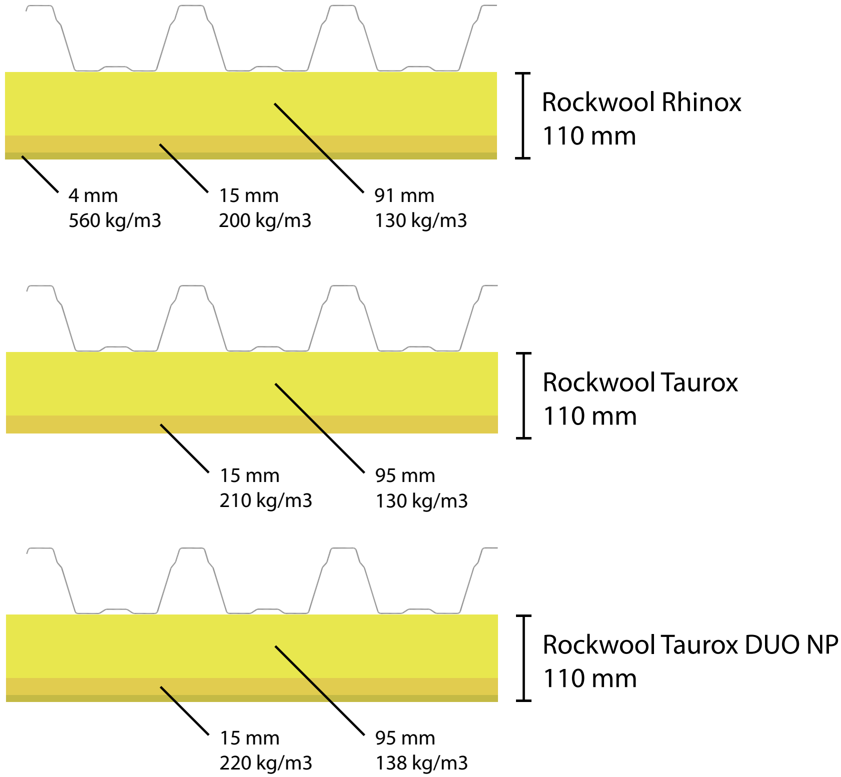
D = distance to rigid backing wall

p_0= 10^5 [Pa]

The calculation below is based on the idea that the tick part (light yellow) starts vibrating on the thin (darker yellow) layers below. For Rockwool Rhinox, Taurox and Taurox DUO panels, the resonance frequency then is:

Panel resonator	Rhinox	Taurox	Taurox DUO	Rhinox	
fr	126	138	134	105	
m1	11.83	12.35	13.11	12.8	heavier
D	0.019	0.015	0.015	0.025	with thin layer of air

So, this could be an answer to the phenomenon. But, since the formula is based on air, and the stiffness of the panel is different than air, we need to use another formula. Besides that, section 14.5.3 of this chapter will show that the same peak is also found for mono-density panels. Therefore, the phenomenon is probably not caused by the different densities in one panel.



Fig,22. *Different layers of thermal insulation panels. (source: Rockwool)*

Hypothese 2: Panel resonator; floating screed formula

A more useful formula to calculate the resonance frequency of a panel resonator is given below. This formula is based on the calculation method for floating screeds (Drolenga 2005).

$$f_r = \frac{1}{2\pi}\sqrt{\frac{s'_t}{m}}$$

With:

f_r = resonance frequency

m = mass of the mass

s'_t = stiffness of the spring

$$s'_t = \frac{E_{dyn}}{d}$$

$E_{dyn} \approx 0.3$ - 1.3MPa for rock wool

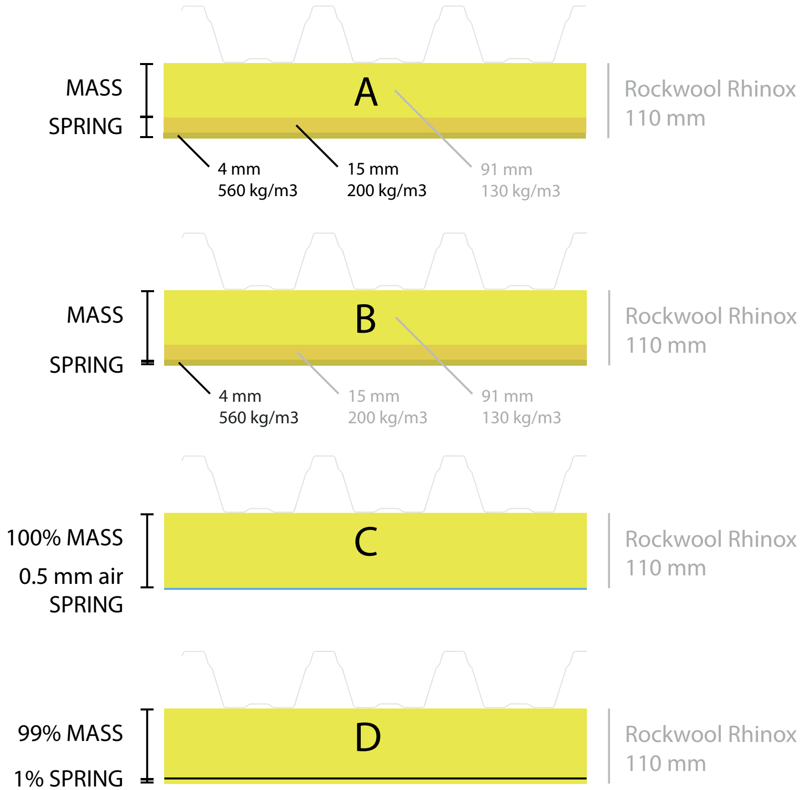
d = thickness of the spring

Options A, B, C and D are investigated for the Rhinox rock wool panel. Option A has a mass of the top, lowest density layer and a spring of the two other layers. Option B has a mass of the two top layers and a spring of the lowest layer. Option C has just the total mass and a spring of a 1mm thick layer of air under it. This layer could be there because of the material being placed on the concrete ground surface of the reverberant room. Option D is comparable to option C but here the mass consists of 99% of the total mass and thickness and the spring of the 1% of the total mass and thickness. The last option is a little strange to test because of the different densities in the panel. It is less strange for mono-density panels.

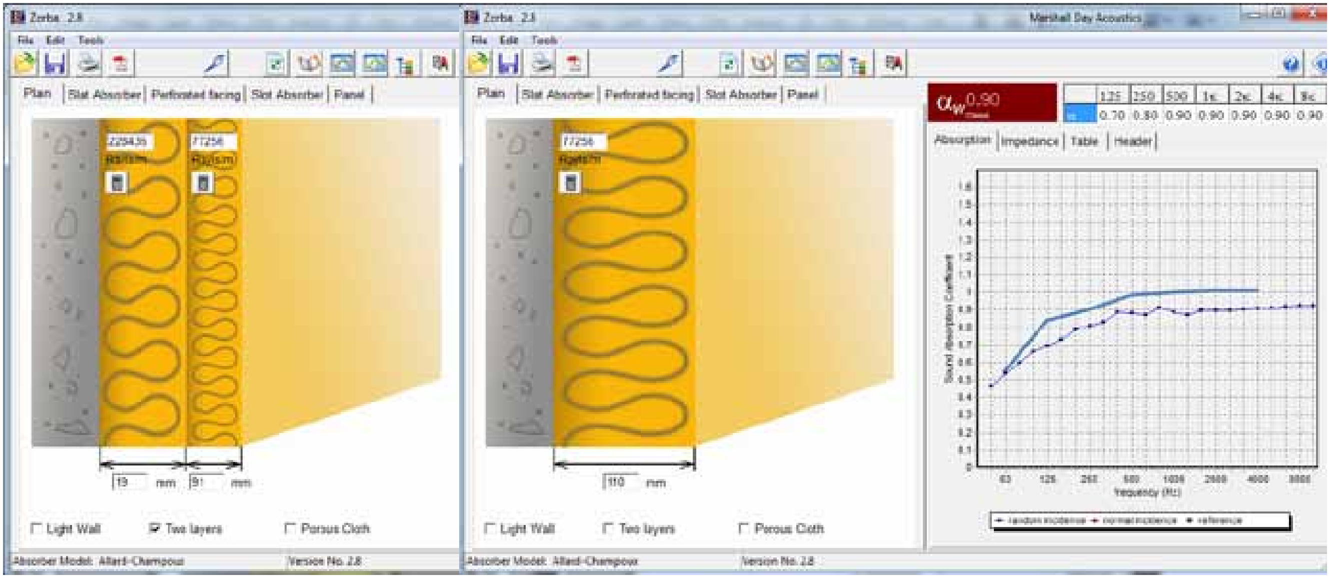
The resonance frequency as found in the measurements lays in between 80 and 125 Hz for Rhinox rockwool panels.

		Rhinox 110mm option A		Rhinox 110 mm option B		Rhinox 110 mm option C		Rhinox 110 mm option D	
	fr [Hz]	61	29	115	25	103		212	102
mass	m [kg/m2]	11.83	11.83	15.8	15.8	17.07		16.9	16.9
spring	Edyn [Pa]	1300000	300000	1300000	300000	140000		1300000	300000
spring	d [m]	0.019	0.019	0.004	0.019	0.0005		0.0011	0.0011

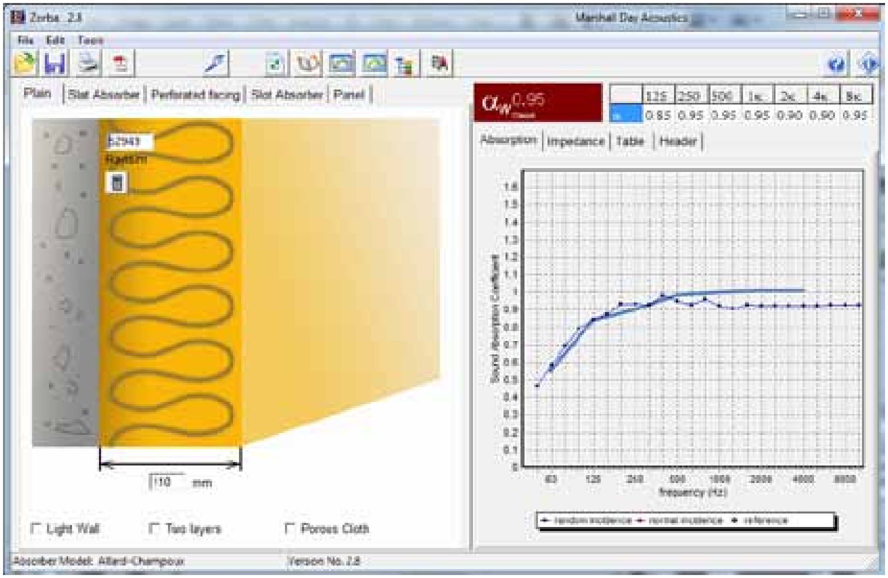
Option A is falsified. Option B, C and D are possible solutions.



Fig,23. *Rhinox panel resonator options*



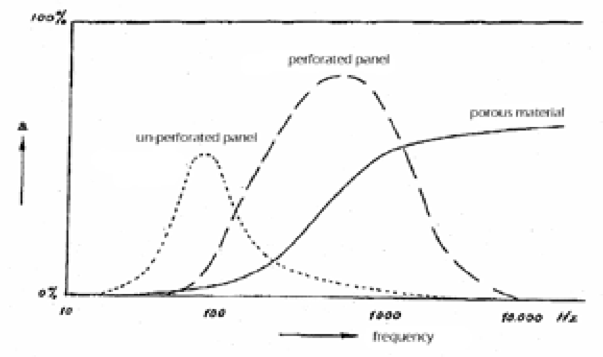
Fig,24. *Expected sound absorption for simplified triple layer principle (l) and mono layered rockwool (r).*



Fig,25. *Best comparable result: 110 mm, 100kg/m³ , rockwool.*

14.4. Sound absorption expectations

In principle, the sound absorbing behaviour of the test samples can be calculated beforehand. Perforated panels behave differently than for example porous materials, see figure 26. The programme Zorba is used to calculate the absorption coefficients shown in this section.



Fig,26. *Different mechanisms compared (Nederhof 2005)*

14.4.1. Thermal insulation

Test sample G consists of thermal insulation framed by a MDF construction. When the properties are filled out in the programme, the sound absorption graph slightly differs from the measured results. See figure 24 (r), the measured result is shown by the thick blue line, this is the octave band result of sample G. The outcome does not change when the triple layer principle is simplified in the model. The results stay the same, see figure 24 (l). This finding probably falsifies option A and B of figure 23. The panel has to be seen as one mass.

When trying to simulate the Rhinox insulation as good as possible, the best results are found for a 110 mm thick (mono or triple layered) panel of rockwool with a density of 100 kg/m^3. This is less than the known 130 kg/m^3 for the main layer of a Rhinox panel.

The slightly higher sound absorption value found for the measurements might be caused by the wooden frame. It is possible that the material of the frame absorbs a part of the sound.

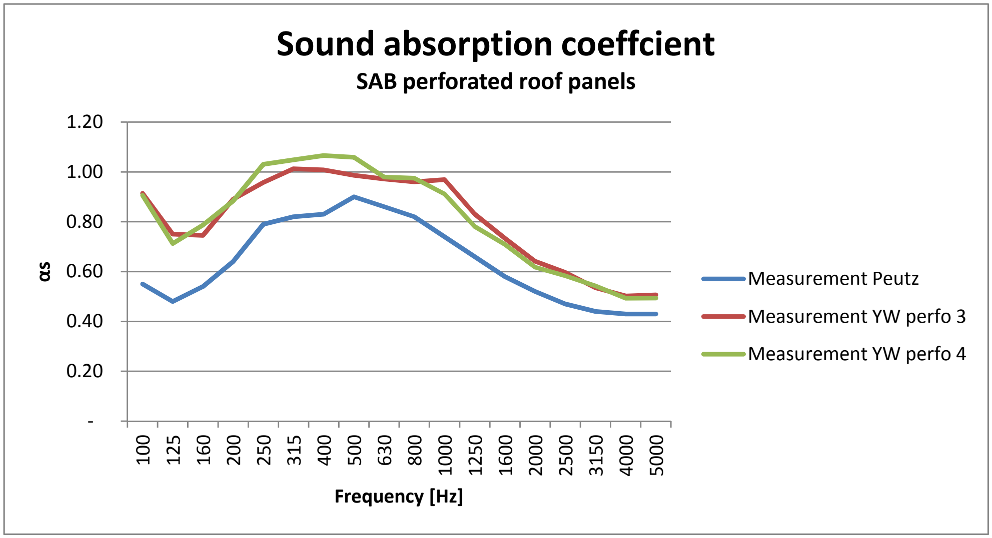
14.5. Comparison with other reports

This section will discuss results of other test reports of SAB roof panels.

14.5.1. Peutz report A 1568-3, figure 27

Peutz, an independent consultant in the field of acoustics and other disciplines has tested SAB roof panels. A page of the report is shown in figure 27.

The tested sample is comparable to sample A, tested for this research, although I did not create a cavity with mineral wool on the floor of the reverberation room. Also, a thicker SAB panel is used. The results of sample A3, A4 and the sample tested by Peutz are shown below.



Somehow, the construction (see figure 27) absorbs less sound than sample A3 and A4. The shape of the curve, though, is very much the same. Striking are the increasing values for low frequent sound. The behaviour of the Rhinox panels seems to be comparable. The good sound absorbing behaviour for low frequent sound is probably also found for frequencies lower than 100 Hz, but not displayed in the report because the graph does not show results of frequencies lower than 100 Hz.

LABORATORIUM VOOR AKOESTIEK

PEUTZ

GELUIDABSORPTIEMETING IN DE NAGALMKAMER CONFORM ISO 354:2003

opdrachtgever: Rockwool Benelux BV

TESTEN RvA L 334

onderzochte constructie: dakvariant 1

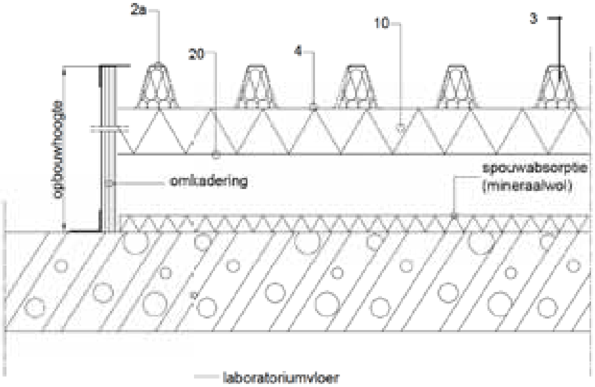
2a. Geprofileerde stalen dakplaat 106/0,88 geperforeerd

3. Steenwol cannelurevulling ingeseald in PE-folie

4. Dampremmende laag: 0,2 mm PE – folie

10. Rhinox 110 mm; ca. 17,0 kg/m²

20. gewapende PVC dakbedekking , 1,2 mm, ca. 1,5 kg/m²



volume nagalmkamer: 214 m³

oppervlak monster: 12,35 m²

opbouwhoogte: 0,4 m

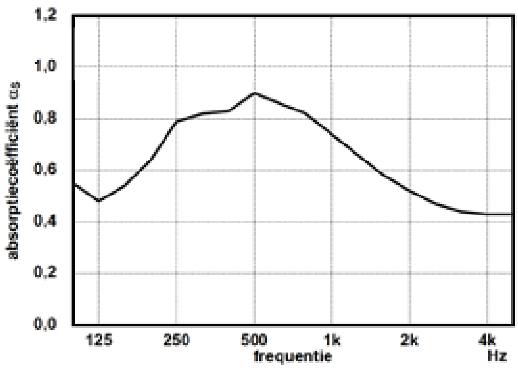
gemeten in: laboratorium

signaal: breedband ruis

bandbreedte: 1/3 octaaf

α_w (ISO 11654) = 0,55(LM)

NRC (ASTM - C423) = 0,75



		125	250	500	1k	2k	4k
——— 1/3 oct.		0,55	0,64	0,83	0,82	0,58	0,44
	1/3 oct.	0,48	0,79	0,90	0,74	0,52	0,43
* 1/1 oct.		0,54	0,82	0,86	0,66	0,47	0,43
	1/1 oct.	**0,52**	**0,75**	**0,86**	**0,74**	**0,52**	**0,43**

publicatie is slechts toegestaan in de vorm van dit gehele blad

Mook, 10-03-2006

rapport nr. A 1568-3

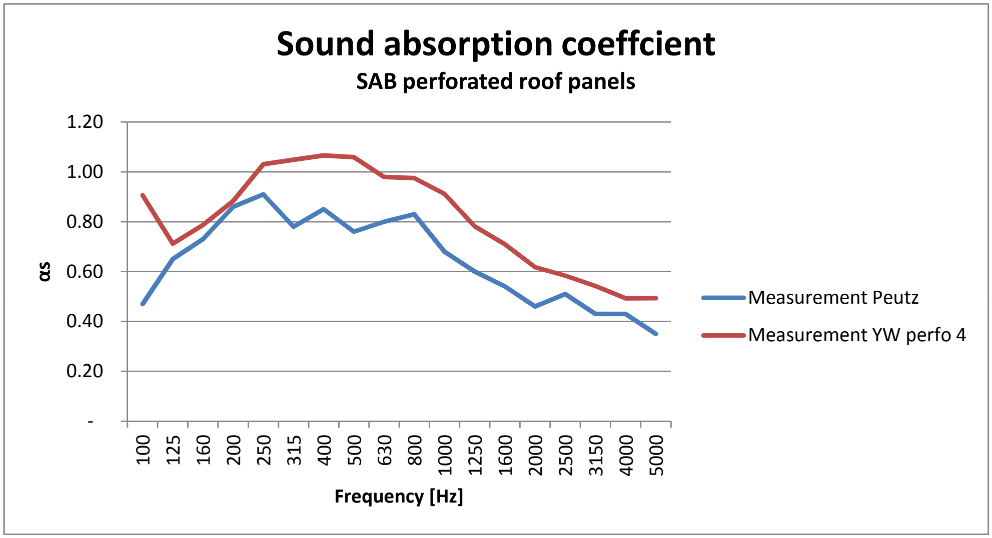
figuur nr. 17

Absorb versie 4.2.1, mode 7 bestandsnaam: a1568 E#:1095-1130 F#:1134-1171 A#:1228 T_1 = 16,1 °C T_2 = 16,4 °C p_1 = 99,7 kPa p_2 = 99,7 kPa h_1 = 48,0 % h_2 = 53,2 %

Fig,27. *Testreport SAB panels, 2006. (www.rockwool.nl)*

14.5.2. Peutz report A 330-1, p.6

Earlier, in 1991, Peutz got the assignment to test the sample shown in figure 18. It is constructed from SAB 106/P4 panels, cannelurevulling, thermal insulation (100 mm rock wool, 35 kg/m³) and bitumen. The difference with the measurement mentioned before, is the thermal insulation layer. This layer is much less dense than the Rhinox panels and is not constructed out of 3 different layers.



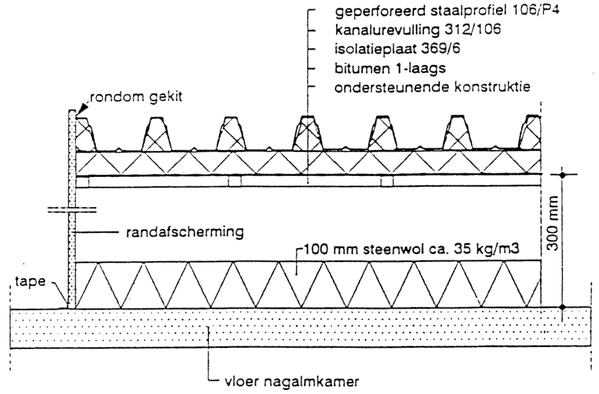
Fig,28. *Sound absorption perforated panels, tests compared.*

Somehow, the construction absorbs less sound than sample A4. This could be caused by for example the different mountings and the different insulation with an other density. The shape of the curve, though, is very much the same. Still, the increasing sound absorption for low frequent sound is not found. This could be explained by the fact that the thermal insulation layer is less thick, has a different density or that it is not constructed of different layers of rockwool with different densities.

ADVIESBUREAU PEUTZ & ASSOCIES B.V. rapport nr. A 330-1 figuur nr. 8

GELUIDABSORPTIEMETING in de nagalmkamer volgens ISO 354-1985
Opdrachtgever: Rockwool Lapinus B.V.

onderzochte konstruktie:



volume nagalmkamer: 196 m^3

oppervlak monster: 10.8 m^2

hoogte: 0.4 m

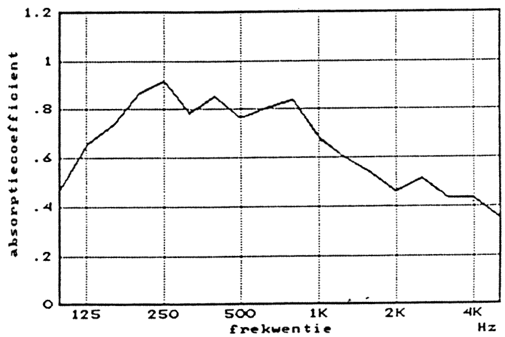
gemeten in: laboratorium

signaal: roze ruis

bandbreedte: 1/3 oktaaf

temperatuur: 12.0 °C

relatieve vochtigheid: 50.0 %



NRC (ASTM - C423): 0.70

———: 1/3 okt

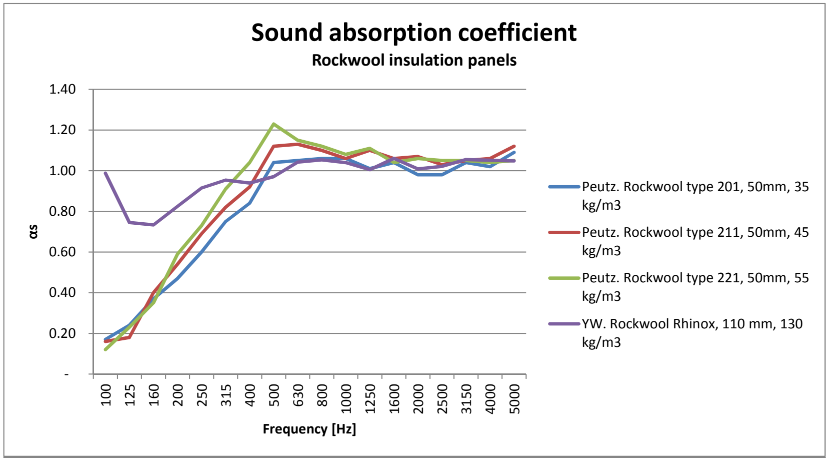
frekw.	125	250	500	1K	2K	4K	Hz
1/3 okt	0.47	0.86	0.85	0.83	0.54	0.43	
	0.65	0.91	0.76	0.68	0.46	0.43	
	0.73	0.78	0.80	0.60	0.51	0.35	
1/1 okt	0.62	0.85	0.80	0.70	0.50	0.40	

datafile: A330 A#:163

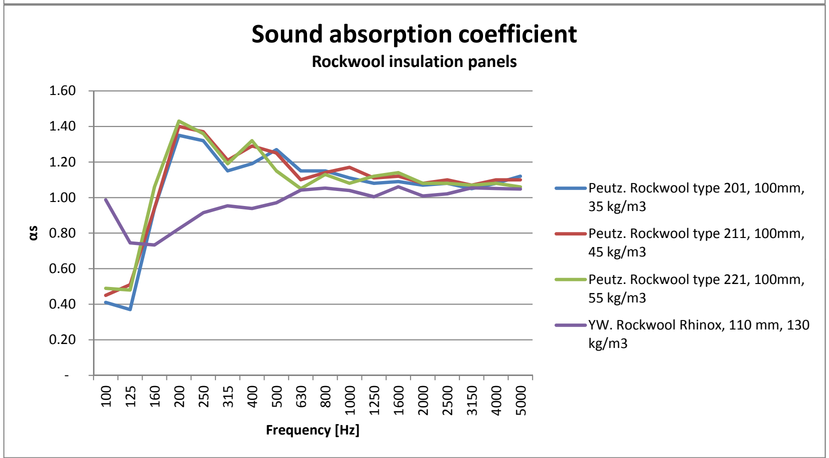
publikatie is slechts toegestaan in de vorm van dit gehele blad

Nijmegen, datum: 03-04-91

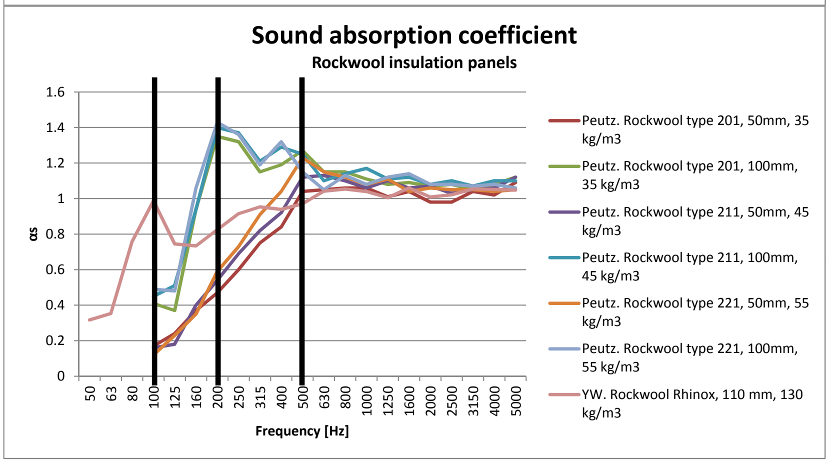
Fig,29. *Testreport SAB panels, 1991. (source: LBP|SIGHT)*



Fig,30. *Rockwool panels 100mm, sound absorption coefficient.*



Fig,31. *Rockwool panels 50mm, sound absorption coefficient.*



Fig,32. *Rockwool panels, sound absorption coefficient, moving peak.*

14.5.3. Peutz report A 423-1

Report A 423-1 of 1992, gives information of different Rockwool insulation panels. The panels have different thicknesses and densities. The results of the report are compared to the found result for test sample G.

The comparison on the left page gives us information about the effects of different densities of the panels. The denser the material, the higher the peak in the sound absorption graph (see fig. 30 and 31). The frequency of this peak does not change with the density.

Also, different thicknesses of the panels are compared. The thicker the panel, the lower the peak-frequency. Though, the found peak is not expected from the theory. Figure 26 in previous section shows a smooth curve for the sound absorption of porous materials.
When we have a look at figure 32, it seems to be the case that this 'unexpected peak' moves to the left for denser tested materials. This is visualized by the black vertical lines.

The hypothesis of the panel being its own mass-spring resonator is confirmed by this information. Although the phenomena also appears for panels with just one density. The part of the hypothesis that the upper layer starts to vibrate on the thin, more dense, lower layers is therefore falsified. Still, the panel resonator theory is plausible since the peak moves to low frequent sound when the density is increased.

The charts on the next page give the resonance frequencies for the different rock wool panels with a thin layer of air (0.5 mm and 1.5 mm) between the panel and the floor. It shows that a 0.5 mm thick layer of air can cause a resonance frequency of about 200 Hz for 100 mm thick rock wool samples. On the other hand, a thicker layer of air is necessary to achieve resonance frequencies of about 500 Hz for 50 mm thick rock wool panels. It is not plausible that a heavier panel has a thicker layer of air under it than a less 'lightweight' panel. It falsifies the hypothesis of the layer of air.

The last hypothesis I tested is the mass-spring ratio of 99%/1%=99% of the mass and thickness of the panel osculates on 1% of it. The charts in figure 33 and 34 show that this hypothesis is most plausible. Although the E_{dyn} value has large influence on the results, the calculations show that the resonance frequency of heavier panels is lower than the resonance frequencies of the more 'lightweight' ones.

All in all leads the comparison with other reports and the checks of different hypotheses to the conclusion that the found high values for sound absorption of low frequent sound (about 100 Hz) are likely caused by the very dense, thick thermal insulation panels. This way of shaping causes a panel resonator principle inside the thermal insulation panels.

		Rockwool type 221 100 mm, 55 kg/m3		Rockwool type 211 100 mm, 45 kg/m3		Rockwool type 201 100 mm, 35 kg/m3	
	fr [Hz]	181		200		227	
mass	m [kg/m2]	5.5		4.5		3.5	
spring	Edyn [Pa]	140000		140000		140000	
spring	d [m]	0.0005		0.0005		0.0005	

		Rockwool type 221 50 mm, 55 kg/m3		Rockwool type 211 50 mm, 45 kg/m3		Rockwool type 201 50 mm, 35 kg/m3	
	fr [Hz]	256		283		320	
mass	m [kg/m2]	2.75		2.25		1.75	
spring	Edyn [Pa]	140000		140000		140000	
spring	d [m]	0.0005		0.0005		0.0005	

		Rockwool type 221 50 mm, 55 kg/m3		Rockwool type 211 50 mm, 45 kg/m3		Rockwool type 201 50 mm, 35 kg/m3	
	fr [Hz]	467		516		585	
mass	m [kg/m2]	2.75		2.25		1.75	
spring	Edyn [Pa]	140000		140000		140000	
spring	d [m]	0.00015		0.00015		0.00015	

Fig,33. *Check of 'air layer' hypothesis*

mass/spring ratio		50/50		99/1		99/1		99/1	
		Rockwool type 221 100 mm, 55 kg/m3		Rockwool type 221 100 mm, 55 kg/m3		Rockwool type 211 100 mm, 45 kg/m3		Rockwool type 201 100 mm, 35 kg/m3	
	fr [Hz]	78	37	391	188	433	208	491	236
mass	m [kg/m2]	2.75	2.75	5.445	5.445	4.455	4.455	3.465	3.465
spring	Edyn [Pa]	1300000	300000	1300000	300000	1300000	300000	1300000	300000
spring	d [m]	0.05	0.05	0.001	0.001	0.001	0.001	0.001	0.001

		99/1		99/1		99/1	
		Rockwool type 221 50 mm, 55 kg/m3		Rockwool type 211 50 mm, 45 kg/m3		Rockwool type 201 50 mm, 35 kg/m3	
	fr [Hz]	783	376	865	416	981	471
mass	m [kg/m2]	2.723	2.723	2.228	2.228	1.733	1.733
spring	Edyn [Pa]	1300000	300000	1300000	300000	1300000	300000
spring	d [m]	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005

Fig,34. *Check of '99%/1 % mass-spring ratio' hypothesis*

14.6. Conclusions

The conclusions of the laboratory measurements are as follows.

The sound absorption of air increases with frequency. It has a big influence on the results of high frequent sound. Because this research focuses on low frequent sound, this is not a big problem. Though, it is recommended for laboratory employees to work very accurately with the air properties.

Octave band results are less informative than terts band results.

Reverberation times found for the empty reverberation room are stable. This factor does not cause differences in results when compared to a room with test sample. The possible differences are mainly caused by the sound absorption of air.

The shape of the graph displaying the test results is typical for a Helmholtz resonator. Also, the porous material acts as expected.

The difference between perfo 3 and perfo 4 is smaller than expected for samples with backing insulation and vapour barrier.

Because of the open structure of perfo 3 panels, its results are strongly influenced by the backing layers. For perfo 4 panels, this is less applicable.

The vapour barrier, being a thin layer of foil, absorbs high frequent sound. The thermal insulation layer without foil absorbs more low frequent sound. The layer does not dramatically decrease the sound absorbing properties of the sample.

The effect of cannelurevulling is not big when a vapour barrier is used. When this is not the case, the difference caused by the cannelurevulling increases.

Rhinox thermal insulation panels absorb sound as expected like a ‘normal’ porous material except for a strong increase of absorption around 100 Hz sound. This additional peak might be the answer to the main question of this research. Since the peak is not visible in octave band results, the panels are not expected to strongly absorb 100 Hz sound. Therefore, a measurement in a sports hall constructed with perforated panels and this type of insulation is not expected to give very high sound absorption results of low frequent sound.

The peak in the absorption graph is probably caused by the panel being its own plate resonator. Though, further research on this hypothesis is recommended.

IV. Conclusions

15. Conclusions

This chapter will be the capstone of all chapters so far. Of all conclusions, the most important ones will be mentioned. In this way, it is a summary of the most useful conclusions which can be used during the design phase of this graduation project. The different main sections of this report are used to organize the conclusions.

Section 15.4 gives the answer to the research question.

15.1. The problem

Reverberation times are used to test acoustics in sports halls. Of all possible parameters, it is the most stable, easy to measure one. It has disadvantages too, such as unreliable values when the Schroeder curve has a sagging shape. This sagging shape is caused by the difference in the amount of sound absorbing material between the vertical sound field and the horizontal one. The horizontal sound field is so important because of the used materials on the walls. Because of the activities in the hall, hard, sound reflecting materials are used to construct a wall of a sports hall. These materials do not absorb much sound. The roof, however, does absorb much of the sound as a result of which this vertical field quickly decays. In this way, the horizontal sound field is more important than the vertical one when talking about acoustics in sports halls.

Legislations and standards on acoustics in sports halls used in the Netherlands have been changed during the last years. Though, the aim is to find a way (or parameter) to define and create good acoustics in sports halls. This might become part of the Dutch Bouwbesluit in the future.

The scattering coefficient is a very important input value for simulation programmes like Catt Acoustic and ODEON. Changing the scattering coefficient of a surface has large effect on the parameters: reverberation time, C80 and D50. A standard value of 0,10 often gives acceptable results but more knowledge on scattering coefficients is needed to make realistic simulations.

15.2. Sound absorption

There are three types of sound absorbers: porous materials, plate resonators and perforated panel resonators. Perforated panel absorbers belong to the group of Helmholtz resonators. Their sound absorbing working depends on: the percentage perforation, the thickness of the facing, the diameter of the holes, the tickness of the backing material, whether or not an air space between the backing material and the facing is used and whether or not an impervious layer between the facing and backing material is used.

15.3. Measurements

15.3.1. Scale model

Although they are two totally different phenomena, scale model measurements show that the effect of scattering is comparable to the effect of sound absorption (for low frequent sound). The shape of the surface is very important when talking about sound absorbing properties of that surface.

The scale model tests also show that the Q-value of the surface has big influence on the sound absorbing properties. In combination with the angle of incidence, the Q value can cause a decreased or an increased sound pressure level at interference frequencies.

The reflection coefficient measurements on a scale model of a roof structure show that tapered structures have smaller scattering effects than profiles with straight slits. This scattering effect results in an interference pattern at which some frequencies have a very high sound absorption value, and some a very low one.

The orientation of the slits is important for the scattering effects.

15.3.2. Laboratory

The laboratory measurements lead to the following useful conclusions for to the design phase.

Rock wool insulation panels can absorb low frequent sound (around 100 Hz, depending on the thickness and density) very well. Rhinox rockwool panels strongly absorb 100 Hz sound.

The perforation degree strongly influences the sound absorbing properties. The more open the structure, the bigger the influence of the backing layers on the sound absorbing properties of the whole sample.

Sound absorption of air plays a large role for high frequent sound. Especially in sports halls, which have a large volume, this factor should not be forgotten.

A vapour barrier can increase the sound absorbing properties on high frequent sound. On the other hand, it slightly reduces the low frequent influence of the backing thermal insulation layer.

15.4. Answer to the research question

The scale model and laboratory measurements are done to test two hypotheses on the research question. The answer to this question is derived from the conclusions of two tested hypotheses.

Hypotheses:

1. A perforated steel panel behaves differently in practice than in a laboratory situation on absorption coefficient because the shape of the panels causes sound absorption of parallel striking sound based on a phase shift principle.

2. A perforated steel panel behaves differently in practice than in a laboratory situation on absorption coefficient because the backing construction has influence on the result.

Research question:

'Why do perforated steel sound absorbing panels seem to behave differently in absorption coefficient in sports halls than expected from laboratory test results?'

15.4.1. The answer

Perforated steel sound absorbing panels absorb more low frequent sound than expected. This good sound absorption is probably achieved by the profiled shape of the steel panels in combination with the backing thermal insulation.

The profiled shape diffuses the sound, which causes a local decrease in sound pressure level. In this way, it causes the effect of sound absorption. Also, the ground effect of the surface influences the sound absorbing behaviour. Interference of direct and reflected sound can cause (when extinguishing) the effect of sound absorption aswell.

The measurements done in the laboratory show that rock wool insulation panels have a high peak in the absorption graph at 100 Hz. (Or higher up to 500 Hz, depending on the thickness.) The stiff, heavy and thick insulation panel is probably its own panel resonator because of its proportions. The thicker, the more the peak shifts to lower frequencies. This peak is probably part of the reason for the good low frequent sound absorption of perforated steel panels in sports halls.

The results from the laboratory measurements show that rock wool insulation panels are very good sound absorbers. Measurements show that the sound absorption slightly decreases when the perforated steel panels are added to the insulation. The steel panels are necessary though, because the profiled shape causes the effect of sound absorption by diffusion and interference as mentioned before. Besides that, a roof structure without the steel panels can not exist seen from a structural and practical point of view. The combination of the rock wool insulation and the perforated, profiled steel roof panels is good. The roof structure is a good sound absorber.

The research does not give strong guidelines to improve the roof structure. The gained knowledge is used in the design of a wall panel. The walls up to three meters are still the weakest link when talking about acoustics in sports halls. The designed panel should improve this situation.

15.4.2. Recommendations

Since not all possible hypotheses on the research question are tested in this graduation research, I recommend further research on the following topics:

- Which part of the (low frequent) sound is transmitted through the structure and therefore absorbed?
- Is the good low frequent sound absorption of perforated steel panels caused by vibration of the panels?

I also recommend further research on the found peak in the rock wool sound absorption graph.

- The insulation panel is a porous material and probably a panel resonator in one, can this be proved?
- What is the influence of ‘theoretical oscillation’ in the sound absorption graph of Rhinox rock wool panels?

V. Design

16. Sports sound

The sound produced in a sports hall depends on the type of activity. For example, a bouncing basketball produces low frequent sound and beeping shoes produce high frequent sound. But what kind of sound (which frequencies) are most important for different sports?

This chapter gives the results of a short and not very scientific research on background noise in a sports hall. The aim is to learn what kind of sound belongs to different activities. This knowledge is necessary when designing a wall of a sports hall.

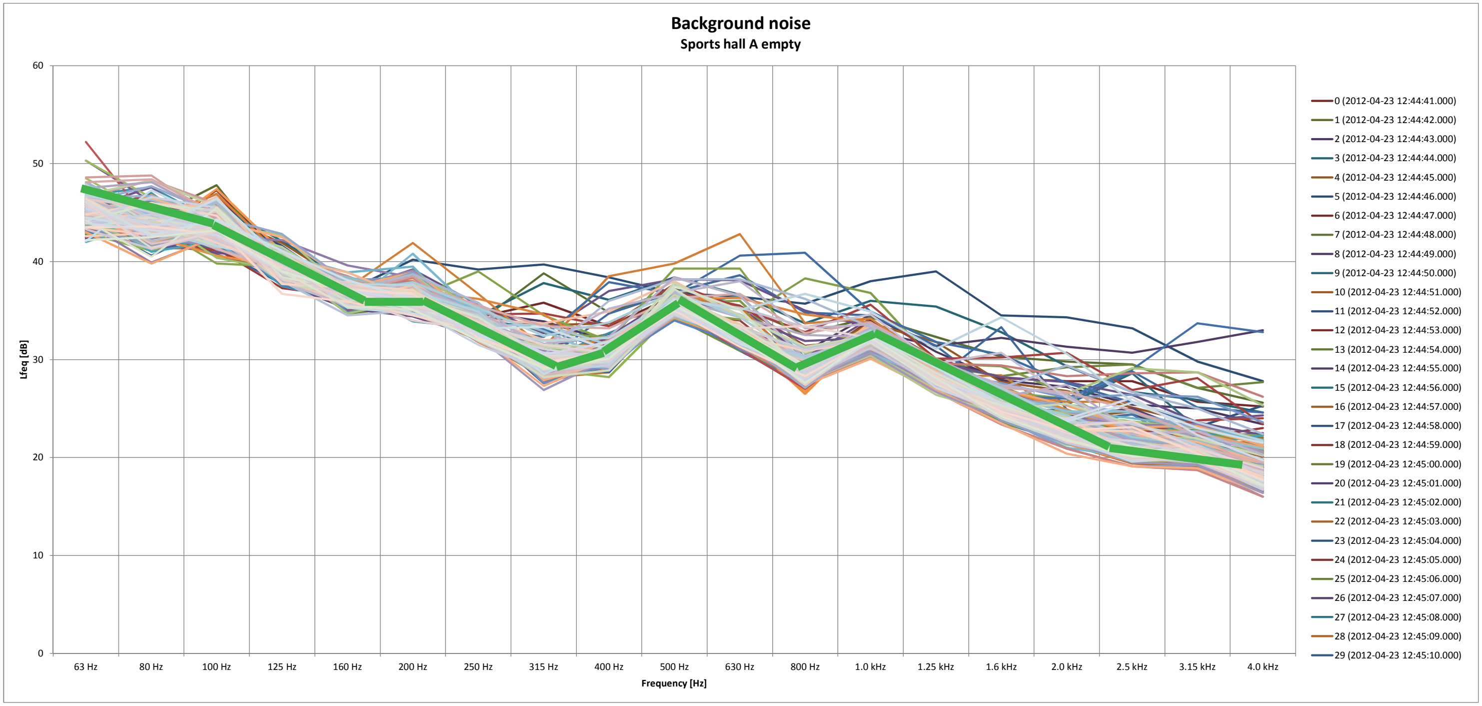
The measurements are done with a handheld sound measurer, type Norsonic 140. The graphs in this chapter show a curve for every second of the measurement (total 5 minutes).

The sound absorbing materials in the hall have effect on the results. This is not taken into account since the research focuses on global information about frequencies of sounds produced by shoes, balls and voices.

16.1. Sports hall A, empty

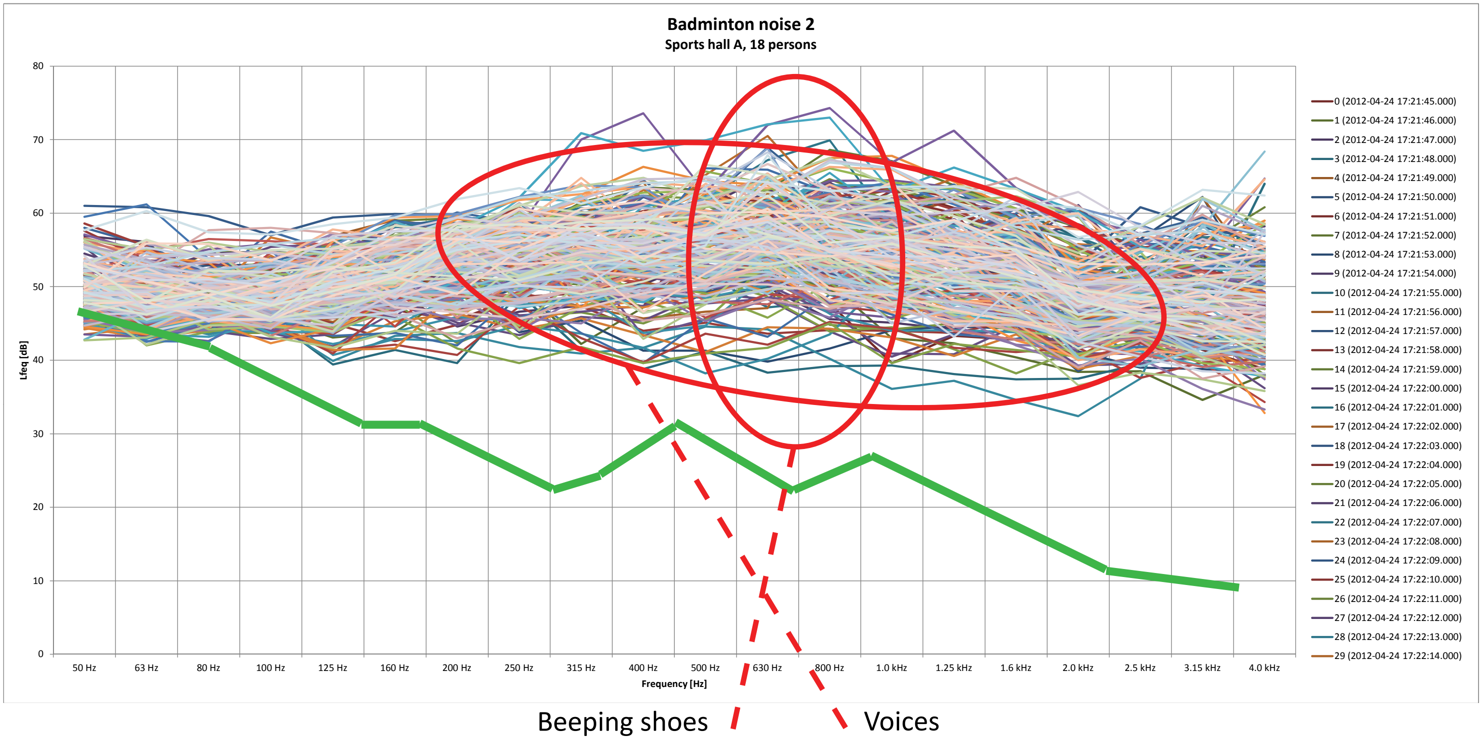
Firstly, the background noise in sports hall A is measured. The background noise is shown in the figure below.

Typically is the low frequent caracter of the sound. The system produces a peak in sound pressure level for 500 Hz and 1000 Hz. Furthermore, the sound pressure decreases when the frequency increases. The green line is an average which is used for the comparison with other measurements in this chapter.



16.2. Badminton hall A

A group of 18 badminton players produces the following sound:

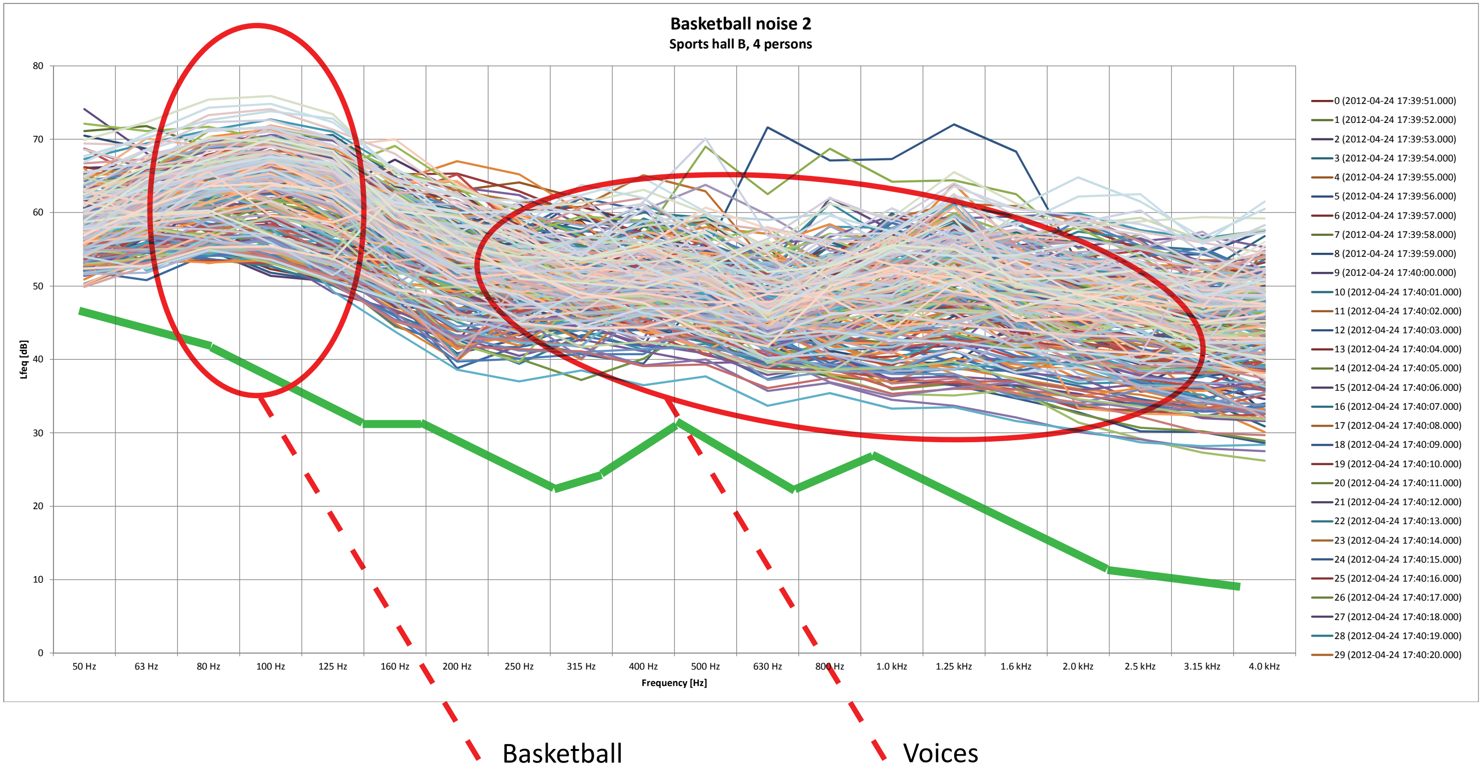


When comparing to the background noise, the sound produced by the persons in the hall is spread out over most mid-frequency bands. The sound of beeping shoes is probably found in the frequency bands between 500 and 1000 Hz. Furthermore, the differences between the empty hall and the used hall are bigger when the frequency increases.

The sound pressure level in a used hall is much higher than in an empty hall. This could also be caused by the ventilation system which was probably not turned on fully in the empty hall.

16.3. Basketball hall B

A group of 4 basketball players produces the following sound:

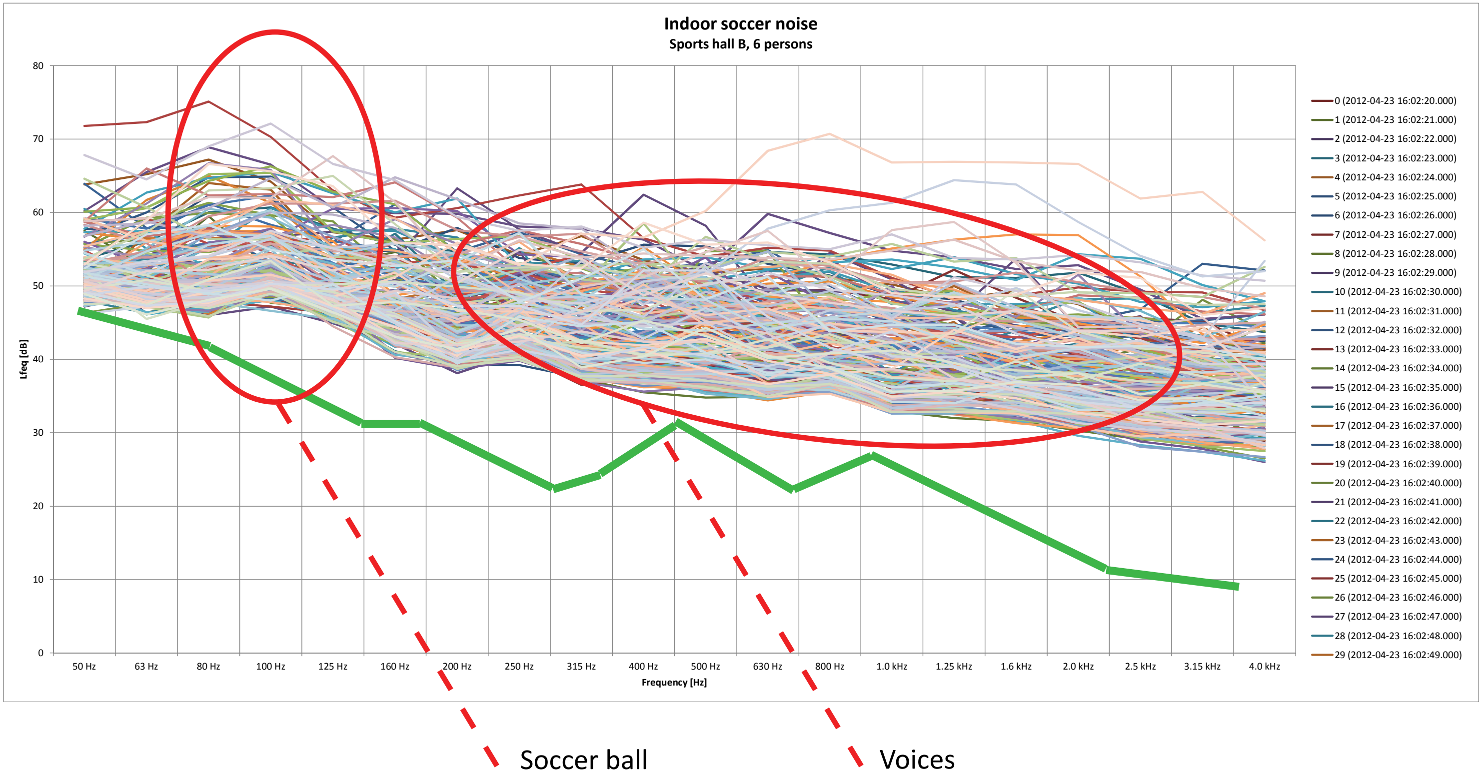


The measurement in sports hall B, a smaller sports hall, on basketball sound shows two clear things. The first is the sound caused by the ball, which is low frequent (63-125 Hz). The second is the sound caused by the voices of the guys. They made much more noise than the badminton or soccer players. This is visible in the curve.

The background noise is for convenience thought to be the same as for hall A.

16.4. Soccer hall B

A group of 6 soccer players produces the following sound:



The six man playing indoor soccer did not talk a lot, this is visible in the graph. No increase of sound is found in the graph for frequencies between 250 and 2500 Hz. An increase caused by the sound of the ball is found for frequencies around 100 Hz. Again, the background noise level is much lower than the average sound level of all measurements. This can be caused by a different setting of the ventilation system.

16.5. Conclusion

Conclusions of this tiny research are:

There are no clear specific frequency bands that are much louder than others. On the other hand, specific sports sound like bouncing basketballs, gives a slightly higher sound pressure in a small frequnecy band. The sound created by balls (bouncing, kicking etc.) consists of 63-125 Hz sound. The other main sounds are the sounds of human voices (250-2500 Hz) and beeping shoes (500-1000 Hz).

Over all, the found curve is an almost horizontal line. The background noise is a decreasing line. This means that the sporting people add more high frequent sound to the room than low frequent sound.

Because the sound of (e.g.) the ventilation system needs to be absorbed too, the design of a wall system needs to pay attention to all frequencies. Special attention (more sound absorption) can be given to the following frequency bands: 63-125 Hz, 250-2500 Hz and 500-1000 Hz. This depends, though, on the type of sport for which the hall is built.

I will focus on all frequency bands mentioned above for the design because the wall should be useable for all types of sports.

17. Design

This chapter gives a reflection on the design presented at the P4 presentation. It will give a description of the panel in section 17.2, choices are substantiated in section 17.3, positive and negative points are appointed in section 17.4. Finally, an advise is given to improve the design during a further research.

Since other designers would probably not come up with the same design as I did, the goal of this reflection is also to explain why this design is my design, why it fits me.

17.1. From roof to wall

During the graduation process, I investigated the acoustical behaviour of perforated steel roof panels. This research lead to different conclusions. Unfortunately, the conclusions gave no clear guidelines for a design. So, I took a step back to oversee the bigger problem of acoustics in sports halls. I concluded that, since the roof structure was working fine and improving it was not very challenging, the problem was mainly caused by the walls up to 3 meters. These walls are often made of hard and strong materials to rebound balls. The materials reflect a large part of the sound. This causes a high sound pressure level in the horizontal sound field and can cause flutter echoes. The reverberation time of ‘bad’ halls is mostly too long because of hard materials on the walls. Improvements can be made by designing a strong wall that absorbs sound.

Could I have made the decision to design a wall panel earlier in the process? This is a good question. Of course, the problem of the walls up to 3 meters was found early in the process. But, the broad demarcation of the subject was very important for the design. The gained knowledge on for example the effect of scattering of sound is very important in the design. It plays a major role in solving the problem. So, by a small detour, the research gave clear guidelines for the design. Without the gained knowledge, the design would not be as good substantiated as it is now.

This is why I decided to design a wall panel in stead of improving the roof structure.

17.2. Description of the design

The designed wall panel is intended for use in sports halls. The panel should cover the lower 3 meters of every wall, or at least one end wall and one long wall. The cassette is constructed as follows: two aluminium U-profiles are connected to the wooden facing from the back, a Rhinox rock wool thermal insulation panel is clamped in between, covered with a vapour barrier. The connection to the steel construction of the hall is not (yet) designed.

The wooden facing is constructed from wooden slats of 6mm thickness with different heights. These slats with a zig-zag pattern are glued together and form a strong facing of the cassette. The facing is perforated with holes of 2,5 mm in diameter in order to absorb sound.

Fig,1. *Impression of the designed wall panel*

17.3. Substantiation of choices

In this section, an explanation will be given why different choices are made. The requirements in the design are shown in figure 3. These requirements form the base of the design together with some starting points:
- A sports hall type B1 is used as 'default' (28 x 16 x 7 m);
- A sports hall is constructed from steel columns (centre-to centre distance: 4 m) and steel beams;
- The wall of a sports hall should be easy to build, like other industrial wall panels;
- Sports sound consists of mainly voices (250-2500 Hz), balls (63-125 Hz) and shoes (500-1000 Hz).

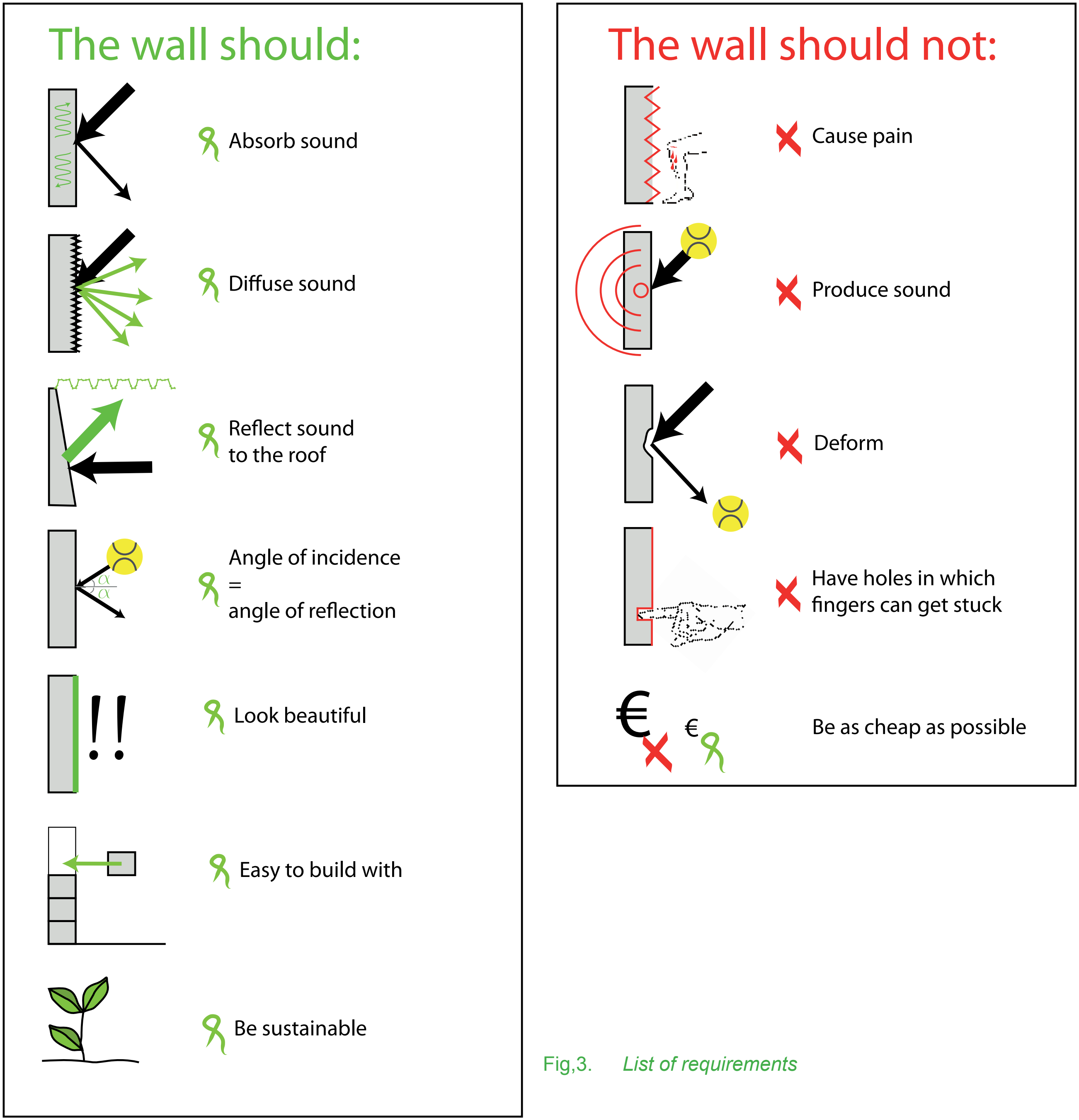
17.3.1. The material

The decision for wood is taken after a comparison of steel, wood and plastics/polymers. These three types of materials are compared on their looks, production methods, connection methods and material properties. Ceramics are not taken into account because of their nonindustrial character.

The most important substantiations are given below. From this list of properties, I found the warm and friendly look very important because architects often want such a look in a sports hall. A cold and hard look can be already created by using for example BMI concrete blocks. These blocks have an open structure and are good sound absorbers. Besides that, wood scores good for most properties. And, last but not least, most wood types are eco-friendly. The challenge in the choice for wood can be found in creating an 'industrial product' with it.

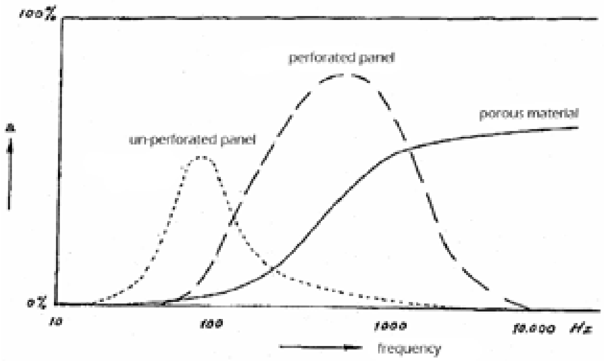
	Wood	**Steel**	**Plastic/Polymer**
Density: weight for 5 - 40 mm thickness	+	-	+
Warm, friendly look	++	--	+/-
Pain: can hurt (-), friendly/soft (+)	+	-	+/-
Industrial character: large spans	-	+	+
Strength for 5 mm thickness	+	+	+
Sound source: yes (-), no(+)	+	-	+/-
Thermal insulation properties	++	-	-
Costs: high (-), low (+)	+/-	+	+/-
Eco-friendly	++	+/-	-

Fig,2. *Comparison wood, steel and plastic/polymer*



Fig,3. *List of requirements*

17.3.2. Sound absorber



Fig,4. *Different mechanisms compared (Nederhof 2005)*

In this report, different types of sound absorbers are mentioned. The wall that needs to be designed should absorb sound, this is one of the most important properties it should have. The information of previous chapter reveals the different frequency bands on which the panel should work. These are: 63-125 Hz, 250-2500 Hz and 500-1000 Hz. Which is not very easy to achieve with one panel. Out of the three basic sound absorbers: porous material, panel absorber and a perforated panel absorber, the last one is chosen. A porous material is not strong enough to rebound balls. A plate resonator is a less effective sound absorber than a perforated steel panel.

By designing a panel with different thicknesses of the facing, a broader absorption peak can be created. In this way it is possible to design a panel that is a good sound absorber for most sport-sound-frequencies.

17.3.3. The shape of the facing

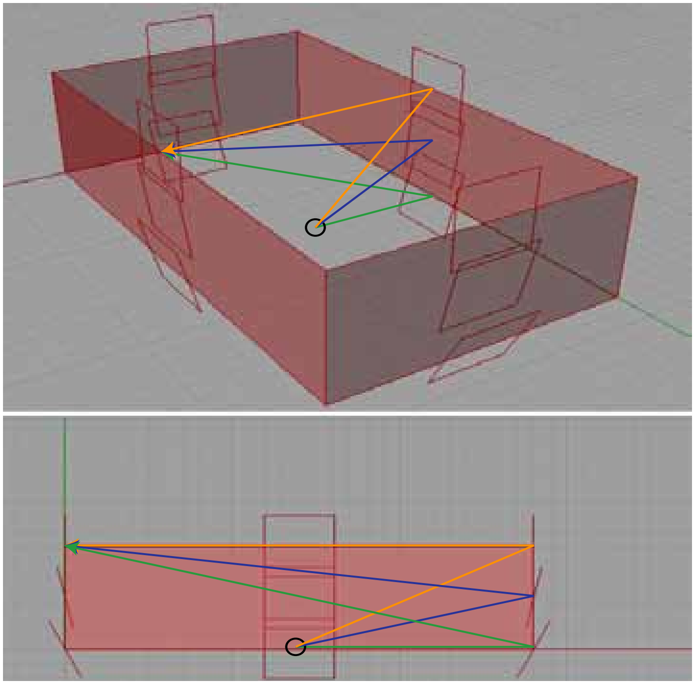
The shape of the panel is of course very important. The requirements that contribute to the result are: 'it should diffuse the sound', 'it preferably projects the sound to the roof', 'people may not be in pain after running into it', 'fingers may not stick in it' and 'balls should be reflected in an angle according to the expectations'.

Base shape

Figure 6 and 7 show the base shape, the first models. Half of the incident sound is directly reflected to the roof, the other half will reach the roof after a reflection at the floor surface. The sound absorption of the panel is realized by the perforations. As visible in the figures, the perforations of 1 mm are not visible and will not work because of the dust in the holes. The perforations should therefore be larger than 1mm diameter.

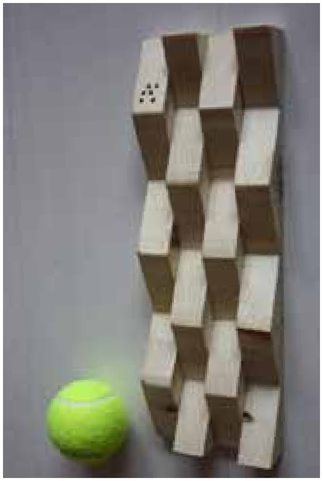
Angle of oblique surfaces

The minimal necessary angle of the oblique surface that reflects the sound to the roof is defined by the size of the hall. As shown in figure 5, the angle decreases when closer to the roof. The worst case scenario (sound source on the ground) gives an angle of 23.6 degrees for sports hall type B1.



Fig,5. *Angle of oblique faces*

Fig,6. *Basic shape side*



Fig,7. *Basic shape top*

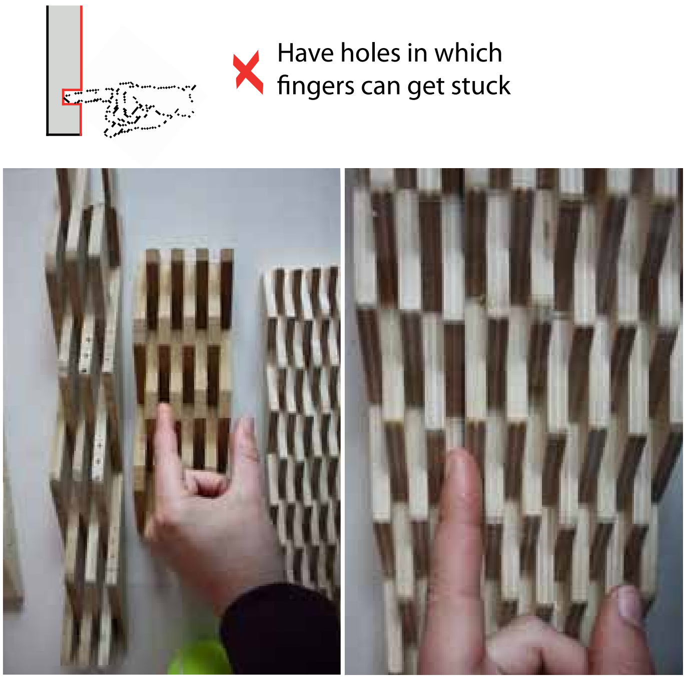
Fig,8. *Perfo 1 mm*

Fig,9. *Perfo: 3 mm*

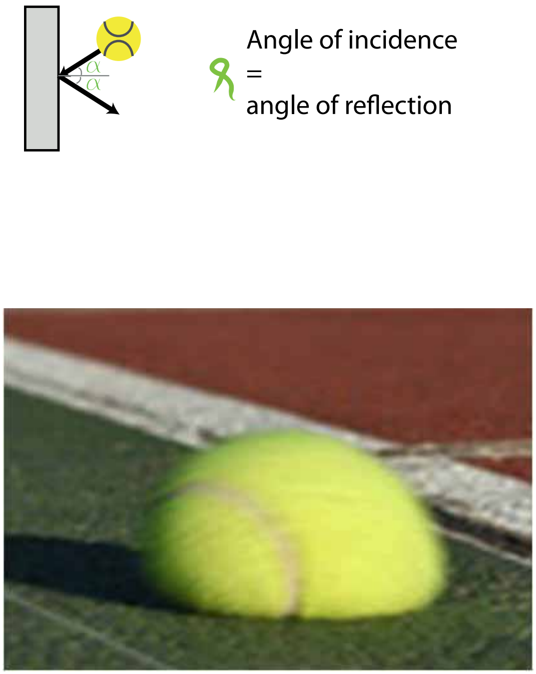
Size of the elements

The sizes of the wall panel are chosen in a way that: no fingers can get stuck (figure 10), a reflection of a tennis ball is not bothered by the shape of the panel (figure 11) and that the panel absorbs sound of a useful frequency area. Next section will explain more about the perforations that cause the sound absorbing properties.

When a (tennis) ball bounces, it dents when it touches the floor. The higher the speed, the more it dents. See also figure 11. The surface area of the dent determines whether the ball will reflect in a ‘normal’ way, or in a bothered way (when the normal direction of the surface seems not perpendicular to the surface).



Fig,10. *Can fingers get stuck?*

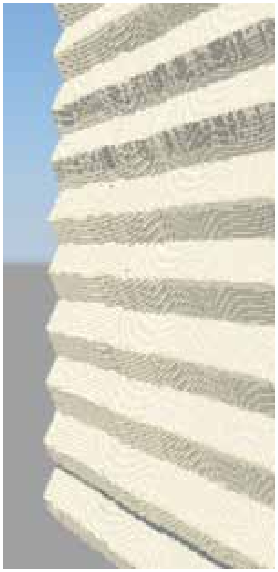


Fig,11. *Deformation of tennis ball.*
Src: http://deansomerset.com/2011/11/28/

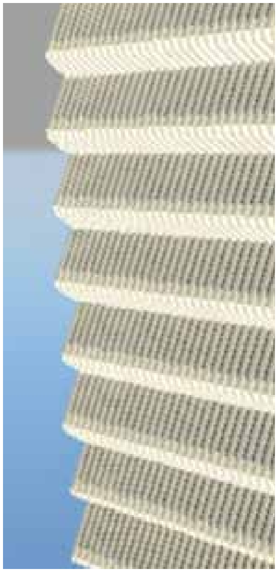
Zig-zag pattern

The pattern of the zig-zag shape of the wooden parts have influence on the bounce-behaviour of the panel, the roughness of the panel and on the sound diffusion properties of the panel. All three mentioned factors are very important.

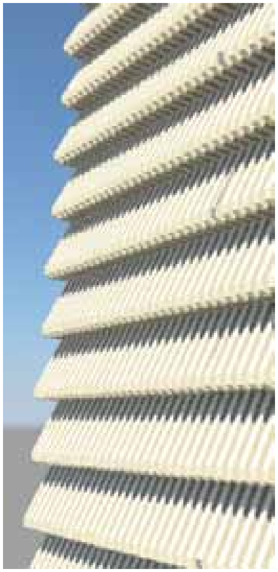
Besides the symmetrical zig-zag pattern, other patterns are investigated too. The other patterns shown below will reflect a bigger part of the sound to the roof. Pattern B and C have better rebound properties than pattern A. The best option is pattern B, where most incident sound is reflected to the roof. Pattern C has the problem that reflections are reflected downwards by its own shape. This is not useful.



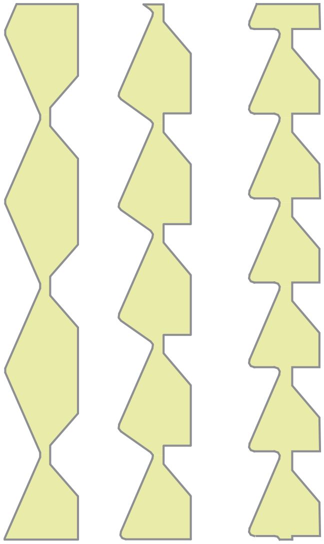
Fig,12. *Pattern A*



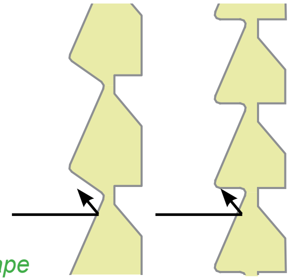
Fig,13. *Pattern B*



Fig,14. *Pattern C*



Fig,15. *Pattern A, B and C*



Fig,16. *Reflected by the shape*

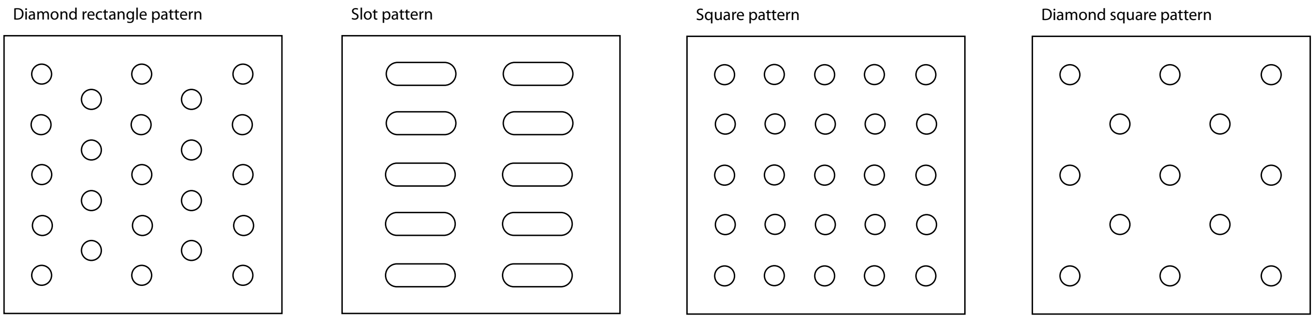
17.3.4. Perforations

Different types

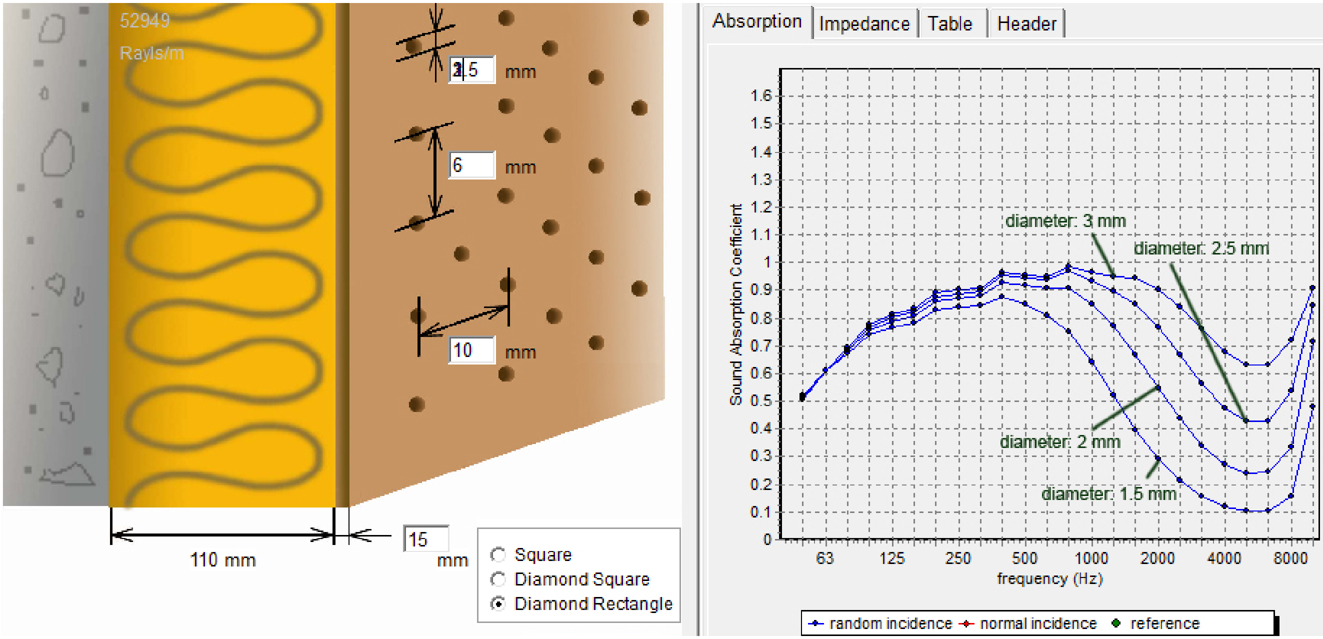
To make the designed wall a good sound absorber, perforations need to be added. There are different types of perforations: slot perforations and round perforations. The last one can be divided into: square pattern, diamond square pattern and a diamond rectangle pattern, see figure 17.

The width of the wooden strips should be maximal 6-8 mm. In this way no fingers can get stuck in it. This means that there is a 6-8 mm distance between two rows of perforations. The more perforations, the better the sound absorbing properties, so 6 mm would be better from this point of view.

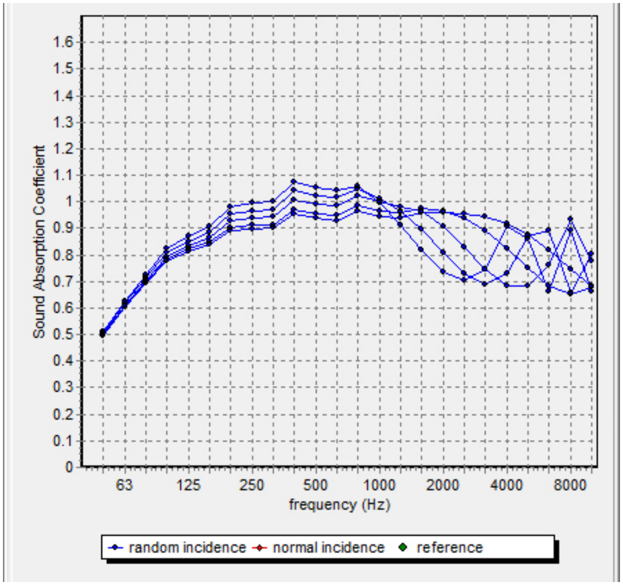
The size of the perforations and therefore the perforation degree, is important for the sound absorbing properties of the panel. Different diameters of the holes give different results. This is shown in figure 18.



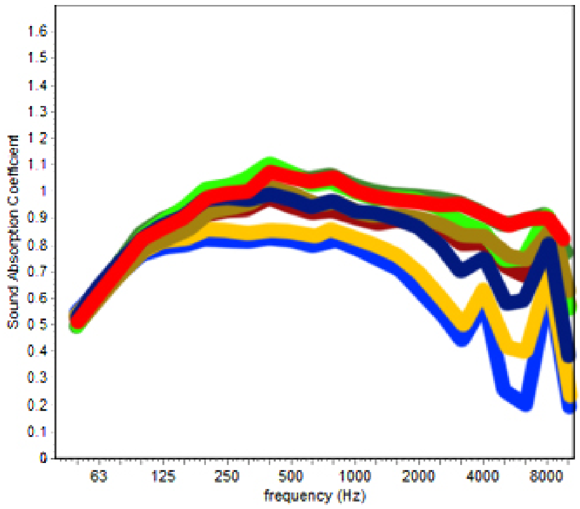
Fig,17. *Perforation types*



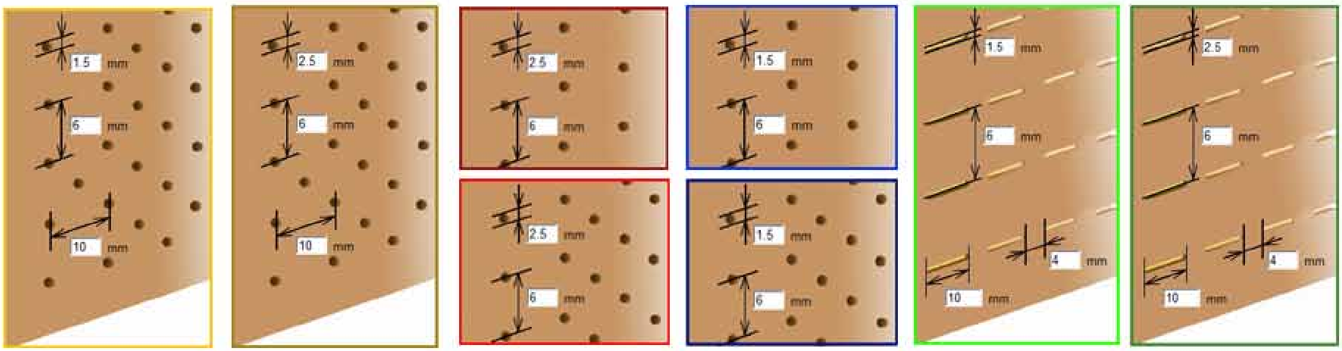
Fig,18. *Influence of perforation size on sound absorption*



Fig,19. *Sound absorption per thickness of facing / height.*



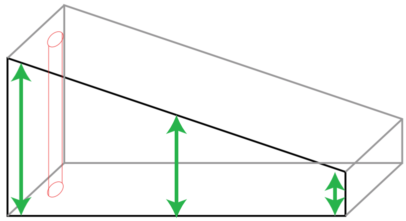
Fig,20. *Combination of different heigths, different types of perforations compared..*



Fig,21. *Types of perforations, the key of figure 5.*

Comparison of different types

The different types of perforations are compared in figure 20. Because the wooden strips have different heights (see figure 22), the 'thickness of the facing' differs too. A perforated panel normally has one peak frequency for which it absorbs most of the sound. By having different thicknesses, different peak frequencies come together in one element. In this way it is possible to combine the qualities. The peak will become much broader than for perforated facings with a constant thickness. Figure 19 shows the sound absorption coefficient for different thicknesses (5, 10, 20, 30 and 40 mm). The thicker the facing, the lower the peak-frequency. All curves combined are summarized by the red line in figure 20.



Fig,22. *Different heights*

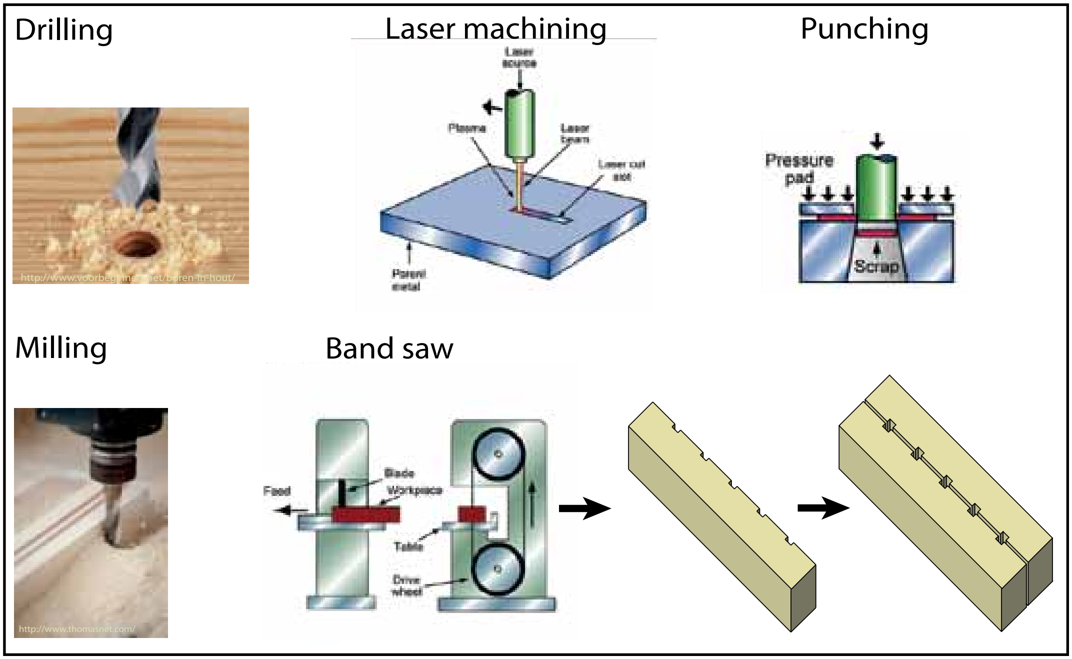
The maximal thickness of the facing is chosen to be 40 mm. This is a balance between sound absorbing properties, weight of the facing and material use.

Slot absorbers

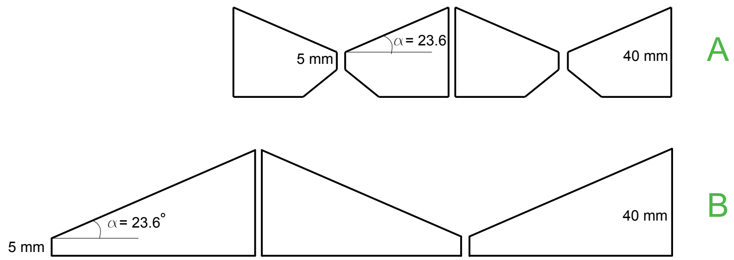
The best results are found for the slot absorbers. This panel absorbs most of the sound because of the open facing. For the design it is not a very realistic option though, because a slot has a certain length and because of this length, the height can never be 40 mm for the whole slot. This 40 mm height is important for the absorption of low frequent sound. So, with the shape of the wood shown in figure 22, a slot absorber is not the best choice.

Best option

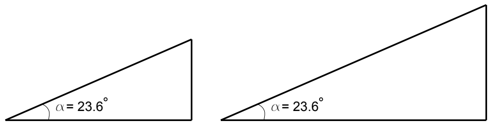
The best option then is the red line, the diamond square perforations with a diameter of 2.5 mm and a centre to centre distance of 6 mm (which is, of course the thickness of the wood).



Fig,23. *Methods of creating perforations in wood*



Fig,24. *Tapered perforations: solution for design.*



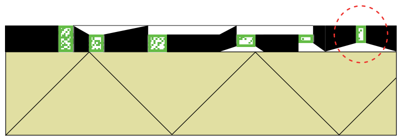
Fig,25. *Set angle gives set length and width*

17.3.5. Creating perforations in wood

How to create perforations in wood? Round perforations can be created by drilling. Free form shaped perforations can be made with a laser cutter. Punching is also a method which can be used but it works only for thin layers of wood. Square or rectangular shaped perforations can be formed by the blade of a saw. The width of it will create a small slot in the wood. Combining two of those elements gives a perforation. See also figure 23. The squared perforations give the panel a modern look. Besides that, round perforations have a smaller surface area than squared perforations with the same diameter. The bigger the opening, the better the absorbing behaviour.

Vertical section

The section of the perforation is also important for its working. The Helmholtz resonator just works when the air in the perforation is a rigid block of air. This means that the inner sides of the tube should be as smooth as possible. Figure 25 shows different sections. The green block represents the block of air.



Fig,26. *Differently shaped sections*

Link to design

According to the article 'On Helmholtz resonators with tapered necks' by S.K. Tang (2005) has a tapered shape of the inner side of the perforation a positive effect on the sound absorbing properties. The tapered shape is shown in the red circle in figure 25. The tapered shape can improve the design in two ways: by better sound absorption and by decreasing the thickness of the facing.

Since the angle of the oblique side of the block is set, the height and length are set with it. This is demonstrated in figure 26. This means that if the wanted 40 mm height is used in the design (good low frequent sound absorption), the profile length is 80 mm if the thinnest part of the facing is 5 mm. The big difference in height is necessary for the best sound absorption but causes problems on the rebound angles. Balls will suffer from the rough profile. The angle of reflection will depend on where the ball hits the surface. This is an unacceptable situation. By tapering the inner sides (A), the 5 mm thin facing can be created and the profile is much less rough. The rebound qualities will be much better than for situation B, shown in figure 24. Besides that, the sound absorbing properties increase.

17.3.6. Type of wood

Different types of wood are investigated. Accoya wood is chosen to be used in the design.

	Oak	Birch	Spruce	Plywood	MDF	OSB	Accoya	Teak
Looks	+	+	+/-	+/-	-	+	+	+
Hardness crosscut (Janka) [N]	3500	1650	7400	4880	2980	260	6600	1550
Shredding: yes (-), no (+)	+	+/-	+	-	-	+/-	+	+/-
Thickness 3mm available: yes (+), no (-)	-	-	-	+	+	-	-	-
Eco-friendly (embodied energy) [kcal/lb]	820	860	820	1626	1193	1267	600	820
Density [kg/m^3]	969	692	498	500	720	650	510	623
Differential shrinkage: radial, tangential [%]	0.3, 0.5	0.2, 0.3	0.2, 0.3	0.1, 0.2	0.1, 0.2	0.1, 0.2	0.1, 0.2	0.1, 0.3
Price [USD/lb]	1.1	0.5	0.5	0.6	0.3	0.5	2.0	4.1

Fig,27. *Comparison wood. SRC: CES Edupack 2011, www.accoya.com, www.greenspec.co.uk/embodied-energy.php, www.woodgears.ca/hardness_test/index.html*

The Accoya wood scores best in this comparison. Especially the eco-friendly property, high strength and relatively low costs are important factors for the decision.

17.3.7. Backing material

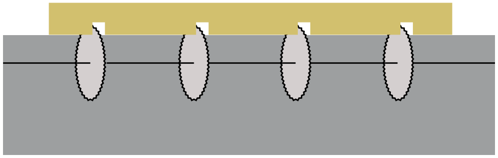
A Helmholtz resonator gives a broader absorption peak when the cavity is filled with a porous material. Also, a porous material has positive effects on the thermal insulation of the building skin. And, as I learned from the laboratory results, rock wool panels have good (low frequent) sound absorbing properties. This is why Rhinox rock wool panels are used in the design.

17.3.8. The size of the cassette

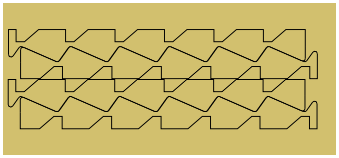
The size of the cassette is defined by the weight one persons may lift; this is maximal 25 kg (Arbowet 2010). The maximal size of the cassette is therefore 0,8 x 1m because a rockwool panel weights 17 kg/m^2, the wooden facing 8.3 kg/m^2. The aluminium frame weights 2.1 kg/m.

17.3.9. Production process

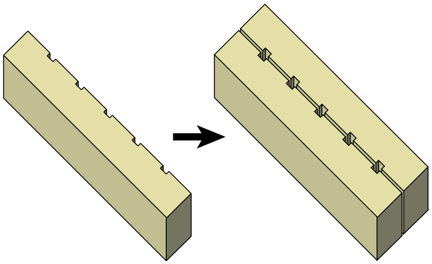
The production of the wooden facing is the most complicated part of the production process of the total cassette. First, 3 mm thick Accoya wood is moved over a row of circular saws with a centre-to-centre distance of 6 mm. In this way, half perforations are created. See figure 28. Next, the zig-zag shape will be punched out of the wood. See figure 29. Finally, two (mirrored) thin slats form a strip of 6 mm thick with a row of perforations in the middle. All rows need to be glued together to finish the facing. This can be done by people or by a computer-controlled machine.



Fig,28. *Creation of 'half perforations'.*



Fig,29. *Creation of slats by punching them out of a 3 mm thick plate.*



Fig,30. *Twice half is whole, the perforation is created.*

17.4. Plus and minus points

An important part of this reflection on my design is the list of plus and minus points. What are the strong parts of the design? What are the weaker points?

Positive

- The look of the panel is good, beautiful. The wanted soft, friendly character is achieved.
- If the perforations stay open, the facing can be painted in any colour. This kind of architectural freedom is not possible with porous materials like concrete BMI stones.
- The sound absorbing properties of the designed panel are good. The special effect on the sound absorbing properties by different thicknesses is innovative. This combination of diffusion and sound absorption is simple but effective. It fits my idea of the 'perfect' design. The solution should be simple but effective, this fits my way of designing.
- The diffusion of the sound and projection to the roof by the shape of the panel works in any case for high frequent sound. The short wavelength of these frequencies fits in the size/scale of the zig-zag pattern. To what extend longer sound waves; lower frequencies also get reflected to the roof is the question.
- The reflection of incident sound is directed to the roof where the angle of incidence and the angle of reflection of balls are the same. The rebound qualities are not bothered by the texture of the facing.
- The cassette shape of the panel makes it easy to construct a wall. Although the connection to the construction is not yet designed, placing the cassettes will not take a long time.
- The character of the design is eco-friendly and sustainable because of the choice for Accoya wood.
- The thick facing combined with the Rhinox rock wool panels, which are placed directly behind, ensure that the panel does not act like a sound source when hit by a ball.
- Fingers of adults cannot get stuck in the facing.
- Because of the large strength of Accoya wood, the facing does not deform easily due to sports activities.
- The panel will improve acoustics in sports halls. Thereby, the amount of complaints from users will decrease.

Negative

- The designed zig-zag pattern diffuses high frequent sound, but probably not low frequent sound. Because of the large dimensions of these waves, the small zig-zag pattern might not be big enough to reflect the sound to the roof. The waves simply do not notice the texture of the facing.

- Although already better than concrete stones, the designed facing is still a bit sharp. Nobody should get hurt when running into the wall. The square edges should get rounded.
- The production costs of the panel are high because the connection of all slats is a time consuming activity.
- The connection of the cassette to the steel construction of the hall is not yet designed.
- Fingers of children might fit in between the 6 mm thick slats of the facing.

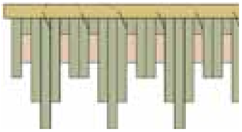
17.5. Advice for further research

If somebody continues designing this wall panel, I recommend to investigate the following topics:

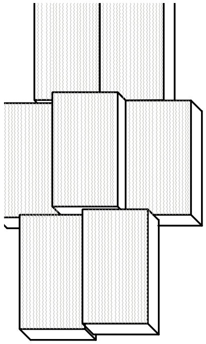
17.5.1. Diffusion of low frequent sound

Wether the texture diffuses low frequent sound or not, should be known before production starts. A measurement on reflection coefficient, see chapter 11, could give information on the reflection of sound with different wavelengths. If the outcome shows that the zig-zag pattern is to small for low frequent sound to get reflected in a certain direction, another solution for this type of sound should be found.

Possible solutions for diffusion of low frequent sound:
A: 'Schroeder diffuser', vary the placing of the panels in such a way that a second, larger, texture arises. This option works, but has a disadvantage: it bothers the rebound qualities of the wall.

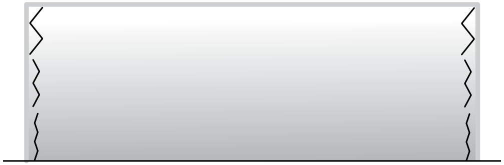


Fig,31. *Schroeder diffuser*



Fig,32. *Solution A: Schroeder diffuser*

B: Increase the size of the zig-zag pattern for every 2 meters height in the hall. Few balls get higher than the lower 3 meters of a wall. In this way, large sound waves get diffused by the larger zig-zag pattern without bothering the reflections of balls.



Fig,33. *Solution B: larger zig-zag pattern*

17.5.2. Less painful, rounded edges

The edges of the wood are a little bit sharp. The final design should have rounded edges. Different options to achieve this are:

- Dust blasting/soda blasting. Like sand blasting but than less rough and strong.
- Milling. The separate slats of the facing could be milled. Milling gives options to rounded edges in every wanted way.
- Sanding. This is a very time consuming task, but it is an option.

Of the three options mentioned above, the first one is most likely because of the costs. The other two are very time consuming and therefore expensive.

17.5.3. Connection method

Next, the connection method should be designed. How are the cassettes connected to the main construction of the hall? Is it necessary to replace panels after being built?

17.5.4. Costs

Since the panel seems to cost more than wanted, cheaper options to create the panel should be investigated. It may be cheaper to create the shape of the facing in plastic, by moulding techniques. In this way, larger panels can be created. This improves the ‘industrial character’ of the design. Although, the wooden look is very attractive, so the look of the plastic version should be an issue.

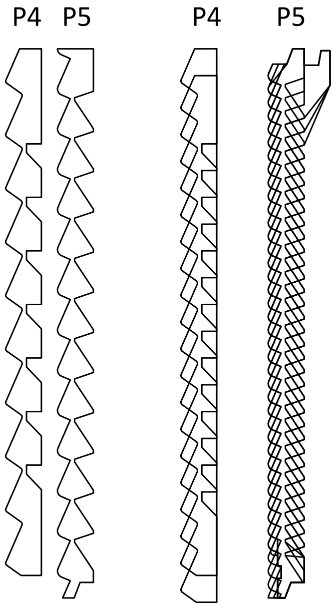
18. The final design

The final design as presented at the P5 presentation is described in this las chapter. Some improvements are made to the design in the last phase of the graduation process. These improvements are described in this chapter.

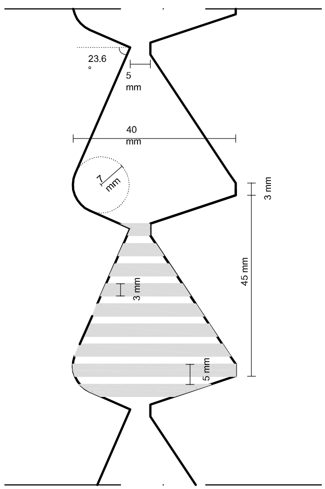
18.1. Less painful

The zig-zag pattern of the wooden facing is slightly changed to make the wall less 'painful'. The pattern has a smaller centre-to-centre distance, so the 'peaks' are closer to each other. Besides that, the 'peaks' are rounded with a diameter of 14 mm. Finally, the sequence of the strips is not made out of 2 any more, but out of 3. In this way, the peaks get closer to each other and the panel has less the shape of a rasp. Figure 1 shows the P4 (before) and the P5 pattern (after).

The changes made for the final design improve the situation a lot. Nevertheless, the aspect of safety is very important. Further research on possibilities to improve the shape on safety aspects, is encouraged.



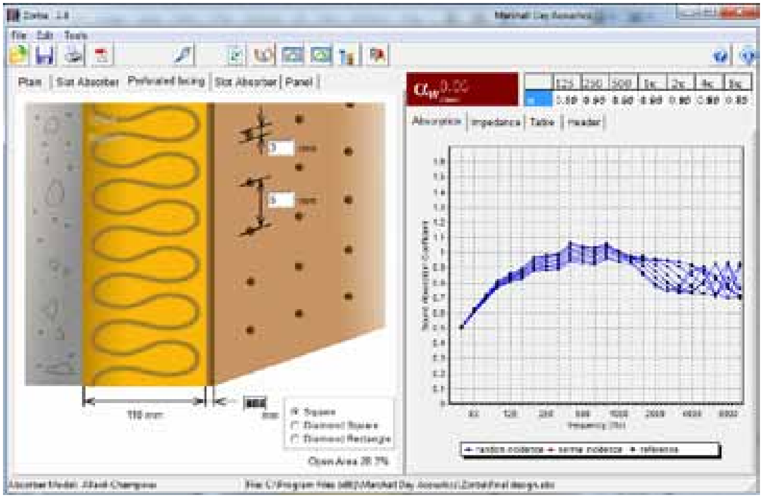
Fig,1. *Zig-zag pattern changes P4 - P5*



Fig,2. *Drawing of one element of the zig-zag shaped wooden strips.*

18.2. Perforations

The model made for the P4 presentation showed that the perforations could become bigger without reducing the strength of the panel. The more and larger the holes, the better the sound absorbing behaviour. The final size of the perforations is 3 x 3 mm. The pattern of perforations is squared, this has been changed to have the same perforated strips for all 3 strips of one sequence. Now, all elements are the same (except for the endings of course). The sound absorbing behaviour of the final design is shown in figure 3.



Fig,3. *Sound absorbing behaviour of designed sports wall.*

18.3. From cassette to panel

The original idea was to design a cassette, one industrial element. The problem I found was the size of these elements. The weight of the wooden facing (8.3 kg/m^2) and the Rhinox rockwool panels (17 kg/m^2) caused a small cassette. The combination of all parts in one element was a good idea but it led to heavy cassettes. One person may lift max. 25 kg. The cassette would become very heavy and smaller (0,8 x 1m) than the initial idea of large industrial cassettes. In short, the idea of a cassette did not fit the requirement: 'easy to build with'. In the P5 design, the wooden facing and rock wool panels are combined during the construction of the sports hall. In this way, the panels are easy to lift by one person (6 kg) and are 'easy to build with'.

Besides the weight, the final design has the advantage of easily replaceable panels as well.

18.4. Thermal properties

The R-value of a facade should be minimal 4,2 W/m^{2}K. This is what the standard (Bouwbesluit 2012) says. In the future, this value will probably go to 5 W/m^{2}K.

The final design could be used very well in the future. When the layer of rockwool (with the wooden facing) of the inner facade is carried forward up to the roof, the R-value of the whole facade would become 5,9 W/m^{2}K. In the design as given in figure 4, the inner facade stops after three meters because nowadays most of the designers would prefer less insulation (= lower costs).

18.5. Use in existing sports halls

Most problems of bad acoustics in sports halls are found after the construction of the hall. Therefore , the renovation aspect of the designed wall panels is important too. Figure 9 shows a way to use the wooden facing with rockwool backing on an existing brick wall.

18.6. Costs

The costs aspect of a design is always important. The designed wall panel (1200 x 600 mm) costs about 322 Euro/m^2 . This includes the perforated steel inner boxes, the thermal insulation and the wooden facing. The costs are built up out of different elements: material, labor, plant and subcontract.

Material (medium influence)

- Accoya wood,
  ≈ 3.5 euro / kg ≈ 21 euro / panel (source: www.Accoya.com 2012)
- Wood glue to connect the wooden strips,
  ≈ 0.70 euro / panel (source: http://www.dhzdiscount.nl 2012)
- Trespa screws to connect the wooden facing to the inner boxes,
  ≈ 0.13 euro / piece ≈ 0.65 euro / panel (source: http://www.deijzerwarenshop.nl 2012)
- SAB perforated steel inner boxes,

≈ 35 euro / m^2 ≈ 25 euro / panel (source: http://www.bouwkosten-online.nl 2012)

- Nails (shoot) to connect the boxes to the steel columns,

≈ 150 euro / 1000 pieces ≈ 1 euro / panel (source: http://www.deijzerwarenshop.nl 2012)

- Rhinox rockwool insulation panels 110 mm.

≈ 25 euro / m^2 ≈ 18 euro / panel (source: http://www.deschachtplastics.be 2012)

Labor (medium influence)

- Connect the inner boxes to the steel columns,
- Push the rock wool panels in the inner boxes,
- Connect the wooden facing to the inner boxes with 5 screws per panel,

Estimation: 30 euro/m^2.

Plant (small influence)

- Small platform to reach up to 3 meters to connect the inner boxes and the facing,

≈ 50 euro / day (116 panels/week) ≈ 2.20 euro / panel (source: http://www.bouwkosten-online.nl 2012)

- Tools to connect the inner boxes and wooden facing, estimation: ≈ 0.2 euro / panel

Production of panel (large influence)

- A factory where the wooden facings are made.

It is a time consuming business to create the wooden facings. Estimation: 200 euro/m^2

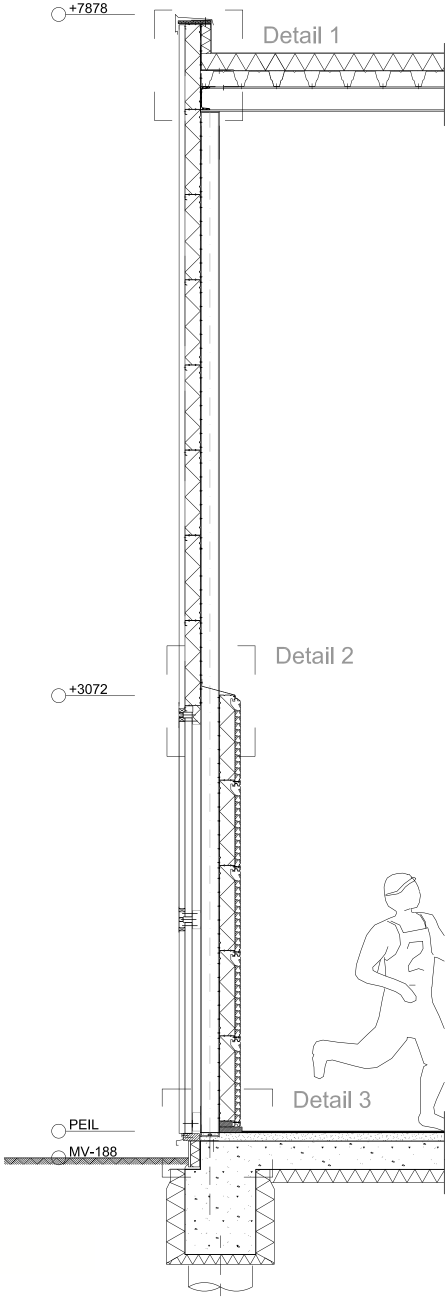
Material:	66 euro / panel	92 euro / m^2
Labour:	21 euro / panel	30 euro / m^2
Plant:	2.40 euro / panel	3.30 euro / m^2
Production of panel:	144 euro / panel	200 euro / m^2
Total:	**≈ 233 euro / panel***	**≈ 325 euro / m^2***

Concrete stone walls cost about 30 euro / m^2; (source: http://www.bouwkosten-online.nl 2012).

* These values are estimations.

18.7. Drawings

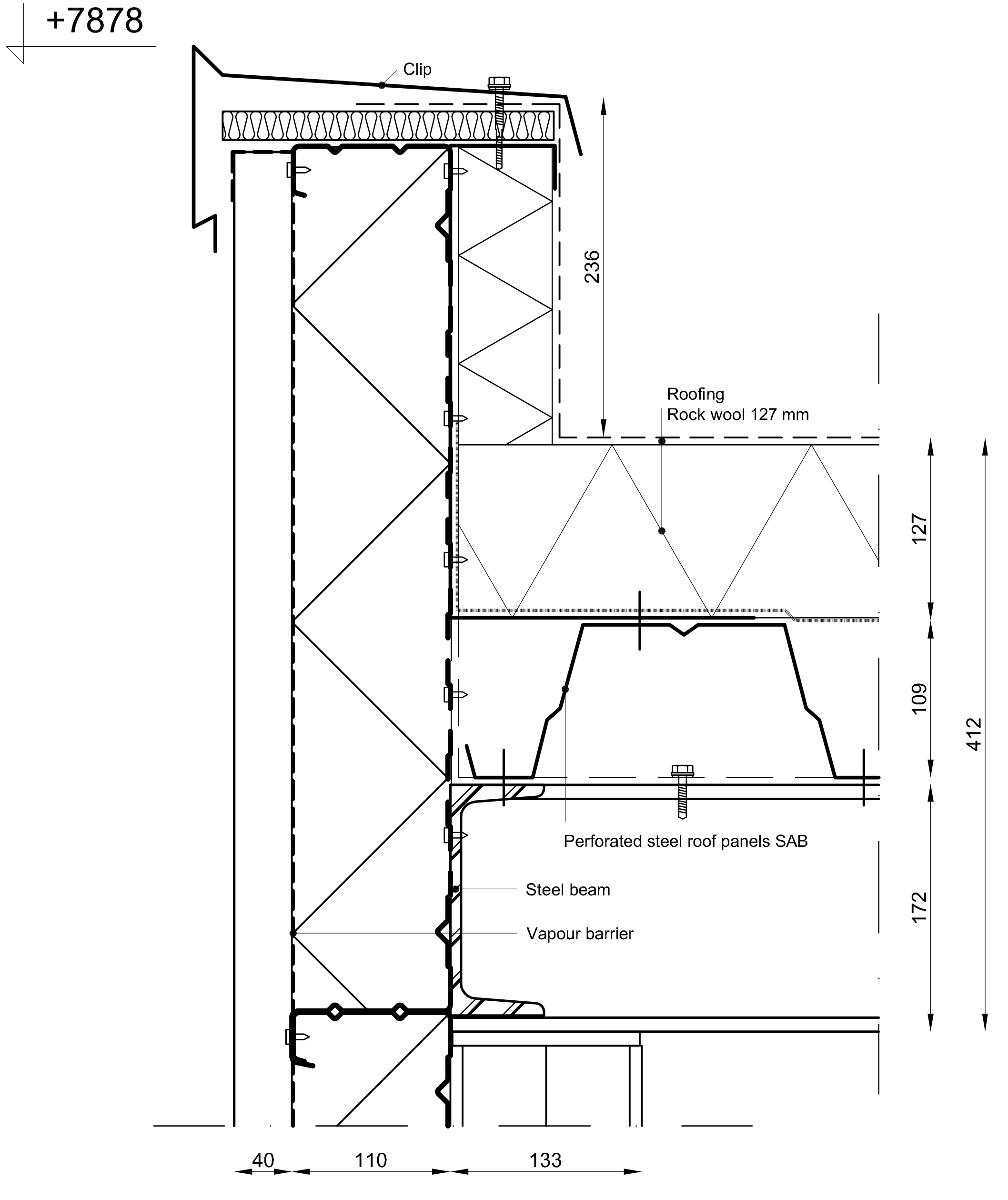
Drawings and impressions of the design are shown on the next pages. More drawings can be found on the presentation posters.



Fig,4. *Section with details*
Scale 1:50

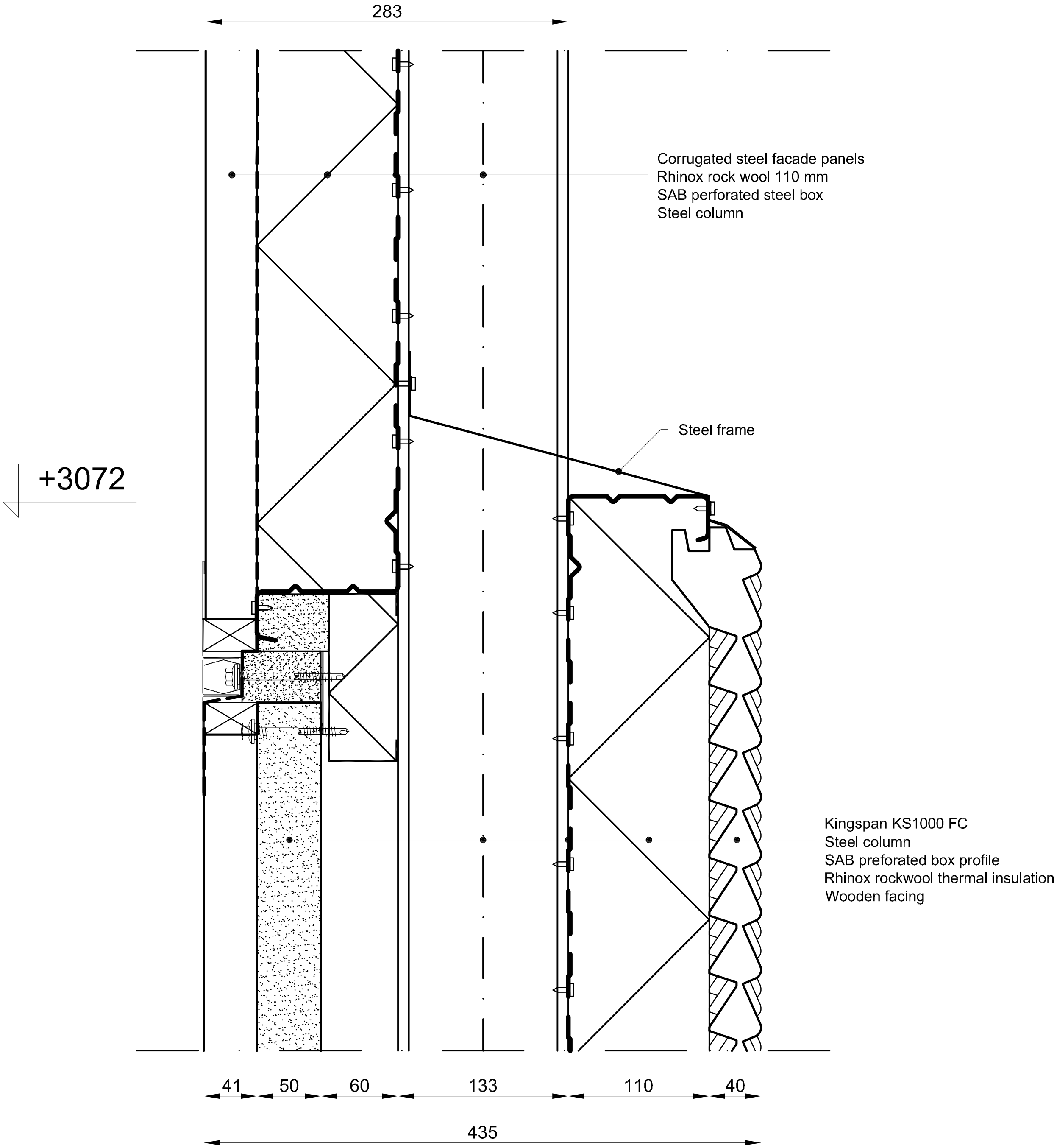
Fig,5. *Detail 1*

Scale 1:5

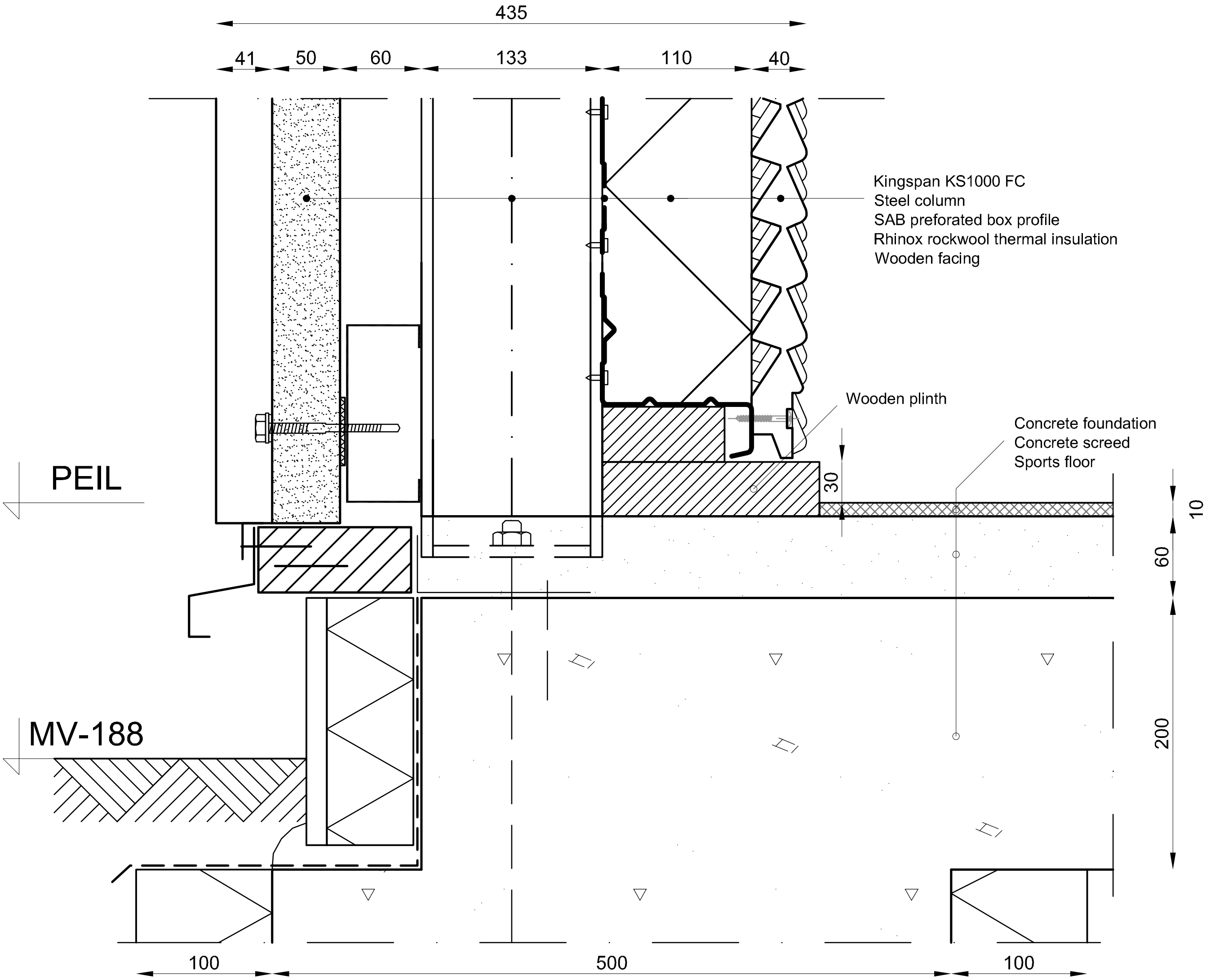


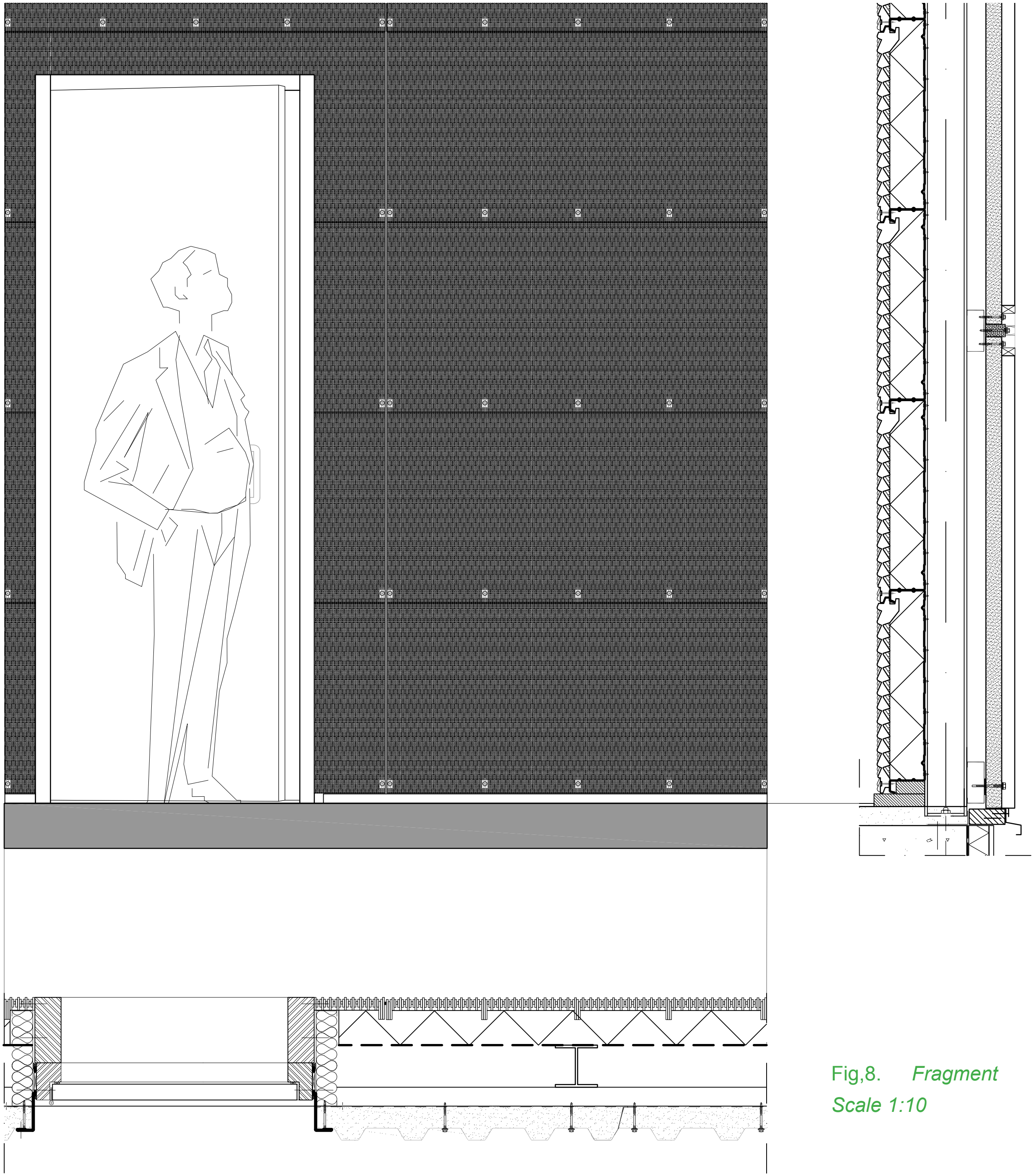
Fig,6. *Detail 2*

Scale 1:5



Fig,7. *Detail 3*
Scale 1:5



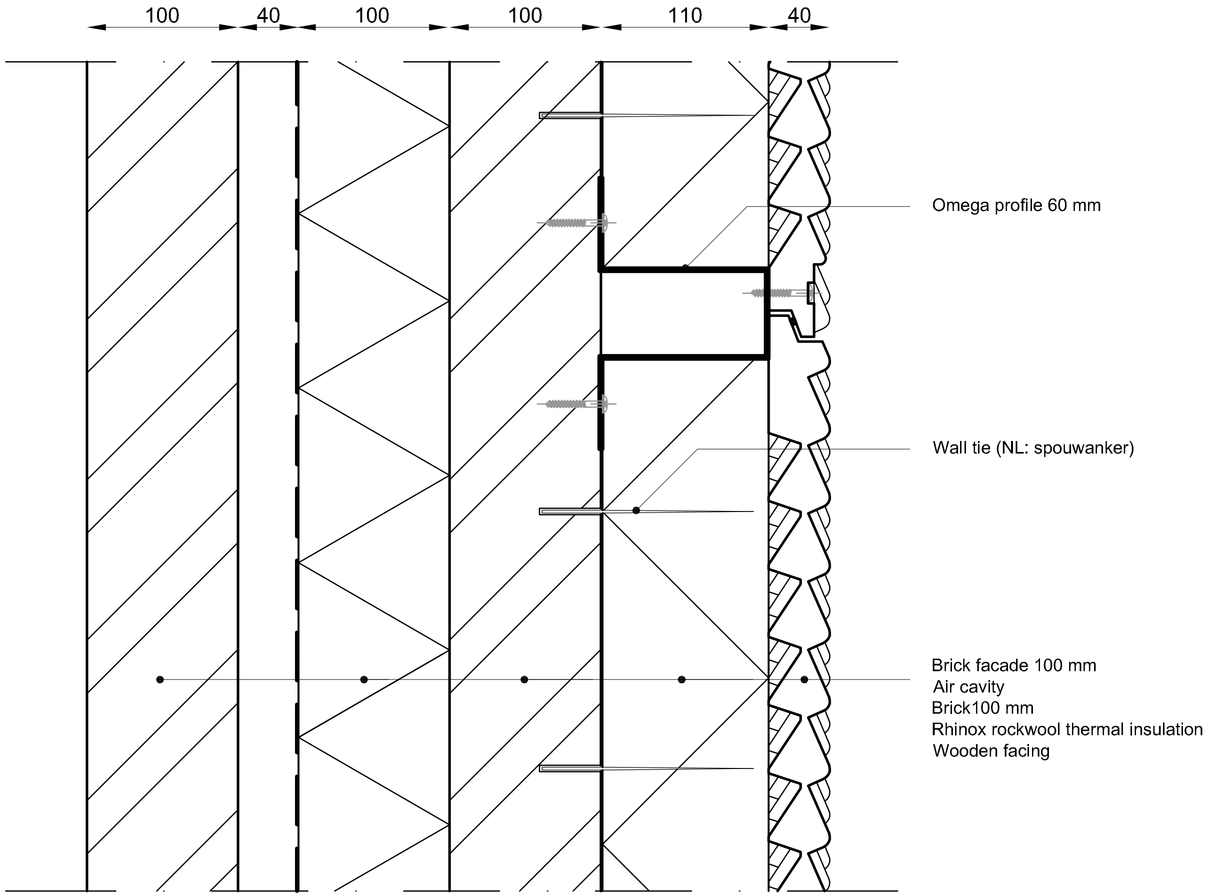


Fig,8. *Fragment*

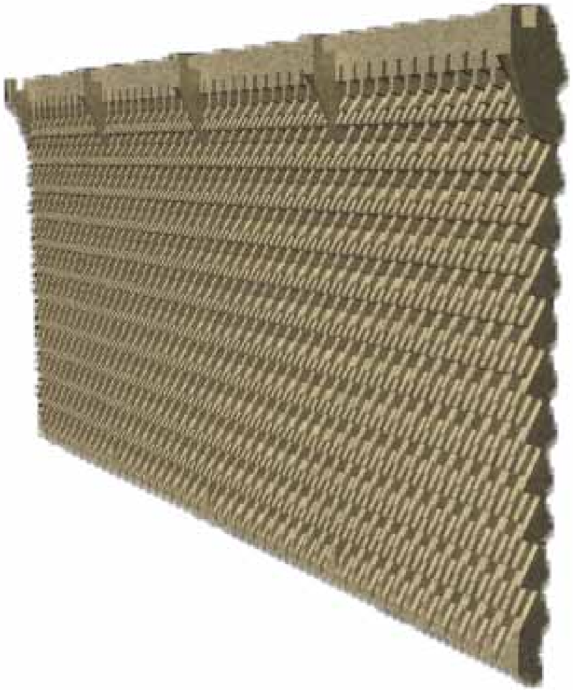
Scale 1:10

Fig,9. *Renovation detail*

Scale 1:5



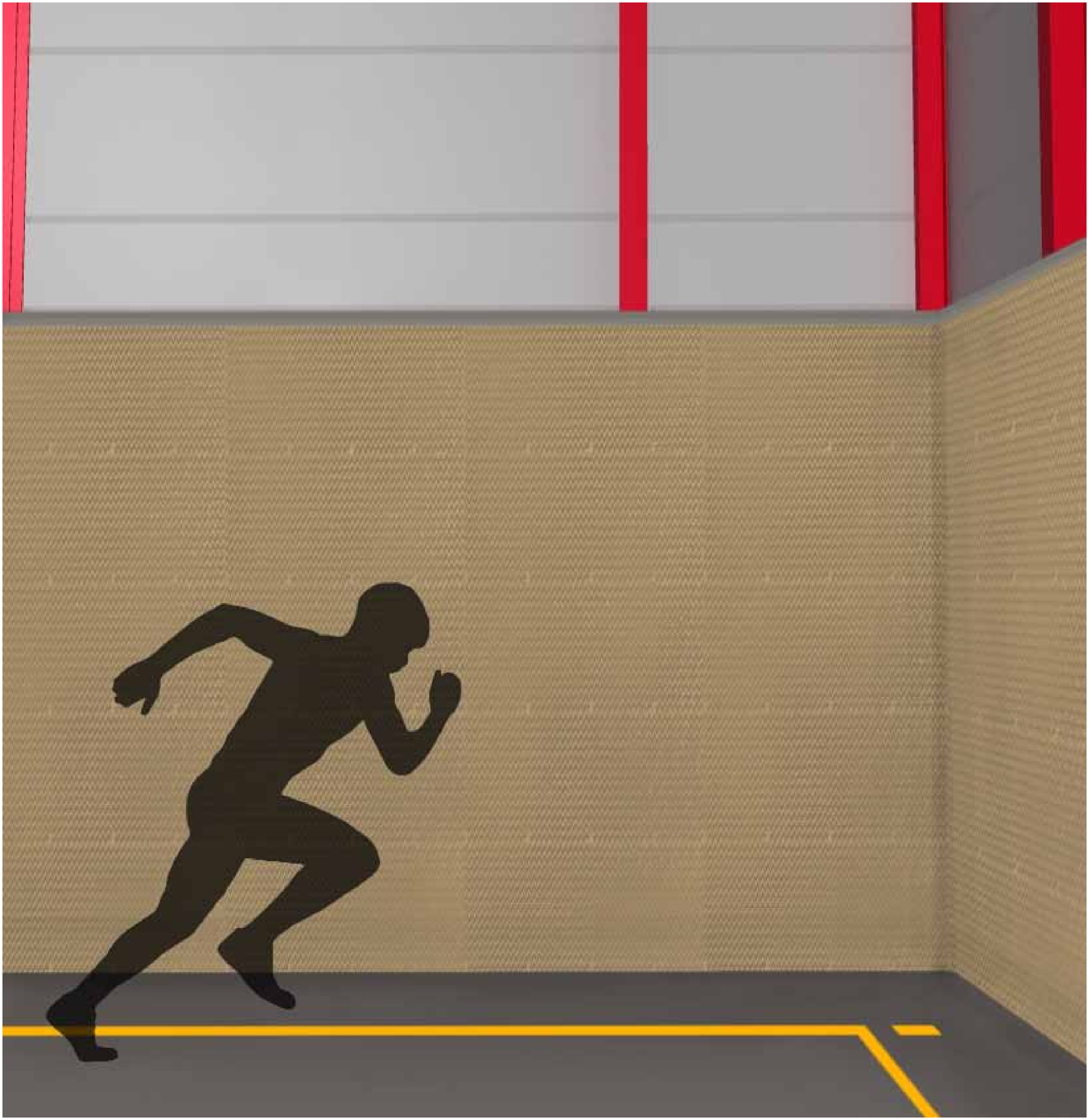
Fig,10. *Impression of panel: front*



Fig,11. *Impression of panel: back*

Fig,12. *Impression of hall with panel*
Source photo: LBP|SIGHT

Fig,13. *Impression of wall*



Reflection

This reflection is written at the end of my graduation project, before the P5 presentation at June 29th, 2012. It reflects on the relationship between research and design, studio versus student, and on the product, process and planning. Feedback will be given on the ‘how’ and ‘why’ of the study plan. Did my approach work or not, and to what extent?

Relationship between research and design

The relationship between the design and the research is the bigger problem: the bad acoustics in sports halls. Both of them aim for a solution of the problem.

Since the research did not result in a clear guideline for a design, I took a step back to oversee the bigger problem again. This resulted in the conclusion that I needed to design an innovative wall panel that could improve the acoustics in a sport hall. The gained knowledge on the working of the perforated panel was very useful to find a good solution and to make a good design.

Studio versus student

When I started my graduation project, I clearly had an approach in mind. I wanted to do a small literature study first to demarcate the problem, then I wanted to do proper research (preferably something for which I could use my creativity) and out of the conclusions, a clear guidance to the design would follow. This is how I saw my process.

The line of the studio though, was slightly different:

- I had to do a broader research on the problem than I thought, which ended up to be useful.
- Martin helped me to get the most out of the very though physics part. I think I would have given up the fight

much sooner without his support. Although it was the most though part, it satisfies me right now.

Most of my ideas of my graduation fitted the frame of the studio, so that was very nice. I found it also very useful that I got the time and space to do what I wanted and that I could show initiative on things.

In short: I chose the good studio, it fitted me perfectly.

Reflection on product, process and planning

When I look back on the last 25 weeks, I am pretty proud of myself. My approach almost exactly fitted the planning. Although, there where some setbacks: the communication with LBP|SIGHT about the laboratory measurements did not go so well. This resulted in a late start of my measurements in the laboratory and some frustration on my side because I couldn't follow my planning because of other parties. It ended up to be not a big problem; I made a the wide planning.

Also, I noticed that the analysis part of the research is not my strongest part. I took me much more time than expected to write good, reliable and proper conclusions of the research I did. So, the delay of the laboratory measurements where used to finish the scale model measurements part.

Finally, I have to admit that I found the physics part of the project very difficult. At one point I thought I was studying physics in stead of architecture. But all in all, I worked according to the plan and it worked out fine for me.

Than something about the product. The design of a wall panel was a bit of a surprise for me. I knew beforehand that the design would follow from the results of the measurements but I did not expect to find an answer to the main question without a strong guidance to the design. To let the design follow from the results was a bit of a gambling but I already looked for a solution for what to do when no answer was found. The 'backup' plan was than to design an as good as possible acoustics for a sports hall. During the process though, I found out that the main issue of the bad acoustics were the walls up to 3 meters. This problem got my interest and I have used the knowledge gained from the measurements to design a wall for a sports hall.

Was it possible to decide to make a design for a wall panel earlier in the process in stead of investigating the roof structure so roughly? The answer to this question is yes. The decision could have made earlier in the process. The throurough research on the roof structure though, was necessary to demarcate the subject and to gain knowledge on sound absorbing behaviour of materials used in sports halls. This knowledge is very useful for designers to choose materialisation and therefore to improve the acoustics in sports halls.

To link the designed wall to the design studio, the ‘Green Building and Innovation’ studio, is not very hard. The wall is typically an innovation. Besides that, when used, the wall will improve the acoustics in sports halls and cause a more healthy climate in the building. Creating a healthy indoor climate, that is what a ‘Green Building’ is in my opinion.

Yvonne Wattez

19. Literature

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