Resistance in Helmholtz resonators
Exploring the potential of sound absorption created by additive manufacturing
B. Gommer
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Exploring the potential of sound absorption created by additive manufacturing

by

B. Gommer

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Student number: 4010566
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Thesis committee: Prof. ir. R. Nijssse, TU Delft
Dr. Ir. M.J. Tenpierik, TU Delft, supervisor
Dr. M. Turrin, TU Delft
Dr. Ir. M.C.J. Hornikx, TU Eindhoven
Ir. J. Vugts, LBP|Sight

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I'm always interested in looking forward toward the future. Carving out new ways of looking at things.

- HERBIE HANCOCK
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Abstract

This research aims at improving the performance of Helmholtz resonators by adding resistance using additive manufacturing. Current use of Helmholtz resonators is focused on tonal applications and modal issues, because of the frequency specific behaviour. A normal Helmholtz resonator curve has a decreasing bandwidth when the peak increases. The research has been conducted by creating different samples, using the printing technique of stereolithography (SLA) and testing these in the impedance tube. Simultaneously the samples are modelled and analysed in Comsol. Starting from a theoretical research, the focus has been two-sided. First a possibility of adjusting the impedance of the absorbing mechanism was investigated, resulting in samples with larger squared geometrical adjustments. The results showed no significant improvement and very low velocity within the voids of the Helmholtz resonator's neck. Changes in the absorption coefficient were related to the reduction of the orifice radius, as was also predicted by analytical solutions. The second focus was the option of increasing the boundary layer thickness. This has been investigated by squared and triangular geometries with similar thickness. Results showed that there was no increase in boundary layer thickness and absorption coefficient. From these results it was possible to conclude that resistance is effective in the travelling direction of the plane wave, leading to a different iteration in the process. Surface area increase effects were researched by adding finned structures into the neck of the Helmholtz resonator and by using curved orifice patterns. Additional research on extending the fins within the volume proved not to be better performing. The results for the curved orifice showed an increase in absorption coefficient without adjusting the resonating frequency. By using the simulations in Comsol Multiphysics it was possible to predict the trend of the absorption coefficient curve, but additional research needs to be done to be able to rely on these results. From the research it can be concluded that there is a high potential for resistance based performance improvement by increasing the surface area of the Helmholtz resonator neck. The height of the absorption coefficient curve can be increased without decreasing (and thus enlarging) the bandwidth, allowing for an easier combination of resonators to create broadband absorption.
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List of Symbols

$\alpha$ Absorption coefficient
$\Gamma$ Propagation constant
$\gamma$ Specific heat ratio
$\delta$ Boundary layer
$\delta_{BL}^{\text{therm}}$ Thermal boundary layer
$\delta_{BL}^{\text{visc}}$ Viscous boundary layer
$\delta(t - t_0)$ Dirac delta function
$\epsilon$ Porosity
$\theta$ Angle
$\kappa$ Thermal conductivity
$\Lambda$ Viscous characteristic length
$\Lambda'$ Thermal characteristic length
$\lambda$ Wavelength
$\mu$ Dynamic viscosity
$\nu$ Kinematic viscosity
$\rho$ Density
$\sigma$ Flow resistivity
$\tau$ Shearing stress
$\phi$ Structural factor
$\phi$ Phase
$\chi$ Partial solution to differential equation
$\omega$ Angular frequency
$\omega_0$ Angular resonating frequency
$A$ Total absorption
$a$ Amplitude
$B$ Total absorption
$BW$ Bandwidth
$c$ Speed of sound
$c_d$ Linear viscous damping coefficient
$C_{50}$ Clarity
$C_p$ Specific heat at constant pressure
$C_V$ Specific heat at constant volume
$D_{50}$ Definition
$d$ Diameter
$d$ Thickness
$d_v$ Depth of resonator volume
$F_{\omega}$ Sinusoidal force
$f$ Frequency
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Introduction
1.1. Introduction

We are currently living in a world where the demand for customization is increasing rapidly (Gandhi, 2013). The search for sound absorbing materials that fit the aesthetic demand grows with the designers looking for a more minimalistic and clean solution. Integration and sustainability are an important issue within current designs and also the field of acoustics is expected to live up to this trend. Besides a simplistic design approach, the acoustic solution should mainly facilitate a comfortable sound environment, without noise from outside or sources within the space. In room acoustics, the focus is on the combination of sound absorbing, scattering and reflecting materials that will provide a high quality sound environment. Characteristics that determine the amount of absorption and reflection are depending on the type of absorbing principle but often include dimensions, thickness, material, material finish, and characteristics that are more dependent on the material properties, like pore size, pore percentage and air flow resistance (Arenas and Crocker, 2010; Kuttruff, 2007).

When creating the perfect sound environment, a good distribution of sound absorption and targeted frequencies is important. This is often done by combining different types of sound absorption, such as a porous material for mid and high frequency absorption and a panel absorber for low frequency absorption (Valk et al., 2006). However, when targeting specific frequencies, Helmholtz resonators are a very efficient way of sound absorption. The absorption coefficient curves show small peaks for the frequencies where the absorption is most efficient, which makes them perfect for tonal noise phenomena and room modes. The very steep and narrow curve also highlights the constraint of this absorbing principle. Knowing that the mechanics behind a Helmholtz resonator rely on the resistance of the mass-spring system (Kuttruff, 2007), the challenge is to influence the resistance of the mechanism in such a way that the bandwidth of the Helmholtz resonator can be increased and the use and efficiency of a resonator will be improved. This is where the technique of additive manufacturing comes along. The combination of additive manufacturing and sound absorption theories can create new complex shapes and therefore be a based on requirements, sustainable and compact alternative to the current absorbing techniques. The use of additive manufacturing can now become a potential of tuning sound absorption for the specific demands.

When looking for improvement of the sound absorption in general, one can focus on the surface impedance and resistance of the system (Seddeq, 2009). For Helmholtz resonators this can be done by looking at the actual damping of the mass spring system, which allows for loss of energy due to the viscous dissipation. Is it possible to influence the use of resistance by creating complex and random shapes with additive manufacturing? By doing this, is it possible to broaden the absorption spectrum of a Helmholtz resonator by adding resistance to the material or surface, everywhere or maybe in more specific places? Will this improve the characteristics, or diminish the effect of the actual system? These questions are a few examples of what can be the start of an interesting potential in the world of acoustics and additive manufacturing and will cover a part of what this research is focused on. If we think one step ahead, the results might lead to a thin broadband absorber or absorbers that target specific room acoustic issues.

1.2. Previous works

Additive manufacturing entered the market end of the 20th century (Wohlers, 2015) and little is known about its applications within the field of sound absorption. Two recent larger researches are from Godbold (2008) and Setaki (2013), including an ongoing research on Acoustic Design by Additive Manufacturing (Setaki et al.), also known as the ADAM Project. In the research by Godbold, the potential for additive manufacturing when designing more complex absorbers is discussed and proven to have large potential (Godbold, 2008). His work investigates the design and manufacture of acoustic absorbers incorporating resonant geometric features and improved porous solutions to demonstrate the performance as broadband acoustic absorbing devices. The starting point for the promotional research of Godbold are the possibilities of additive manufacturing, rather than the theoretical acoustics. Some of his results are presented in Appendix F. The potential of additive manufacturing is captured in the following aspects (Godbold, 2008).

- Design freedom
- No limitations due to tooling requirements
- More complicated geometries
- Minimal labour and constant material costs
Another research that uses the combination of additive manufacturing and sound absorption is from Setaki (2013). This research focuses on passive destructive interference absorbers (Setaki et al., 2013). The question arises if the sound absorption is completely caused by the principle of PDI, or that a large part of the sound absorption is caused by friction within the tubes. Currently the research is extended to quarter and half wavelength tubes, with a focus on low frequency broadband absorbers. Other recent work is done by Costa (2016), with a focus on creating a very thin low frequency absorber.

1.3. Research questions and objectives

Helmholtz resonators are used for frequency specific problems. The absorption coefficient curve shows steep peak behaviour. In addition, there is a great potential for the incorporation of additive manufacturing. The analytical equations show an interesting possibility for increasing resistance and therefore changing the performance of the absorbing material. Additive manufacturing can be the solution where we can turn to for creating the possible shapes and geometries, something that can not (always) be done with conventional techniques. This potential leads to the following research questions.

Main research questions

What is the effect of adding resistance to a Helmholtz resonator? Is it possible to improve its performance and create a broadband absorber with additive manufacturing?

Secondary questions

- Which sound absorbing mechanisms are known? How are they used?
- What are the potentials of these mechanisms when it comes to additive manufacturing?
- Which mechanism has the highest potential?
- Which additive manufacturing techniques can be used for the production of absorbing material?
- How does the roughness of the material effect the use?
- Will the addition of resistance or obstacles improve the sound absorption?
- Where in the system is the addition of resistance most effective?
- Are the improvements on specific parts of the frequency spectrum?
- What are the consequences or possibilities for the future?

1.3.1. Aims & Objectives

The goal of this research is to investigate the effect of using resistance to enhance sound absorbing mechanisms, creating a broadband absorber by using additive manufacturing techniques. The aim is not on designing an entirely new absorbing element, but in understanding the possibility of resistance within absorbing mechanisms that can be incorporated in several new design strategies. The research will look at a broad range of frequencies, from 125 to 4000 Hz. This frequency range is chosen to create a margin around the speech frequencies, which range between 250 and 2000 Hz. When creating a broadband panel around that frequency range, the strength of combining different mechanisms can be utilized to its maximum.

In order to keep the research within certain boundaries, the scope is defined as follows.

- Focus on the Helmholtz resonator as a starting point for the sound absorbing mechanism
- Use one material and additive manufacturing technique for the simulations and testing
- Consider the frequency spectrum between 125 and 4000 Hz
- The sound absorption elements will be used for specific room acoustic problems
- Limit the dimensions of the test object to the dimensions of the small impedance tube, which has a diameter of 29 mm.
- Limit the dimensions of the test object to a maximum of 5 cm in order to obtain a small and efficient sound absorbing mechanism.
- Aim for an absorption coefficient of \( \alpha_w \) of 0.8, corresponding with SA Class B, according to the NEN-EN-ISO 11654:2015.

1.4. Research method

To answer the main question, the research will consist of several sections. First, a literature study to the different aspects will give more insight in the current knowledge in this field. A division is made in terms of
acoustics, absorption methods, additive manufacturing techniques and materials. Based on this literature research, a first design for a sound absorber is made. There will be an iterative process between the design and the simulation. This simulation will be done with the software Comsol Multiphysics. The resonators are also printed and tested in the impedance tube at the faculty of Architecture, Delft University of Technology. The impedance tube is of the type Bruel & Kjaer 4206 and can hold samples of diameters 29 or 100 mm. It has a frequency range of 50 Hz to 6.4 kHz (Brul and Kjaer, 2012). The use of the two microphone method is as described in ISO 10534-2.

As mentioned, the software that will be used for this work is Comsol Multiphysics. This software is available at the TU Delft and allows the combination of acoustical simulations and CFD simulations. The wide range of applications will make it suitable for all aspects of this graduation thesis. Rhino and Grasshopper will be used to create the geometry. Implementing the geometry is possible with .SAT file formats. The measurements and simulations will be validated in the impedance tube at Eindhoven. An extensive description of the method is given in chapter 3.

1.5. Relevance
1.5.1. Science and research
For the research field the main relevance is knowledge. The effectiveness of the different absorbing mechanisms are discussed extensively during the last century, but what happens if we try to influence its performance by geometrical change? Besides that, the combination of sound absorption and additive manufacturing is new within this field. The potential of this combination opens up new possibilities. The main relevance for science and research is:

- Knowledge on tuning resistance within sound absorbing resonators
- Knowledge on potential for creating broadband absorbers with additive manufacturing
- Combination of sound absorption and additive manufacturing, input for further research

1.5.2. Practice
With respect to the building engineering practice, the relevance is mainly in the room acoustic design and the opportunity to make the product fit the demands perfectly. This will be good for the product development
branch, but in a later stage as well for the engineering and acoustic design practice. Another interesting part is the sustainability of using additive manufacturing. Especially the current use of fibrous materials are mostly synthetic, and difficulties with recycling occur. The use of recyclable plastics and minimizing the amount of material needed to create a product improves the sustainability of the sound absorption materials. The main relevance for practice is:

- Room acoustic design, integration with other disciplines
- Additive manufacturing perfect for adjusting to specific demands
- Reduction of material

1.6. Report outline

In chapter 2, background information on both acoustics and additive manufacturing is presented. The acoustical theory is split up in practical and theoretical acoustics, making a difference between application and measurements, talking about for example reverberation time, room acoustical parameters and speech intelligibility, and the fundamentals of acoustics, where the focus is on the derivations of sound absorption mechanisms and the way sound is attenuated by them. Thereafter, chapter 3 discusses the method for the research and the experiments. The software and measurement equipment will be highlighted. The results are presented in chapter 4. After this the conclusions to the research are presented in chapter 5. To translate the gathered information, a short case study will be discussed in chapter 6, where the adjusted Helmholtz resonators are used for improving a meeting room at the faculty of Architecture at the TU Delft. Chapter 7 will address several discussions and recommendations for further research.
Acoustics is the science of sound. It covers a broad range of topics, all related to the production of sound, sound propagation and the perception of sound. This chapter will provide the fundamentals needed to perform the research on both practical and theoretical level. It ends with the basics on additive manufacturing.
2. Background

2.1. Fundamentals of acoustics

Theoretical acoustics focuses on the fundamentals behind sound and absorption. The physics of sound waves and the absorbing mechanism and modelling of a Helmholtz resonator will be discussed here.

Wave equation Acoustic waves are defined by the propagation of small fluctuations in pressure, built up on the background atmospheric pressure which is stationary. The propagation of a sound wave in a medium can be described by the wave equation. It is derived from the conservation laws for mass and momentum and the additional constitutive equation (Rienstra and Hirschberg, 2004).

\[
\frac{\partial \rho}{\partial t} + \nabla (\rho v) = 0 \quad (2.1)
\]

\[
\frac{\partial}{\partial t} (\rho v) + \nabla (\rho vv) + \nabla p = 0 \quad (2.2)
\]

\[
dp = c^2 d\rho + \left(\frac{\partial p}{\partial s}\right)_{\rho} ds \quad (2.3)
\]

With \( \rho \) being the density of the medium in kg/m\(^3\), \( v \) the velocity in m/s, \( p \) the pressure in N/m\(^2\) and \( c \) the speed of sound in m/s. With these equations we are able to determine the wave equation and its solution. For the full derivation, please look at Appendix C. The wave equation stated below is the homogeneous wave equation that is valid for many applications.

\[
\frac{\partial^2 \phi'}{\partial t^2} - c_0^2 \nabla^2 \phi' = 0 \quad (2.4)
\]

The general solution to the wave equation, when assuming plane waves is:

\[
p' = f(x - c_0 t) + g(x + c_0 t) \quad (2.5)
\]

\[
v' = \frac{1}{\rho_0 c_0} \left( f(x - c_0 t) - g(x + c_0 t) \right) \quad (2.6)
\]

\( f \) is defined as the right-running wave, \( g \) is the left-running wave. With these equations for \( p \) and \( v \) we can determine the impedance, and thus the absorption coefficient. Having a sound wave in air, the impedance equals \( p/v = \rho_0 \cdot c_0 \). The pressure and velocity travel in phase and the impedance only contains a real part.

The vibration of the pressure disturbances described by the wave equation is the fundamental definition of sound. These pressure waves contain high and low pressure areas, having a certain amplitude and period because of the sinusoidal behaviour. The wavelength \( \lambda \) is the distance between two high pressures and defines the frequency \( f \) of the sound wave. Figure 2.1 shows the different representations of the soundwave with first the high and low pressure of the longitudinal plane wave and second the sinusoidal representation of the high and low pressures.

Figure 2.1: Representations of a sound wave

\[
f = \frac{c}{\lambda} \quad (2.7)
\]
With \( f \) being the frequency in Hz, \( c \) the speed of sound, equal to \( 343 \frac{m}{s} \) and \( \lambda \) the wavelength in m. The frequency is the reciprocal of the period \( T \) that it takes to perform one oscillation, which means that 1 Hz equals 1 \( \frac{1}{s} \). From this we can see that low frequencies have a long wavelength and high frequencies travel with short wavelengths.

**Sound absorption**  When such a sound wave reaches a boundary it can either be absorbed, reflected or transmitted.

![Figure 2.2: Absorption, reflection and transmission through a boundary element](image)

Within the reflection, a distinction has to be made between mirrored reflections and diffuse reflection. Mirrored reflection occurs when sound coincides with a very smooth surface. Diffuse reflection causes a large amount of scattering due to a rough surface.

The absorption coefficient is a quantity that is used to show the amount of the sound energy that is not reflected from the wall or transmitted through the wall. One of the ways to determine the absorption coefficient is by determining the acoustic impedance. At the boundary the acoustic impedance shows how much of the incoming force (the motion induced by a pressure) is damped (Rienstra and Hirschberg, 2004). This results in a decrease in amplitude and a shift in the velocity phase. The phase shift results in a partially imaginary impedance. We can see this as the maximum pressure being ahead or behind the maximum particle velocity. As discussed earlier, the impedance is defined as the ratio between the sound pressure and particle velocity, resulting in:

\[
Z = \left( \frac{p}{v} \right) \left[ \frac{Pa \cdot s}{m} \right]
\]  \( (2.8) \)

With \( Z \) being the impedance, \( p \) the sound pressure and \( v \) the particle velocity.

The imaginary part then represents the part that is out of phase. This impedance can be used for calculating and determining the reflection and absorption coefficients. It can also be formulated using the reflection coefficient.

\[
Z = \frac{Z_0 \cdot 1 + R}{\cos \theta 1 - R} \left[ \frac{Pa \cdot s}{m} \right]
\]  \( (2.9) \)

\( Z_0 \) is the specific impedance of air, and is equal to \( \rho_0 c_0 \). \( R \) is the reflection factor, and is calculated with the following equation. When only normal sound incidence is considered, \( \theta = 0 \) and we can rewrite:

\[
R = \frac{Z - \rho_0 c_0}{Z + \rho_0 c_0} \quad [-]
\]  \( (2.10) \)

This equation also contains \( \rho_0 c_0 \), the characteristic impedance of air, that only consists of a real part. This reflection coefficient is only true for normal incidence, which holds for the measurements done in the impedance tube. However, it is never the case that only normal incidence occurs in room acoustics. The absorption coefficient can be derived with:

\[
\alpha = 1 - |R|^2 \quad [-]
\]  \( (2.11) \)
The Helmholtz resonator is used to absorb sound within a specific frequency range. The mechanism can be described as a mass-spring system with one degree of freedom. The resonating frequency is determined by the solution to the differential equation that defines the mass-spring system, as shown in equation 2.12 (Gavin, 2014).

The $k$ denotes the linear elastic stiffness coefficient of the spring, which is the volume of the resonator. The system is damped by $c_d$, the linear viscous damping coefficient. $m$ determines the mass of the system, equal to the mass of the air in the neck of the resonator with the effective length taken into account. $f(t)$ is the external excitation force, which in our case is a sinusoidal pressure wave. The displacement of the mass is given with $x(t)$. This gives us the equation of motion in the form of:

$$m\ddot{x}(t) + c_d\dot{x}(t) + kx(t) = f(t)$$  \hspace{1cm} (2.12)

Implementing the sinusoidal forcing and derivations of $x(t)$ in time:

$$- (m\omega^2 + c_d i\omega + k)\chi(\omega)e^{i\omega t} = F(\omega)e^{i\omega t}$$ \hspace{1cm} (2.13)

With $\chi(\omega)e^{i\omega t}$ being the partial solution. To define the resonating frequency, the real part of the solution must equal zero, causing:

$$k - m\omega^2 = 0$$ \hspace{1cm} (2.14)

$$\omega_0 = \sqrt{\frac{k}{m}}$$ \hspace{1cm} (2.15)

To obtain the known Helmholtz equation, we need to implement the equations for the mass and the spring stiffness.

$$k = \frac{\rho \cdot c^2 \cdot S^2}{V}$$ \hspace{1cm} (2.16)

$$m = L_{\text{eff}} \cdot \rho \cdot S$$ \hspace{1cm} (2.17)

$$\omega_0 = c \cdot \sqrt{\frac{S}{V(L+2\delta)}}$$ \hspace{1cm} (2.18)

With $S$ the surface area of the orifice, $L_{\text{eff}}$ the efficient length of the neck, equal to $L + 2\delta$, and $V$ the volume of the cavity. Reformulating this for frequency ($f$) instead of the angular frequency ($\omega$):

$$f_0 = \frac{c}{2\pi} \cdot \sqrt{\frac{S}{V(L+2\delta)}}$$ \hspace{1cm} (2.19)

Within the mechanical analogy, the damping of the resonating frequency is provided by $c_d$, the linear viscous damping coefficient in equation 2.12. This is determined by the dissipation of the sound energy due to the transfer of motion energy to internal energy (heat). This is mainly observed in boundary layer regions (2.1 Acoustic boundary layer), where large velocity gradient occur, and in turbulent flow areas. The derivation and effectiveness are often neglected in large systems, but are very relevant for small scale research. Energy dissipation is the result of a gas-borne sound with solid structure interaction (Rossing, 2007). This interaction can be caused by viscous shear, leading to viscous dissipation, or thermal relaxation and expansion, leading
to thermal dissipation. Thermal dissipation focusses on the constant contraction and relaxation of high and low pressures, which we see in wave propagation. Because of the small pressure deviations that occur when transferring sound, the thermal dissipation is of a very small scale and is often neglected. (Blackstock, 2000)

In terms of impedance, we can find the viscous damping in the following equations, that are used to calculate the impedance of a Helmholtz resonator with (optional) porous material in the cavity. The impedance at the entrance of the Helmholtz resonator is given by \( z_3 \) (Cox and D’Antonio, 2009):

\[
z_3 = \frac{\rho}{\epsilon} \left( \frac{l}{2r} + 1 \right) \sqrt{8\nu \omega + (2\delta r + l) \frac{j\omega\rho}{\epsilon}} + z_2 + r_m
\]  

(2.20)

with

\[
r_m = \frac{\rho}{\epsilon} \left[ t + 2\delta r + \sqrt{8\nu \left( 1 + \frac{l}{2r} \right)} \right]
\]  

(2.21)

and

\[
z_2 = \frac{-z_1 j \rho c \cdot \cot(k d_v) + \rho^2 c^2}{z_1 - j \rho c \cdot \cot(k d_v)}
\]  

(2.22)

with

\[
z_1 = -j z_i \cot(k_i d_i)
\]  

(2.23)

With \( z_i \) the characteristic impedance, \( k_i \) the wavenumber and \( d_i \) the thickness of the porous absorber (if applicable). \( z_1 \) is the impedance at the top of the porous material in the cavity, but this is neglected when it is not implemented. \( z_2 \) is the impedance at the top of the air layer in the resonator, so right below the perforation.

The last term in equation 2.20 and described in equation 2.21 accounts for the boundary layer effect. The equations furthermore contain \( l \) as the length of the Helmholtz neck (also often described as the sheet thickness), \( r \) the neck radius, \( \epsilon \) the porosity or fractional open area, \( k \) the wavenumber, \( d_v \) the depth of the Helmholtz resonator volume, \( \delta \) the end correction, \( \nu \) the kinematic viscosity of air, \( \rho \) the density and \( c \) the speed of sound.

When we consider individual elements, a resonator provides damping not only for the surface area of the orifice, but it has a larger absorption cross section (Beranek and Ver, 2005). For optimal use of the resonator, this should be taken into account when designing a panel with Helmholtz resonator combinations. Placing multiple resonators that are tuned to the same frequency within their absorption cross section can cause interference.

\[
A_{b}^{max} = \frac{\lambda^2_0}{4\pi}
\]  

(2.24)

From equation 2.24 we can calculate the spacing between the different resonators.

\[
r_{A_b} = \frac{\lambda_0}{2\pi}
\]  

(2.25)

The resonating frequency can be affected by adjusting the geometrical features of the Helmholtz resonator. A research by A. Selamet and I. Lee (2003) shows the effect of extending the neck into the volume of the resonator. Additionally the possibility of perforating the neck extension was researched, in order to increase the neck length, but still incorporate the entire resonator volume.
These results show that increasing the porosity of the neck extension increases the resonating frequency and lowers the transmission loss up to a porosity of 1%. Further perforations lead to an increase in transmission loss.

**Acoustic boundary layer**  The acoustic boundary layer is a combination of the viscous and thermal boundary layer, that is created when a sound wave propagates along a solid surface (Mikhail, 1991). Including this in calculations is essential when analysing the correct physical behaviour of small dimensions, where the losses can become significantly important. Working with boundary layers, there are two important assumptions. At the wall a no-slip condition is assumed, meaning that the velocity equals zero and builds up within the boundary layer to the bulk velocity. Also, it is assumed that there is an isothermal condition on the wall, thus \( T = 0 \), where in general we assume adiabatic conditions. When looking in more detail to the boundary layer, we see that it is defined by viscosity and adherence of the fluid to the wall. This leads back to the frictional forces, that are defined per unit of area. The most well known and basic definition of viscosity is Newton’s law of friction (Schlichting, 1979).

\[
\tau = \mu \frac{du}{dy} \tag{2.26}
\]

With \( \mu \) is the dynamic viscosity of the fluid, with unit \( Pa \cdot s \). The derivative \( \frac{du}{dy} \) is the change in speed over the cross section. The kinematic viscosity is known as the ratio of the viscosity \( \mu \) to the density \( \rho \) and has unit \( m^2/s \).

\[
\nu = \frac{\mu}{\rho} \tag{2.27}
\]

This means that the frictional forces slow down the motion of the fluid. At the exact location of the wall there is no velocity, and at the other end of the layer the velocity corresponds to the frictionless flow (travelling with the bulk velocity). The boundary layer is found by L. Prandtl (2001 (original 1904)). A smaller viscosity reduces the thickness of the boundary layer. Small viscosity corresponds to large Reynolds numbers.

\[
Re = \frac{ud}{\nu} \tag{2.28}
\]

With \( u \) the fluid flow speed, \( d \) the diameter of the pipe and \( \nu \) the kinematic viscosity of the fluid. The flow field can be divided in two parts: first the normal flow, where we can consider the perfect fluid theory, and second the boundary layer, where we should consider friction. The formation of vortices and therefore acoustical energy losses are the result of boundary-layer separation. This occurs when the layer becomes thicker and the flow reverses. This phenomena is often observed at bodies, such as cylinders and spheres.
2.1. Fundamentals of acoustics

Figure 2.5: Boundary layer velocity profile (source: Rienstra and Hirschberg)

Figure 2.5 shows a typical boundary layer velocity profile. The boundary layer depth ($\delta$) is the acoustic boundary layer, again the combination of the thermal and viscous boundary layer. The thickness of these layers has been determined with an experiment by Meyer and Güth (Blackstock, 2000). The viscous boundary layer excites a shear force on the surface.

\[ \delta_{BL}^{\text{visc}} = \sqrt{\frac{2\mu}{\omega \rho_0}} = \sqrt{\frac{2\nu}{\omega}} \]  

(2.29)

With $\mu$ the dynamic viscosity, $\nu$ the kinematic viscosity, $\omega$ the angular frequency and $\rho_0$ the density of the medium. The thermal boundary layer is in the same order of magnitude as the viscous boundary layer and uses the fact that very close to the wall the process is no longer adiabatic, but isothermal. The thermal boundary layer transfers heat through the surface to the surrounding volume (Blackstock, 2000).

\[ \delta_{BL}^{\text{therm}} = \sqrt{\frac{2\gamma}{\rho_0 \omega C_p}} = \sqrt{\frac{2\mu}{\omega \rho_0 Pr}} = \delta_{BL}^{\text{visc}} \]  

(2.30)

With $\gamma$ as the specific heat ratio. $Pr$ is the Prandtl number, which is equal to 0.713 [-] for air at $20^\circ C$. $\nu$ is the kinematic viscosity of air, which is equal to $15.1 \cdot 10^{-6} \ [m^2/s]$ in air at $20^\circ C$. $\mu$ is the dynamic viscosity, which is equal to $1.983 \cdot 10^{-5} \ [Pa \cdot s]$ in air at $20^\circ C$. The equations clearly show that the thickness of the boundary layer is depending on the frequency obtained. The following figure shows this relation. Unless mentioned otherwise, the medium in which the system operates is air. The m-file of the following figure can be found in Appendix G.1.

Figure 2.6: The thickness of the boundary layer as a function of the frequency
2.2. Room acoustics

Within the defined space of a room, sound is reflected and can cause disturbance or annoyance. A good quality of the comfort is important for health and work efficiency (Kuttruff, 2007). Important acoustical quantities are the impulse response, the Schroeder frequency, reverberation time and room quality characteristics, such as the sound transmission index (STI) and C50. These quantities are determined by parameters such as dimensions, volume, materials and shape. As an introduction to the propagation and use of sound in a room, a couple of these quantities are discussed here. Additional parameters are discussed in Appendix A.

**Sound pressure level** The frequencies that we can perceive as humans range between 20 Hz and 20 kHz, depending on various factors such as the distance of the source and the age of the perceiver. Normal speech ranges mainly between 500 Hz and 2 kHz, with the vowels and bass in lower frequencies and consonants at higher frequencies. The sound pressure level is the root mean square of the instantaneous sound pressure measured over a specific period of time, in decibels (dB). (Gracey & Associates)

\[ SPL = 20 \cdot \log_{10} \left( \frac{\bar{p}}{p_b} \right) \]  

(2.31)

With \( \bar{p} \) being the root mean squared pressure and \( p_b \) the reference pressure, equal to \( 2 \cdot 10^{-5} \). The fact that this is a logarithmic equation makes that doubling the sound pressure in free field conditions results in a 6 dB increase of the SPL and a ten-time pressure increase results in a 20 dB SPL increase (at 1 m distance). In our surroundings this is not that straightforward, because the sound that enters our ears is a combination of direct sound coming from the source and the indirect sound, that has been reflected by surfaces of the room that we are in. These reflections are of big influence on the sound level and the reverberation time of the room and therefore its acoustical quality. Where the sound is reflected by a surface, a part of the sound energy is absorbed. This depends on the sound absorption coefficient of the material. All the remaining reflections in the room combined cause the indirect sound. Both the direct and indirect sound contain information about the acoustical quality of the room.

![Figure 2.7: Direct and indirect sound in a room](image)

**Impulse response** When we have an impulse within a closed volume, it is possible to measure at a certain point the response to that impulse. This is called the impulse response of the room. The receiver will first hear the direct sound, since it has travelled the shortest distance within the room. The rest of the impulse response is generated by the reflections. The impulse itself is idealised as a Dirac delta function. The impulse does not necessarily need to be an impulse. Measurements are often done using a sweep.

\[ g(t) = \sum_n a_n \delta(t - t_n) \]  

(2.32)

The delta function is a representation of the short impulse and contains a very short signal at \( t = t_n \). The constant \( a_n \) contains information about the amplitude of this impulse. The response of the room is the sum of the signals that are received by the listener.

\[ s'(t) = \sum_n a_n s(t - t_n) \]  

(2.33)
With $s(t - t_n)$ being a single signal received, and $s'(t)$ the sum of all the signals, giving the complete impulse response. This impulse response is often seen as the 'signature' of the room, since it contains the characteristics of the room (Kuttruff, 2007). It is used to calculate acoustical parameters and all acoustic modelling software such as Odeon, EASE and CATT Acoustics calculate room parameters based on the ability of build realistic impulse responses.

**Reverberation time**  The most important acoustical quantity to assess the quality of a room is the reverberation time, which is the time it takes to let the sound pressure level in a room decrease with 60 dB. This can be measured, using a sound level meter or calculated with the theory from Sabine. Sabine’s formula for the reverberation time is the most well known and commonly used because of its simplicity and therefore quick estimate of the reverberation time. The equation is mainly suitable for rooms that have an evenly distributed sound absorption and equal width, length and height.

\[
T_{60} = \frac{24 \cdot \ln 10}{c} \cdot \frac{V}{\sum_{i=1}^{n} \alpha_i \cdot S_i} = 0.161 \cdot \frac{V}{A}
\]  

(2.34)

With $c$ being the speed of sound, $V$ is the volume of the room and $A$ the total amount of absorption in $m^2 \text{ sabine}$ which is the sum of all the absorbing areas times their absorbing coefficients. The reverberation time $T_{60}$ is measured in seconds. It is frequency dependent and often shown per octave band. To calculate the reverberation time, the material finishes and their absorption coefficient must be known and serve as input for the total amount of absorption, $A$. Other methods of calculating the reverberation time are:

- **Eyring**: \(RT = 0.161 \cdot \frac{V}{-S \cdot \ln(1 - \alpha_{\text{avg}})}\)  

(2.35)

- **Fitzroy**: \(RT = 0.161 \cdot \frac{V}{\left(\frac{S_x^2}{A_x} + \frac{S_y^2}{A_y} + \frac{S_z^2}{A_z}\right)}\)  

(2.36)

- **Millington-Sette**: \(RT = 0.161 \cdot \frac{V}{\sum (S \cdot \ln(1 - a))}\)  

(2.37)

- **Arau-Puchades**: \(RT = \frac{0.161 \cdot V \left(\frac{S_x}{A_x}\right)}{A_x}, \frac{0.161 \cdot V \left(\frac{S_y}{A_y}\right)}{A_y}, \frac{0.161 \cdot V \left(\frac{S_z}{A_z}\right)}{A_z}\)  

(2.38)

These methods are presented without the addition of sound absorption in air. In larger spaces the effect can be significant and additional absorbing area of the air volume must be taken into account:

\[A_{\text{air}} = 4mV\]  

(2.39)

With $m$ as the intensity attenuation coefficient in $m^{-1}$ and $V$ the volume of the room in $m^3$. Every method has its own characteristics and advantages and it should be investigated per room to see which method most carefully captures its characteristics. Other than Sabine’s equation, these four approaches are based on the axial differences in distribution of the sound absorbing material. In addition, a very extensive calculation method is presented in the appendix D of EN12354-6. When measuring the reverberation time, there are several ways to determine the time for the 60 dB decay. Most common are the T20 and T30, calculated by taking the 20 and 30 dB decay and extrapolating that to a 60 dB decay as illustrated in figure 2.8. In a diffuse sound field and with equally distributed sound absorption the two parameters will most likely be similar. However, this is not always the case, as shown in the figure. This phenomena where the T20 and T30 show significant differences is called a double-decay and often observed in rooms with unequal distribution of the sound absorbing material.
The speech transmission index (STI) is a measure for the quality of the speech intelligibility in a room. This is ranked in classifications, shown in the table 2.1 (Rossing, 2007).

Table 2.1: STI ranking

<table>
<thead>
<tr>
<th>STI</th>
<th>Ranking</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00-0.30</td>
<td>Bad speech intelligibility</td>
</tr>
<tr>
<td>0.30-0.45</td>
<td>Poor speech intelligibility</td>
</tr>
<tr>
<td>0.45-0.60</td>
<td>Fair speech intelligibility</td>
</tr>
<tr>
<td>0.60-0.75</td>
<td>Good speech intelligibility</td>
</tr>
<tr>
<td>0.75-1.00</td>
<td>Excellent speech intelligibility</td>
</tr>
</tbody>
</table>

The STI can be calculated using a modulation transfer. When speech propagates through a room it gets reflected and absorbed before it reaches the receiver. The amplitude and characteristics of the speech will be changed. These modulation depth changes are captured in the modulation transfer function MTF or m(F). It describes the amount of noise or disturbance that is added to the initial sound, such as speech, and thus reducing the intelligibility.

\[
m(F) = \frac{1}{1 + \left(\frac{2\pi F T}{13.8}\right)^2} \quad \frac{1}{1 + 10^{-S/N/10}}
\]

(2.40)

\[
(S/N)_{app} = 10 \log_{10} \left( \frac{m}{1 - m} \right)
\]

(2.41)

\[
(RA)STI = \frac{(S/N)_{app} + 15}{30}
\]

(2.42)

With \(m(F)\) being the modulation transfer of that frequency, \(S/N\) the signal to noise ratio, and \(T\) the reverberation time of the room. The STI is often measured.

Flutter echoes A flutter echo is defined as a repetitive echo caused by two parallel highly reflective surfaces. It can be heard when a sound source is excited between these boundaries and can clearly be seen in the impulse response because its acoustical energy is higher than the surrounding reflections. It is possible to reduce this echo by placing obstacles to diffuse the sound field or adding sound absorption on one of the two reflecting surfaces.

Sound absorption methods Sound absorption can be categorized in three principles:

- Porous absorption
- Resonators
- Panel absorption
2.3. Additive manufacturing

Porous absorbers are most efficient on mid and high frequencies. This is caused by the wavelength, which is small, and therefore less material is needed to create an effective absorber. To get optimal results from the friction of the porous material, the sound should not be reflected. The porosity should be large and the airflow resistance of the material should be low, because the transition from air to material has to be smooth. However, it should not be too low, because that would decrease the amount of friction and therefore the amount of absorption (van der Linden and Zeegers, 2006). Sound loss in a porous material is mainly due to the effect of the viscous and thermal boundary layer. The friction along the sides of the pores cause a layer that is of submillimetre size. Other important causes for sound pressure reduction are loss in momentum due to changes in the sound path and thermal conduction, which mainly has effect on the low frequencies. Losses due to the vibration of the material are usually less important. The material properties of porous materials are mainly defined by the flow resistivity, porosity, tortuosity and characteristic length. (Cox and D’Antonio, 2009)

The second method is the resonator. An example of this is the Helmholtz resonator. It has an enclosed air volume and a neck which contains vibrating air. The system works as a mass-spring damper and will be extensively explained in the following section. Resonators can target a specific frequency and are most efficient in low and mid frequencies. A more extensive explanation of Helmholtz resonators is given in section 2.1.

A different absorber is the panel absorber, which targets low frequencies. By using a massive plate, without perforation, the sheet forms a mass-spring system with the air layer behind. The effective absorption is between 50-500 Hz. By filling the air cavity with porous material, the effectiveness is reduced to max $\alpha = 0.5$, but the frequency range is wider. (van der Linden and Zeegers, 2006) The mechanism is very similar to the Helmholtz resonator, both using the mass-spring system to absorb sound energy.

![Figure 2.9: Different sound absorbing mechanisms and their typical curves](image)

Figure 2.9 shows the typical curves of porous, helmholtz and panel absorbers. A more extensive explanation of the different absorbing mechanisms is attached in Appendix A.

2.3. Additive manufacturing

Additive manufacturing is one of the techniques of rapid prototyping. Creating prototypes can be done in three ways, and is therefore often divided in three categories, mentioned below (3DPI, 2014). Nowadays these techniques are more and more implemented as production techniques as well. By using CNC production (Computer Numerical Control), the system movements are programmed and can therefore easily create an automated production.

- **Subtractive.** Starting from a block of material, the material is removed until the desired shape is obtained.
- **Additive.** Materials or elements are combined to create the desired shape.
- **Formative.** A specimen is shaped with the use of forces.

Most techniques within rapid prototyping are additive techniques (Liou, 2008). A lot of techniques can be seen as a form of rapid prototyping. There are three conditions that the technique has to meet before calling itself a rapid prototyping technique (Kruth et al., 1998).
- The technique should not contain a lot of skilful human intervention.
- No special tooling is needed to create a specific element. This condition is set with a focus on post-
  processing.
- There are very little geometric limitations.

The choice for the technique that should be used depends on the needed accuracy, need for support structure
and complexity of the model. For this research high precision is needed and therefore the used technique is
stereolithography (SLA). Stereolithography works with photopolymer resins. When it gets in contact with a
laser, the liquid forms a solid. Within the bed of resin, a moveable platform lowers the object layer by layer.
An accurate object can be printed with this technique. However, it needs supporting structures when the
printed object contains overhangs or fragile parts. These supporting structures can be removed afterwards.
This is something to take into account, in case the object has closed or hard to reach volumes. Before the
final product is ready, it needs to be cured in an oven-like machine where it is fully hardened. The material
used for the samples is an acrylic-based photopolymer with an accuracy of 0.1 mm. Additional techniques
and further information on additive manufacturing can be found in Appendix B.

Applications Currently the fastest growing sectors that are using additive manufacturing are the medical
and dental, automotive and aerospace industries. They all create products with a very high level of com-
plexity, directly showing the incredible advantage of additive manufacturing for these projects. Besides these
industries, there is a lot of potential for other applications. There are several obvious business oriented advant-
geas such as time savings and reduction in errors, but there are several points that can be of added value for
additive manufacturing in combination with acoustics and other disciplines (Royal Academy of Engineering,
2013; Wohlers, 2015; Bosschaert, 2016).

- **Complexity.** The use of optimised design allows for the production of very complex structures.
- **Integration.** Integration between different disciplines can be even more exploited than using conven-
tional techniques.
- **Low volume production.** The replacement of machine tooling facilitates customization.
- **Lower-cost production.** Currently true for very large complex structures that require intensive con-
struction with conventional techniques.
- **Responsive production.** With a quick analysis of the performance the production of new elements can
  be done within a very short time frame.
- **Short supply chains.** Parts can be produced on site, reducing the transportation costs.
- **Optimised design.** Rethinking design and creating structures that serve the purpose to perfection is a
  great advantage of the optimization of the design.

Besides the stated advantages that are already being exploited, there are several constraints and challenges
for the future. The application of additive manufacturing is still growing and a lot of focus is currently on the
following topics.

- **Materials.** New materials can create new opportunities for different markets.
- **Software.** When designing very complex and innovative structures for additive manufacturing, the
current CAD software can give undesired limitations.
- **Data management.** Faced limitations are often in strong relation to computational capability. Instead
  of looking at machine advancements, the focus can be on software developments for better data man-
agement.
- **Reducing waste.** Even though the general thought of additive manufacturing as a sustainable issue is
  partially true, a strong challenge is at the home designer, that creates a lot of waste.
- **Affordability.** The high overhead costs are a current limitation for the commercialisation of additive
  manufacturing.
- **Speed.** High volume production is considerably slower than conventional techniques.
- **Reliability.** The reliability of the very precise reproducing of elements is currently not high and is a
  limitation for several industries.
- **Standards.** With additive manufacturing being in a very quick lift, the demand for new (ISO) standards
  is high.
- **Education.** The current industry is built on the knowledge of the previous generation. Convincing and
  learning everyone about the potential and capabilities of additive manufacturing is a must to incorpo-
rate it in today's practice.
2.3.1. The use of additive manufacturing in acoustics
Additive manufactured resonators have a lot of chance to be implemented in a lot of high-end applications, such as the aerospace industry. For example, we have seen that acoustic liners have to be built in very small spaces and optimized for certain frequencies (Howerton, 2009). There are many other researches within this field, showing the successes. Complex projects are more likely to implement additive manufacturing, also from an acoustic viewpoint. However, when looking at room acoustic applications the amount decreases rapidly. The figures in appendix F are results from the PhD research done by Godbold (2008). These figures show the effect from dimensional changes to a Helmholtz resonator and the absorption coefficient changes of porous materials with different porosities. The geometrical adjustments are done with additive manufacturing, mainly using SLS. For the Helmholtz resonator the results show that:

- A smaller cavity depth gives a higher resonance frequency. Half of the depth heightens the resonant frequency with around 100 Hz. The width of the resonant peak is not much influenced by the depth.
- A longer orifice gives a lower resonance frequency and reduces the width of the resonant peak.
- A smaller orifice width lowers the resonant frequency drastically. This is combined with a smaller peak width.

Another recent project is from S. Costa (2016), that focuses on the design of thin low-frequency sound absorption by additive manufacturing. Using spiral tubes, the elements contained different lengths and thus different resonating frequencies. A special focus was on the spacing of the different resonators, creating a pattern with different types of tubes. The current research from F. Setaki (2013) focuses on the potential of acoustic design by additive manufacturing (ADAM Project). It is performed at the Delft University of Technology and aims at creating broadband absorption by resonators, such as quarter and half wavelength tubes.

**Characteristic demands** For this research, there are several characteristics that should are important for creating the optimal condition of the material and printing technique.

- The ratio of smoothness/roughness might have a significant influence on the results.
- Acoustically hard, meaning a very low porosity.
- High accuracy, to reduce possible effects on the boundary layer.
- Strength, so it will not break easily.
- Removal of support structure or powder, depending on the proposed technique.
This chapter describes the methodology that is used for the general research and the design of the different samples.
3.1. Main research methodology
In a broader design methodology, the larger steps of the research show the process from the starting point, a simple plain Helmholtz resonator, to the final step which will be the use of the sound absorbers in a practical case study.

For this research the Helmholtz resonator is chosen to be a good starting point due to its well known mechanism and the targeted frequency range. Additional research on the possibilities of using the three different absorbing mechanisms has been performed and is attached in Appendix A.

The addition of resistance will be based on theory and the analytical equations given in Chapter 2 Background. These will guide the design and alterations of designs, depending on the results and explanations of the findings. The focus will be on impedance and boundary layer formation.

The results are gathered in a constant flow of simulations, prints and measurements. This is the main focus of the research.

To create an absorber that can be applied to the current practice, some research will be done on the tuning of the Helmholtz resonator, the combination of different elements and the way it can be applied in a small meeting room by using a case study.

3.1.1. Design methodology
The research that will be conducted fits within the section of Resistance, impedance and boundary layer theory. The ground for the addition of resistances lies in the theory of the Helmholtz resonator (section 2.1), where the boundary layer and impedance changes are important factors for sound absorption. This led to several first designs, that are explained in figure 3.2.
As discussed in chapter 2 the focus on absorption coefficient adjustments is based on impedance and the effect of the boundary layer. Impedance is directly related to the sound pressure and velocity. Abrupt changes in geometry tend to create velocity and pressure changes and are therefore key to varying the impedance in the Helmholtz resonator. On the other side, boundary layers work in a sub millimeter region next to the hard walls. Using small scale squares in the neck increases the surface area and could increase the boundary layer. From this starting point, the focus was on two aspects. First the differences in geometry size and second the different geometries. The results and explanation of the physics will be presented in Chapter 4: Results. The following figure shows the different geometries used during the research in a schematic way.

Figure 3.2: Design methodology

Figure 3.3: The different types of geometries explained using icons and sections
3.2. Software
During the research, different software is used to create, simulate and test the samples. Both the software and the compatibility is shown in figure 3.4. Rhino is a 3D modelling software, with a plugin Grasshopper for parametric design. This is used to easily develop multiple samples with different geometries. From here the models are exported in different formats to import in Comsol Multiphysics and upload at Shapeways for printing. Comsol Multiphysics is a CFD analysis software with different modules for various applications (Comsol Multiphysics). For this research we use the thermo-acoustics module that has the possibility of boundary layer analysis. Within Comsol we simulate the measurements in the impedance tube, using the transfer function method and additionally analysed the fluid flow and energy dissipation within the resonator. The results are post-processed in Matlab, together with the results from the real measurements in the impedance tube.

Figure 3.4: Compatibility and software overview

3.3. Simulations and testing
A great part of the research consists of simulations in Comsol Multiphysics and testing in the impedance tube. This section explains certain settings and boundary conditions that are used to conduct the experiments.

3.3.1. Comsol Multiphysics
The numerical calculations of the resonators are done in Comsol Multiphysics, using the thermoacoustics module. This allows for a boundary layer calculation and can, besides pressure acoustics, also calculate temperature and velocity changes within the domains. As described in 2.1, it is important to take these viscous and thermal losses into account. Within Comsol Multiphysics, the sample is imported and the impedance tube is recreated by using workplanes, circles and the extrude tool. The microphones are placed by surface point probes that measure the pressure per frequency as a result from the input pressure in the tube. By doing this, it is possible to create a one-to-one modelling of the impedance tube set-up, since both methods will use the transfer function to process the data from the pressure points. Instead of modelling the surrounding material, Comsol works with the boundary of the medium (air). This means that the inverse sample is modelled, and only the air paths of the resonator and the tubes are used as geometry. The assigned material is air from the material library, having already some pre-set material properties. Several other boundary conditions and generic parameters are defined, these are listed below.

- $c_0 = 343 \, [m/s]$
- $\rho_0 = 1.2 \, [kg/m^3]$
- $Z_0 = c_0 \cdot \rho_0 = 411.6 \, [Pa \cdot s/m]$
- Normal impedance at change from large to small tube equals $Z_0$
- Bulk viscosity assumed to be zero, done for all ideal gases. Air is in this case close to an ideal gas.
- Input pressure equals 1 Pa
- The initial values for the velocity field, pressure and temperature variation are all equal to zero
- Mesh input varies from normal (large tube) to extra fine (resonator’s neck), using free tetrahedral geometries. For the finer regions, the mesh is adjusted manually to make sure that the smallest element is small enough for the targeted frequency (maximum 1/10 of $\lambda$) and boundary layers can be formed.

1The choice for this pressure is arbitrary since eventually the calculations use the ratio between the two pressure points in the tube. However, in real impedance tube measurements the input pressure should be large enough to discard the background noise. An input pressure of 1 Pa equals a sound pressure level of around 95 dB.
3.3. Simulations and testing

- The boundary layer contains 8 layers with growth factor 1.2 to achieve a gradual growth to the normal mesh size in the volume.
- The frequency domain over which most of the models are calculated is the following: range(100, 10, 6400) \(^2\). When this is not possible, it will be noted in the results.

The results for the pressure point probes are gathered in a table and exported in a .csv format. This is used as input for Matlab, where the two microphone method is applied. Other figures are post-processed in Comsol, using both 2D and 1D plot groups. For this, additional data sets are created to provide the plane or line location.

3.3.2. Impedance tube

The measurements are conducted following ISO 10534-2. The impedance tube used is of the type Brüel & Kjaer 4206, which uses the two microphone method to calculate the sound absorption coefficient. The derivation of the two microphone method, also known as the transfer matrix method can be found in Appendix C.2. This derivation is also used as a basis for the Matlab file that calculates the absorption coefficient based on the output of Comsol Multiphysics, as explained in the previous section. The m-file is attached in Appendix G.6. The impedance tube contains a small and large tube, with a diameter of respectively 29 and 100 mm. For this research we will use the small tube, thus having a diameter of 29 mm. This tube is able to deliver accurate results between 500 and 6400 Hz. The large tube has a measurement range of 50 to 1600 Hz. The precision of the measurements is 1 Hz. The measurement microphones are two 1/4" condensor microphones type 1487 (Bruel and Kjaer). Before measuring the samples, a calibration is done by using an included calibration sample. A schematic diagram of the impedance tube is given in figure 3.5. The full measurement setup is displayed in figure 3.6.

Figure 3.5: Schematic diagram of the impedance tube

Figure 3.6: Measurement setup

\(^2\)Range(100,10,6400) means: Startpoint is 100 Hz, endpoint is 6400 Hz, steps are 10 Hz.
3.3.3. Validation
Validation of the measurements is done by testing the elements in the large tube, giving a range between 50 and 1600 Hz. The fitting elements that were made for these tests consist of three 6 mm thick MDF plates that are laser cut to exact size. We sealed the elements to prevent undesired resonances in the cavities. A second validation is done in the Kundt's tube that is present at Eindhoven University of Technology (TU/e). Since this tube has a diameter of 40 mm, the fitting elements have a smaller effect on the results. Fifty percent of the samples are tested. The results will be presented in section 4.7 Validation and are added to the figures of the corresponding measurements in Delft.

Figure 3.7: Kundt's tube at Eindhoven University of Technology

The Kundt's tube uses six microphone positions to calculate the absorption coefficient of the sample. It generates a sweep, and measures the impulse responses that are afterwards post-processed in the Matlab script presented in Appendix G.7.

3.4. Performance coefficients
An important part of the analysis is the evaluation of different samples. Because of a larger variety of geometrical features, it is important to introduce parameters that can evaluate the performance of the different samples. Therefore two performance parameters are introduced, that give information on the bandwidth and total absorption of the sample.

3.4.1. Bandwidth calculation (Q)
As an example for the bandwidth calculation, we use the results from Squares 0.1 mm. In section 4.10 we can see the peak frequency and the frequencies that give us 50% of the peak absorption. The bandwidth is defined as the frequency range between these 50% absorption values. For the quality of the bandwidth, we introduce a relation between the width and the height of the peak. This relation is similar to the Q-factor calculation of speaker system, where the quality is defined by the frequency range of a 3 dB drop on both sides of the cut-off frequency. In this case we will multiply the bandwidth with the height of the peak. This means that a larger Q equals a better performance. For our frequency range (125-4000 Hz), a maximum Q of 3875 can be obtained. This would equal $\alpha = 1$ for all frequencies.

$$Q = BW \cdot \alpha_{\text{peak}}$$ (3.1)

3.4.2. Total absorption (B)
A different approach to the quantification of the results is to look at the total absorbed energy by the system. This can be done by integrating the absorption coefficient curve, and thus calculating the total amount of absorption or the total amount of energy lost to the system. This calculation is done using the trapz-function in Matlab and is calculated for two frequency ranges. First the range that is focus of the research: 125-4000 Hz. Second the calculation is done for 250-2000 Hz. This is due to several measurement errors between 2000 and
4000 Hz that could influence the results. These performance coefficients are then plotted against relevant geometrical data, such as length or surface area of the neck.

\[ B = \frac{4000}{125} \int \alpha \cdot d(f) \]  

(3.2)

Figure 3.8: Schematic representation of the performance coefficient calculations
In this chapter the results of the research are presented. The methodology presented in the previous chapter will be the guideline for the results, showing the different geometrical samples and their most relevant results. After showing the results from the different samples, the focus shifts to the combination and implementation of these elements.
4.1. Normal

Based on equation 2.19, the resonator is designed such that the resonating frequency is around 600 Hz, making it possible to measure in the small impedance tube. The normal Helmholtz resonator will be the baseline for the results and eventually will be used to evaluate the improvements. The dimensions of this resonator are:

- Neck radius: 4 mm
- Neck length: 11 mm
- Volume depth: 39 mm
- Volume radius: 13.5 mm

The goal is to set a baseline for the performance and validate the model, because the performance and analytical equations are known. The results for the absorption coefficient of the analytical, numerical and measured resonator are presented in the following figure.

In figure 4.2 we see the analytical, numerical and measured results. The peak frequency ranges between 623 and 625 Hz for the different calculations. The numerical and analytical solution show very similar results in both the peak frequency and absorption coefficient. The measurements in Eindhoven and Delft also have a similar behaviour, with a small deviation in absorption coefficient for the absorption peak. All the results are combined in table 4.1. We notice a constant difference between the numerical and measured results, in this measurement and the following. This will be discussed in detail in chapter 7 Discussion.

We will focus on the curve around 624 Hz. The high peak at 4522 Hz is related to the depth of the volume, giving a standing wave with a wavelength of 78 (= 2 x 39) mm. This peak is visible at all measured results and will therefore be neglected. Measurements of the normal Helmholtz resonator also show a peak at 2703 Hz. Other figures show some additional peaks around this frequency, all related to the standing waves in the sample. Therefore these peaks will be neglected in the results. The bandwidth for the normal Helmholtz resonator equals 360 Hz.
Table 4.1: Combined results normal Helmholtz resonator

<table>
<thead>
<tr>
<th></th>
<th>Analytical</th>
<th>Numerical</th>
<th>Measured</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>f [Hz]</td>
<td>α [-]</td>
<td>f [Hz]</td>
<td>α [-]</td>
<td>B (125-4k)</td>
</tr>
<tr>
<td>Normal</td>
<td>625</td>
<td>0.243</td>
<td>625</td>
<td>0.242</td>
<td>624</td>
</tr>
</tbody>
</table>

Figure 4.3: Velocity profile in the neck of a normal Helmholtz resonator

The velocity profile shows that for this (resonating) frequency, the boundary layer has a thickness of 0.2 mm, as expected seen from figure 2.6. The maximum velocity equals 0.044 m/s. Figure 4.4 shows the dissipation density with a clear dissipation at the boundary of the Helmholtz resonator. The maximum value is achieved at the boundary, with 2.48 [W/m²].

Figure 4.4: Total thermo-viscous power dissipation density for a normal Helmholtz resonator at 625 Hz
4. Results

4.2. Squares 0.5 and 1.0 mm

To research the potential impedance change by flow separation, two types of samples are printed. Besides the absorption coefficient, the focus is on the velocity profiles in the neck, in order to analyse the effect on the impedance of the system. The section shows the geometrical profile. The neck radius varies between 4 mm and 3.5 or 3.0 mm, depending on the size of the squares.

- Neck radius: 3.5 or 3.0 to 4 mm
- Neck length: 11 mm
- Volume depth: 39 mm
- Volume radius: 13.5 mm

The following figures show the analytical, numerical and measured absorption coefficient for the different samples.

The results show the same trend in both the numerical and measured results: the peak frequency shifts 75-100 Hz and there is an increase in the measured absorption coefficient. The results are summarized in table 4.2. There is still a large difference between the measured and both analytical and numerical solution, but the peak frequencies show very good correlation between the different solutions. The analytical curve shows the solution for the normal Helmholtz resonator, as a second reference. The absorption curve bandwidth of the 0.5 and 1.0 mm squares equals respectively 320 and 132 Hz. The 0.5 mm squares are also measured in the Kundt’s tube at the Eindhoven University of Technology. The peak frequency is 545 Hz, and thus deviates only 5 and 10 Hz from the numerical results and Delft measurements. The absorption coefficient itself is lower, and close to the measurement of a normal Helmholtz resonator. Even though both the numerical and measured solution show the shift in resonating frequency and the increase in absorption coefficient, there is still a very large difference between the two solutions.
4.2. Squares 0.5 and 1.0 mm

Table 4.2: Combined results Helmholtz resonator with larger squared geometry

<table>
<thead>
<tr>
<th></th>
<th>Analytical</th>
<th>Numerical</th>
<th>Measured</th>
<th>B</th>
<th>Q</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>f [Hz]</td>
<td>α [-]</td>
<td>f [Hz]</td>
<td>α [-]</td>
<td>(125-4k)</td>
</tr>
<tr>
<td><strong>Square 0.5</strong></td>
<td>557</td>
<td>0.299</td>
<td>550</td>
<td>0.295</td>
<td>555</td>
</tr>
<tr>
<td><strong>Square 1.0</strong></td>
<td>492</td>
<td>0.386</td>
<td>500</td>
<td>0.319</td>
<td>495</td>
</tr>
</tbody>
</table>

The analytical result in the table is an estimation for an equivalent Helmholtz resonator that has a radius equal to the inner radius of this sample (radius 3.5 resp. 3.0 mm) and similar volume in the cavity. It shows a good fit in terms of the resonating frequency. With the focus being on the impedance, we will also look at the velocity profiles in the Helmholtz resonator. Figure 4.7 shows the velocity profile in the Helmholtz resonator with 0.5 mm squares. The common velocity profile is recognizable, with 0 m/s at the boundary and a high increase close to the boundary. The bulk velocity equals 0.053 m/s. Figure 4.7 shows the difference in velocity for the two different diameters. The main difference is related to the increase in velocity, especially showing the low velocities in the voids of the geometry. This is also illustrated in figure 4.8.

Figure 4.7: Velocity in Helmholtz resonator with squared geometry, vertical plane

Figure 4.7 shows the local velocity profile for the Helmholtz resonator neck. The two lines represent a section through the squares and in between, thus having a radius of 3.5 and 4 mm. The inner radius shows higher velocity values within the boundary layer.

Figure 4.8: Velocity in the neck of a squared Helmholtz resonator, horizontal plane

The velocity in between the different squares builds up slowly. The maximum velocity is always achieved in front of the smallest radius. The boundary layer thickness is 0.2 mm.
4.3. Squares and triangles 0.1 and 0.25 mm

To investigate the potential of increasing the boundary layer thickness, two different designs are made with small geometries. Both the squared and triangular geometry have been printed in 0.1 and 0.25 mm. The radius of the neck now varies between 4 mm and 3.9 or 3.75 mm. A section of the samples is given below. Figures 4.10 and 4.11 show the absorption coefficient curves. Table 4.3 combines all the gathered data.

- Neck radius: 3.9 or 3.75 to 4 mm
- Neck length: 11 mm
- Volume depth: 39 mm
- Volume radius: 13.5 mm

Figure 4.9: Cross sections of a Helmholtz resonator with squared and triangular geometry

Figure 4.10: Absorption coefficient curves for Helmholtz resonators with squared geometry (0.1 and 0.25 mm)

<table>
<thead>
<tr>
<th>Frequency [Hz]</th>
<th>Absorption coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>0.51384</td>
</tr>
<tr>
<td>250</td>
<td>0.23046</td>
</tr>
<tr>
<td>500</td>
<td>0.18122</td>
</tr>
<tr>
<td>1k</td>
<td>0.41182</td>
</tr>
<tr>
<td>2k</td>
<td>0.26564</td>
</tr>
<tr>
<td>4k</td>
<td>0.14554</td>
</tr>
</tbody>
</table>

Absorption coefficient - Square 0.1mm

<table>
<thead>
<tr>
<th>Frequency [Hz]</th>
<th>Absorption coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>0.57555</td>
</tr>
<tr>
<td>250</td>
<td>0.32698</td>
</tr>
<tr>
<td>500</td>
<td>0.21814</td>
</tr>
<tr>
<td>1k</td>
<td>0.40508</td>
</tr>
<tr>
<td>2k</td>
<td>0.22645</td>
</tr>
<tr>
<td>4k</td>
<td>0.23850</td>
</tr>
</tbody>
</table>

Absorption coefficient - Square 0.25mm

Figure 4.11: Absorption coefficient curves for Helmholtz resonators with triangular geometry (0.1 and 0.25 mm)

<table>
<thead>
<tr>
<th>Frequency [Hz]</th>
<th>Absorption coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>0.39434</td>
</tr>
<tr>
<td>250</td>
<td>0.21814</td>
</tr>
<tr>
<td>500</td>
<td>0.35086</td>
</tr>
<tr>
<td>1k</td>
<td>0.39434</td>
</tr>
<tr>
<td>2k</td>
<td>0.21354</td>
</tr>
<tr>
<td>4k</td>
<td>0.39434</td>
</tr>
</tbody>
</table>

Absorption coefficient - Triangle 0.1mm

<table>
<thead>
<tr>
<th>Frequency [Hz]</th>
<th>Absorption coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>0.51384</td>
</tr>
<tr>
<td>250</td>
<td>0.23046</td>
</tr>
<tr>
<td>500</td>
<td>0.18122</td>
</tr>
<tr>
<td>1k</td>
<td>0.41182</td>
</tr>
<tr>
<td>2k</td>
<td>0.26564</td>
</tr>
<tr>
<td>4k</td>
<td>0.14554</td>
</tr>
</tbody>
</table>

Absorption coefficient - Triangle 0.25mm
The measurements presented in figures 4.10 and 4.11 show very similar behaviour, equal to or less than the normal Helmholtz resonator (both measured and analytical). Interesting is the fact that using 0.5 and 1.0 mm squares very effectively lowers the resonating frequency and increases the absorption coefficient, but 0.1 mm squares increases the resonating frequency (slightly) and decreases the absorption coefficient curve. Using 0.25 mm squares does lower the resonating frequency to 607 Hz, but also lowers the absorption coefficient. The same trend is seen at the numerical results from Comsol Multiphysics. For the triangular shapes, the results also differ from the expectations. The deviating solutions for the triangular shapes can be explained by the fact that these shapes were created not by adding triangles, but by cutting out the voids. This results in an increase in mass and therefore increasing the resonating frequency and lowering the absorption coefficient. This however does not explain the extreme decrease of the absorption coefficient for the 0.1 mm triangles. The absorption curve bandwidth of the of 0.1 and 0.25 mm squares equals respectively 356 and 314 Hz. The absorption curve bandwidth of the 0.1 and 0.25 mm triangles equals respectively 358 and 327 Hz.

Table 4.3: Combined results Helmholtz resonator with small squared and triangular geometry

<table>
<thead>
<tr>
<th></th>
<th>Analytical</th>
<th>Numerical</th>
<th>Measured</th>
<th>B (125-4k)</th>
<th>Q (250-2k)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Square</td>
<td>f [Hz] 608</td>
<td>a [-] 0.25</td>
<td>f [Hz] 625</td>
<td>a [-] 0.226</td>
<td>f [Hz] 645</td>
</tr>
<tr>
<td>Square</td>
<td>f [Hz] 589</td>
<td>a [-] 0.267</td>
<td>f [Hz] 575</td>
<td>a [-] 0.239</td>
<td>f [Hz] 607</td>
</tr>
<tr>
<td>Triangle</td>
<td>f [Hz] 625</td>
<td>a [-] 0.243</td>
<td>f [Hz] 625</td>
<td>a [-] 0.218</td>
<td>f [Hz] 630</td>
</tr>
<tr>
<td>Triangle</td>
<td>f [Hz] 625</td>
<td>a [-] 0.243</td>
<td>f [Hz] 625</td>
<td>a [-] 0.213</td>
<td>f [Hz] 624</td>
</tr>
</tbody>
</table>

The analytical result in the table is an estimation for an equivalent Helmholtz resonator with similar mass in the neck (radius 3.9 resp. 3.75 mm) and similar volume in the cavity. The analytical result for the triangular shapes is equal to the normal Helmholtz resonator, since the inner circle of the elements is 4.0 mm. The shapes are created by cutting out the voids, rather than adding the geometries. The effect on the boundary layer can be seen when looking at the data from Comsol. The data presented is from the squared geometry. The Comsol results from the triangular shapes were very similar and will therefore be not discussed into detail. These results are presented in Appendix D.

Figure 4.12: Velocity in the neck of a squared Helmholtz resonator

Figure 4.13 shows that the maximum velocity is reached within 0.2 mm of the inner radius of the neck. We can also clearly see that the velocity within the squared voids is close to zero. The maximum velocity equals 0.05 m/s, with a bulk velocity of 0.048 m/s.
4. Results

Figure 4.13: Velocity profile line at 625 Hz

4.4. Fins

As a result from previous samples, the focus has turned into the increase of surface area for boundary layer formation. The goal of finned structures is to investigate the potential of surface increase and the effect that the additional loss of energy has on the absorption coefficient curve. This section will cover the resulting absorption coefficient curves, the velocity profiles and the use of the thermo-viscous energy dissipation, related to the energy loss in the boundary layer. All the results will be combined in table 4.4.

- Neck radius: 4 mm
- Neck length: 11 mm
- Volume depth: 39 mm
- Volume radius: 13.5 mm
- Fin thickness: 1 mm

Figure 4.14: Cross sections of a Helmholtz resonator with fins

Figure 4.15: Absorption coefficient curves for Helmholtz resonators with additional fins in the neck (2 and 4)
Figure 4.15 shows the analytical, numerical and measured solution for the finned resonators (and normal resonator for comparison). The measured results are higher than both the analytical and numerical equations. The numerical solutions are calculated with low accuracy due to the available computational capacity and must therefore be seen as an indication. The frequency steps for the numerical calculation are 25 Hz, and the results range between 300 and 1000 Hz. Even with the low accuracy they show the shift in frequency. The validation measurements in Eindhoven have a larger frequency shift (100-110 Hz) than the measurements in Delft (55-70 Hz). The absorption curve bandwidth of the 2 and 4 fins equals respectively 359 and 278 Hz.

Table 4.4: Combined results Helmholtz resonator with fins

<table>
<thead>
<tr>
<th></th>
<th>Analytical</th>
<th>Numerical</th>
<th>Measured</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>f [Hz]</td>
<td>α [-]</td>
<td>f [Hz]</td>
</tr>
<tr>
<td>Fins 2</td>
<td>565</td>
<td>0.262</td>
<td>600</td>
</tr>
<tr>
<td>Fins 4</td>
<td>534</td>
<td>0.294</td>
<td>550</td>
</tr>
</tbody>
</table>

The analytical result in the table is an estimation for an equivalent Helmholtz resonator with similar mass in the neck (radius 3.75 resp. 3.5 mm) and similar volume in the cavity. Calculated parameters B and Q are presented in the table and show the increase in amplitude. The potential is furthermore investigated in detail within the Comsol software, presented in the following figure (4.16).

Figure 4.16: Total thermo-viscous dissipation power density for a 2- and 4-finned structure at peak frequencies

The figure shows the dissipation density for both the thermal and viscous dissipation, combined. For the 2 finned structure, the general density lies around 0.04 W/m$^3$ in the boundary layer, with local extremes at 0.11 W/m$^3$. For the 4 finned structure, the general density lies around 0.06 W/m$^3$ at the outer boundaries, but around 0.02 at the inner boundaries, with local extremes showing at 0.19 W/m$^3$. What is interesting is that the numerical results show a larger dissipation around the outer edges and only a slight dissipation next to the fins. This is most likely related to the velocity profiles in figures 4.17 and 4.18. The locations where less energy dissipation is obtained are also the regions where the velocity is increasing very fast and does not reach zero at the boundary, related to the limitations of the mesh size.

The velocity profile figures show the thickness, increase and differences in boundary layers. Interesting differences are obtained in the angled corners of the neck, where larger boundary layers are present and even reversed flows occur, indicating possible flow separation near the boundary. The bulk velocity reaches a speed of 0.064 m/s for the 4 fins and 0.054 m/s for the 2 fins.
Figure 4.17: Velocity profile in the neck for a 4-finned structure at peak frequency 550 Hz

The velocity figures show that at the sharp corners the velocity profile is rounded, giving back a strong relation to the slightly rounded curve at the thermo-viscous power dissipation density figure (4.16). The same can be seen in the 2-finned structure figures.

Figure 4.18: Velocity profile in the neck for a 2-finned structure at peak frequency 600 Hz

4.4.1. Extended fins

Additionally the possibility of the extended fins has been researched. Due to the lack of time and computational capacity these samples have only been measured for the full frequency range. Based on that result a few single frequency calculations are done in Comsol Multiphysics. Two different types of extended fins are researched: (1) only extending the fins and (2) extending the fins and neck boundary.

- Neck radius: 4 mm
- Neck length: 11 mm
- Volume depth: 39 mm
- Volume radius: 13.5 mm
- Fin thickness: 1 mm
- Fin (and neck) extension: 3 and 6 mm
4.4. Fins

As can be seen from figure 4.20, solely extending the fin gives an equal performance to a (not extended) finned structure. This holds for both the 3 and 6 mm extension. Related to this result, a figure of the velocity flow within the Helmholtz resonator is added. One can clearly see the velocity decreasing rapidly after entering the volume of the resonator. The bandwidth of the absorption coefficient curve equals the 4 finned structure without extension, i.e. 278 Hz.

Figure 4.20: Absorption coefficient curve and velocity profile in the neck for a 4-finned extended structure

The effect of extending both the fins and the neck of the Helmholtz resonator results in a frequency shift due to a change in neck length and volume reduction. The main goal of this additional research is to investigate the trend of bandwidth enhancement due to surface area increase. The main result needed to conclude from this is the performance parameter $Q$, given in table 4.5. The absorption curve bandwidth of the 3 mm and 6 mm extended fins and neck equals respectively 324 and 286 Hz.

Figure 4.21: Effect of extending the fins and the neck boundary
Table 4.5: Combined results Helmholtz resonator with extended fins

<table>
<thead>
<tr>
<th></th>
<th>Analytical</th>
<th>Numerical</th>
<th>Measured</th>
<th>B (125-4k)</th>
<th>Q (250-2k)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fins 4 + 3 mm</td>
<td>565</td>
<td>0.262</td>
<td>555</td>
<td>0.667</td>
<td>260.6</td>
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<tr>
<td>Fins 4 + 6 mm</td>
<td>534</td>
<td>0.294</td>
<td>548</td>
<td>0.684</td>
<td>287.5</td>
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<tr>
<td>Fins 4 + full 3 mm</td>
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<td>0.422</td>
<td>490</td>
<td>0.937</td>
<td>603.8</td>
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<td>Fins 4 + full 6 mm</td>
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<td>0.463</td>
<td>450</td>
<td>0.933</td>
<td>534.5</td>
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</table>

The analytical result in the table is an estimation for an equivalent Helmholtz resonator with similar mass in the neck (radius 3.5 mm) and similar volume in the cavity. There are no numerical results for the extended fins calculations. Because of the computational capacity it requires to calculate this, the calculations can only be done with very coarse mesh sizes and the results are not reliable. The mesh size has a large influence on the results.

4.5. Curved orifice

The potential for enlarging the surface area is furthermore researched by curved orifices, where the mass of the resonator equals the mass of a normal Helmholtz resonator. The total power dissipation density should increase, causing better and/or broader absorption. The geometry has the following dimensions:

- Neck radius: varies, but mass equivalent 4 mm
- Neck length: 11 mm
- Volume depth: 39 mm
- Volume radius: 13.5 mm

Figure 4.22: Cross sections of a Helmholtz resonator with curved orifice surface

The curved shape has smooth edges to reduce effects that we have observed at the finned structures. The goal is to increase the amount of boundary layer formation without sharp corners and loss of mass (thus not adjusting the resonating frequency). The increase of surface area is resp. 17% and 39%. The results for the absorption coefficient curve measurements are presented in the figure 4.24. The absorption coefficient increases with the increase in surface area and the figures show a bandwidth of 349 and 351 Hz respectively, compared to a bandwidth of 360 Hz for the normal resonator. Figure 4.23 also shows an indication for the numerical solution of the Curved 1 sample, with 17% increase in surface area. The results show the trend of the absorption coefficient curve, with similar resonating frequency, but a very low absorption coefficient peak. This could be caused by the mesh size, and must therefore be seen as an indication.
Some fine mesh single frequency calculations are done in Comsol and presented below. Figure 4.24 shows the velocity profile for the curved orifice designs. Both have a bulk velocity speed of around 0.048 m/s, which is almost equal to the speed of a normal Helmholtz resonator (0.044 m/s). The thermo-viscous power dissipation density figures are shown in figure 4.25.
The power dissipation density figures show a lot of similarities to the normal Helmholtz resonator, but also immediately show the reduction of dissipation on inner parts of the geometry (figure 4.25). This is also observed at the finned structures, where the dissipation mainly is noticed at the outer part of the geometry as well. The maximum dissipation density is again obtained at the boundary and reaches values of 1.86 and 1.44 W/m³ for the two geometries.

Table 4.6: Combined results Helmholtz resonator with curved orifice

<table>
<thead>
<tr>
<th></th>
<th>Analytical</th>
<th>Numerical</th>
<th>Measured</th>
<th>B (125-4k)</th>
<th>Q (250-2k)</th>
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</thead>
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<td></td>
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<td>α [-]</td>
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<tr>
<td>Curved 1</td>
<td>625</td>
<td>0.243</td>
<td>625</td>
<td>0.161</td>
<td>624</td>
</tr>
<tr>
<td>Curved 2</td>
<td>625</td>
<td>0.243</td>
<td>-</td>
<td>-</td>
<td>624</td>
</tr>
</tbody>
</table>

The analytical result in the table is an estimation for an equivalent Helmholtz resonator with similar mass in the neck (radius 4.0 mm) and similar volume in the cavity.

### 4.6. Performance coefficients

As presented in section 3.4 on page 26, the performance parameters are a way of analyzing the results for the different samples. The results for B and Q are presented in the following figures. These figures show the relation between geometrical features such as the size of the squares or triangles and the calculated Q and B coefficients. The lower figure shows the relative effective surface area in relation to Q and B. The effective surface area is calculated by using only the surface area that is directly induced by the sound wave, i.e. only the inner part of the squared geometry. For the fins this is the total surface area of the neck. These surface areas are presented relative to the normal Helmholtz resonator.

(a) Performance coefficients for squared geometry

(b) Performance coefficients for triangular geometry
Especially for the Q-factor we notice the increase from Normal to Fins 2, and after that a large drop in the values. In general both the parameters Q and B follow the same trend.

4.7. Validation

The results for the validation in Eindhoven are given in the figure 4.28. These figures are also implemented in the individual figures in previous sections. The figure shows the results for the 40 mm tube at Eindhoven, normalized to the 29 mm tube in Delft that has been used for the measurements. The figure show interesting results, with very equal performance. There are some frequency shifts noticeable due to differences in neck radius, but over all there are no significant changes. For the normal Helmholtz resonator we have seen at figure 4.2 similar resonating frequencies and comparable absorption coefficients. For the other samples the absorption coefficients deviate more, and there are some shifts in frequency obtained. Additionally, some measurements are done in the large impedance tube in Delft. Fitting elements that were made for this tube generated too much side effects to use the results for validation.

Figure 4.28: Validation results at Technical University Eindhoven
4.8. Tuning

Because of the large dependency on the geometry, a Helmholtz resonator can easily be tuned to the desired frequency. It can be adjusted in terms of neck length, neck radius and total volume. These geometry related adjustments are investigated and their effect on the resonating frequency is displayed in the figures below. The starting point is always the Helmholtz resonator used in the research, with the following dimensions.

- Total depth: 50 mm
- Neck length: 11 mm
- Neck radius: 4 mm
- Volume depth: 39 mm
- Volume radius: 13.5 mm

Using the quick resonating frequency calculation as presented in equation 2.19, this resonator gives us a resonating frequency of 614 Hz. The first figure, showing the effect of the volume depth shows us that a larger volume gives a lower resonating frequency. This can be explained with the mass-spring system, where a larger volume equals less spring stiffness. Interesting in this case is that the maximum depth is set at 50 mm, showing the link between the mass gain in the neck and the spring stiffness of the volume.

![Effect of volume depth on resonating frequency](image1)

(a) Resonating frequency as function of volume depth

![Effect of volume radius on resonating frequency](image2)

(b) Resonating frequency as function of volume radius

Figure 4.29: The effect of geometry changes on resonating frequency of a Helmholtz resonator

![Effect of neck radius on resonating frequency](image3)

Figure 4.30: Resonating frequency as function of neck radius
Combining elements  When combining different Helmholtz resonators one can not simply add up the different absorption coefficients. The area over which this absorption is spread decreases the efficiency drastically. This absorbing cross section (equation 2.24) must be taken into account when creating absorbing elements. For example, the absorbing cross section of a Helmholtz resonator that is tuned to 600 Hz equals 0.026 m², which corresponds to a circle with a radius of 90 mm. These 600 Hz tuned elements must have a spacing of 90 mm to avoid interference. However, a broadband absorber is often more valuable than specific frequency absorption. Therefore we need to combine several elements with different resonating frequencies. Figure 4.31a shows three measured elements and the curve that would be the result of a combination. Figure 4.31b shows a theoretical approach, with three resonators that are tuned to three different frequencies: 350, 425 and 575 Hz and their combined absorption coefficient curve. Note that this is purely a theoretical approach and no testing has been done on these results. The results should be validated and tested properly before applying in practice.

(a) Combination of three measured samples  (b) Combination of three theoretical Helmholtz resonators

4.9. Integration

In section 2.3 the advantages and future challenges are discussed. The focus has been on the general investigation of resistance. Being at the starting point of what has potential, the complexity and design optimisation is still very low. Within the research the possibility of adapting to previous results show the potential of responsive production. Together with a short supply chain and low volume production, these advantages has definitely been observed and resulted in the possibility of quick adaptation and implementation of new findings. Reliability and accuracy is one of the main challenges that has been observed during this research. A strong focus is on a very small physics scale, giving a high level of importance on this challenge. Due to this required quality of the prints, the samples are produced professionally, using stereolithography, allowing for a 0.1 mm accuracy. The results show that the geometrical features do not influence the boundary layer and thus the resonating performance. The following figure shows the connections between the potentials, important acoustical goals (section 2.3.1) and four of the most used techniques, as described in appendix B. It shows the most important relations that are resulted knowledge based on the findings from this research.
This research has tackled the challenge of investigating the potential of acoustic performance improvements by additive manufacturing. This chapter contains the conclusions to the results and research questions as formulated in Section 1.3 on page 3.
5.1. Sub conclusions

Impedance The results for the impedance related geometries (squares 0.5 and 1.0 mm) show a clear shift in the peak frequency that can be explained with the reduction of the air mass in the neck. This reduction effectively lowers the resonating frequency and is confirmed by looking at the analytical results for the reduced radius. From this we can conclude that it is not possible to increase absorption in the Helmholtz resonator neck by inducing impedance change, using geometries perpendicular to the main flow.

Boundary layer resistance Boundary layer resistance can be increased by using additional elements in the direction of the flow. The use of small squared and triangular geometry (0.1 and 0.25 mm), that are mainly perpendicular to the flow do not show significant increase of sound absorption, with similar reasoning as the larger geometrical structures, discussed in the previous paragraph. The boundary layer is in both samples equal to 0.2 mm, which is what we expected for that resonating frequency range, thus we can conclude that the thickness of the boundary layer cannot be adjusted by using geometry in the similar size. There is a small frequency shift noticeable and the amplitude of the absorption coefficient curve show similarities to a normal Helmholtz resonator.

Surface area increase The use of finned structures to increase the surface area over which the boundary layer can be formed shows potential. The additional research on extended fins shows no improvement in the results, caused by the diversion of the flow when entering the main volume of the Helmholtz resonator. Additional surface area only works in regions with higher velocities. The thermo-viscous dissipation power density is used as a parameter to communicate the total amount of energy that is dissipated by thermal or viscous damping. This has a relation to the surface area and can be used to optimize the shape to increase the resistance. As we can see from figure 4.16, corners are not always beneficial and the outer curved geometries allow for more energy dissipation. Knowing that the Helmholtz resonators with curved orifice have similar resonating frequencies and equivalent dimensions, it is likely that the increase in absorption coefficient in figure 4.23 on page 41 is the result from the increase in surface area.

Numerical approximation By modelling the impedance tube in Comsol Multiphysics we have been able to provide a valuable tool that can simulate the measurements and predict the trend of the absorption coefficient curve. Figure 4.2 shows a strong correlation between the numerical and analytical equation. Comparing it to the measured results, the resonating frequency can be predicted, but the height of the figures do not match.

Performance coefficients Two design parameters are suggested to give a guideline on the improvement of sound absorption design. The Q-factor and total absorbed energy B are graphically expressed against geometrical features such as size or surface area. It is likely that eventually a trend can be set at the increase of surface area from the samples Normal to Curved 1 and Curved 2. At this moment the amount of samples is too low to predict a trend.

Additive manufacturing The current research does not fully engage all the potentials of additive manufacturing. From the results on the boundary layer, where the use of small geometrical features are discussed, it is likely that a lower accuracy does not decrease the performance of the Helmholtz resonator.

5.2. Final conclusions

What is the effect of adding resistance to a Helmholtz resonator? The way that the resistance is added plays an important role in the effect of its performance. The resistance is effective when placed as an increase in surface area in the direction of the flow. The effect can then be seen in both the increase of absorption and broadening of the bandwidth of the absorption curve. Within this research that has been done by adding vertical fins into the neck of the Helmholtz resonator and creating curved orifices that have a significant surface area increase. These last samples show the potential of increasing the surface area while maintaining the resonating frequency and are likely to be the reason for the increase in absorption coefficient. Additional research needs to be done to validate the current results and determine a significant trend in order to use this in future design stages.
Can we use this to improve its performance and create a broadband absorber made with additive manufacturing?  The final results seem promising for the use in creating a broadband absorber. Current individual resonators show broad absorption curves and the increase in surface area can be a great tool in enhancing the performance. Combination of these elements with the right target and spacing can have good absorption characteristics for low and mid frequencies. For high frequencies the potential most likely lies with porous materials that might be possible to incorporate in the current structures. The application into practice is for now focussed on specific room acoustic issues. By pointing out the problems with measurements, the absorbing element can be customized for each room.
The results and potential of the research will be demonstrated with a case study, being a small meeting room at the faculty of Architecture and the Built Environment. The room will be measured and modelled in Odeon, showing the room characteristics and possible areas of improvement. Using this input that is related to the specific room characteristics, we can design an absorbing element based on the previous research. This chapter explains the calculation and results of the implementation of sound absorbing panels.
6.1. Introduction
The room used for the case study is meeting room 1.WEST.060, at the faculty of Architecture. It is used as a meeting space and must therefore be optimized for speech. The room has a floor surface area of 46 m² and a volume of 263 m³. The floor and ceiling have sound absorbing finishes, but most of the walls are finished with a plaster, causing clearly hearable flutter echoes between two parallel surfaces. The absorbing panel shown on figure 6.1a has been a first step to improve the situation but turns out to not be sufficient.

![Meeting room with absorbing panel](image1)

![Highly absorptive ceiling panels](image2)

Figure 6.1: The room for the case study, 1.WEST.060, faculty of Architecture

Based on the use and dimensions of the room, the final goal would be to achieve a reverberation time of 0.8 seconds. This confines with the upper limit of the Dutch building regulation code, which would be acceptable for a larger volume meeting space. For a good speech intelligibility a STI of at least 0.6 should be achieved, giving a STI ranking of good or excellent when it reaches 0.75 or higher. The room characteristics are determined using calculations, measurements and acoustical software. To get a first understanding of the potential reverberation time, a calculation spreadsheet is used. A large part of the process is done using measurements and modelling software to show the effect of different locations of the absorbing panel.

6.2. Current situation measurements
The measurements will determine the current state of the room and identify certain characteristics that need to be improved. The measurements are performed according to ISO 3382-2: Measurement of room acoustic parameters - Part 2: reverberation time in ordinary rooms, using an omnidirectional source. With the use of an impulse (section 2.2 on page 14) generated by a sweep, we have been able to derive the reverberation time and speech intelligibility (STI) in the room. For the two microphone positions, 3 and 2 receiver positions are chosen. Each measurement consists of three sweeps. The measurement locations are equal to the source and receiver positions in Odeon (figure 6.3b on page 54). Already several measures have been taken to improve the acoustics of the room, such as the use of an absorbing panel on the wall and high performance ceiling panels. The measurement results give the T20, T30 and STI of the different receiver positions.

Using a spreadsheet, the estimated reverberation time is calculated based on the different theories. The hand calculation shows that the reverberation time theories give different results for the room. In most quick calculations, Sabine's equation is used because of its easy implementation and often acceptable results. We are dealing with a box-like volume, with no equal distribution of sound absorbing finishes. For example, the
6.3. Design of absorbing panel

The room has a significant amount of absorption in the Z-axis, but very little in X and Y. Due to this reason, we are mainly looking at the results from Arau-Puchades (Arau-Puchades and Berardi, 2015). Figure 6.2 shows the result for the different types of reverberation time equations and how these align with the measurements. See appendix E for the full calculation spreadsheet. For explanation of the theory, see Chapter 2: Background.

Figure 6.2: Theoretical results and measured reverberation time

The measurements show the issue of having an unequal T20 and T30 and therefore a double-decay in the reverberation time graph. This double-decay shows that the reverberation time is different for different axis. Also, looking at the T30 for both measurement locations, a clear increase in reverberation time can be seen at the 500 and 1000 Hz octave bands. This shows that there are certain places that have a lack of absorbing material where the sound can constantly reflect.

Figure 6.2a shows the different ways to calculate the reverberation time by hand. We see that none of them can really account for the differences in the reverberation time between T20 and T30 and the flutter echoes that appear in a room like this. The result of Arau-Puchades theory fits the measurements the best. Based on the size of the room and location of the sound absorbing material (mainly on floor and ceiling) it is important to incorporate this distribution in the reverberation time equation. The measurements show a clear distinction between the T20 and T30, telling us there is an unequal reverberation time in the different axes (the double-decay), resulting in a high chance of flutter echoes in the room if two parallel walls lack sound absorption. Flutter echoes cannot be taken into account with the theoretical models and needs ray-tracing software to determine or solve. Therefore we will further look at the calibrated model. This will be discussed in section 6.4. The measurements also show a very high reverberation time for low frequencies. This is caused by standing waves, since the Schroeder frequency for this room is 109 Hz. Below this frequency there will be distinctive modal behaviour. Above the Schroeder frequency it is accepted to assume reverberant room behaviour.

Looking at both the measurements and spreadsheet calculations, the focus should be on decreasing the double-decay and thus creating a evenly distributed absorption within the room with a specific target between 500 and 1000 Hz.

6.3. Design of absorbing panel

Based on the problems and the measurements it is possible to tune the absorber for the specific room acoustic issues. The room improvement will focus on decreasing the reverberation time at 500 and 1000 Hz and if possible, increase the STI. The flutter echoes will not be addressed since it will be difficult to model in Odeon and is mainly excited with impulses. With the main use of the room being for speech, it is less of a problem. The panel consists of a combination of three resonators from the measurements within this research. Figure 4.31a shows the combination of three different resonators, that gives a panel tuned to 555 Hz. The sound
Case study

The absorbing area per resonator equals 0.026 m². Having to deal with this sound absorbing area, a panel of 9 m² can contain 352 resonators. For now the design is a panel, but the resonators can be implemented in any form or shape to create a 3-dimensional object within the room that serves room acoustic improvements. This panel is then implemented in the room at two different locations: on the wall and above the table. The locations can be seen in the pictures 6.5 and 6.6. The results are received from the Odeon model, which is calibrated to the measurement results.

6.4. Room improvement in Odeon

A: Odeon model  

The current situation is modelled in Odeon. The calibration of the model is based on the measurements described in the previous section. After calibration, the panels will be added in the form of 9 m² above the table to research the effect on the speech intelligibility within this area and 9 m² on the wall to potentially reduce the flutter echoes. The calibrated absorption coefficients are presented in table 6.1. The inputs for the model are:

- Omnidirectional source
- Number of late rays: 20000
- Maximum reflection order: 2000
- Impulse response resolution: 1.0 ms
- Transition order: 2
- Number of early rays: 2000 (auto)
- Number of early scatter rays: 100 per image source
- Screen diffraction: actual

The following source and receiver locations are defined. These are equal for both the measurements and the Odeon model. The origin is located in the corner close to the door, bottom left at figure 6.3a.

- S1 (4.73; 5.80; 1.5)
- S2 (3.84; 1.60; 1.5)
- R1 (1.91; 2.23; 1.4)
- R2 (3.74; 1.34; 1.4)
- R3 (0.54; 4.70; 1.4)
- R4 (0.54; 4.70; 1.4)
- R5 (4.97; 5.73; 1.4)

![Diagram](image)

(a) The room modelled in Odeon
(b) The different source and receiver positions

Figure 6.3: Odeon model

The peak at 1 kHz that is clearly shown in the measurements, was not found in the model, but the trend of both the T20 and T30 (and therefore the double-decay) were captured using the following absorbing and scattering coefficients. The reverberation time of the calibrated model is presented in figure 6.4.
6.4. Room improvement in Odeon

Table 6.1: Absorption coefficients used in the Odeon model

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<thead>
<tr>
<th></th>
<th>63 Hz</th>
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<th>Transparency</th>
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Figure 6.4: Reverberation time results after calibration of the model in Odeon

B: Additional sound absorption on the wall

After addition of 9 $m^2$ sound absorption on the wall, the reverberation time has dropped significantly, and a clear drop in the curve (figure 6.7a on the following page) can be seen at 500 Hz, close to the part of the spectrum that holds the resonating frequencies of the different Helmholtz resonators. A better alignment between the T20 and T30 is obtained from these results, showing that there is a better spread of the absorbing materials.

Figure 6.5: Absorbing panels on meeting room wall

C: Additional sound absorption above the table

Placement of the absorbing material above the table does not show the drop in the curve around the 500 Hz octave band but does show a reduction in reverberation time. (figure 6.7b on the next page) This is mainly due to the fact that the 9 $m^2$ panel reduces the room volume and therefore lowers the reverberation time, see equation (2.35) on page 15.
However, the effect of the panel above the table is clearly shown when looking at the speech intelligibility improvement of the room. Figure 6.8 shows the averaged STI values for the different source positions in current state (A) and after improvement with sound absorption on the wall (B) and above the table (C). This has been an average of all the different source and receiver positions (5 measurements in total). Because the sound absorption is closer to the source and the amount of early reflections is increased, an increase in the STI is obtained at all of the source positions.

Figure 6.8: Speech intelligibility improvement differences between B and C
6.5. Conclusions

Using the Arau-Puchades equation for the reverberation time in the meeting room, we have been able to capture the trend of the reverberation time at different octave bands. The measurements are used to calibrate the model to study the effect of the absorbing panels. By knowing the specific room characteristics we were able to develop a sound absorbing object that is tuned for this meeting room. Both options (B and C) show an improvement in reverberation time, coming from the amount of absorbing surface area added to the model. The big differences however are visible when comparing the T20 and T30. Adding sound absorbing material on the wall brings the T20 and T30 closer together, showing that the double-decay is reduced and a better spread of sound absorbing material is achieved. Placing the element above the desk decreases the volume of the space that is used for the reverberation of the room and mainly uses this to decrease both the T20 and T30. The large difference around 500 Hz is not noticeable here. However, an increase in STI is observed because of the reduction of the reverberation time and the improvement of early reflections around the table. The panel is tuned for a specific range of frequencies and those will be absorbed. The other frequencies will reflect, thus still creating more early reflections.

Concluding back to the research, this case study shows the effect of implementing sound absorbing material for a specific situation. The results from the research measurements have been used. Because of a current strong focus around 600 Hz, the broadband absorption has not been achieved but already shows a significant improvement in room acoustic quality. After further research and frequency tuning, the absorbing elements will have a great potential for specific room acoustic improvement.
This chapter is devoted to the discussion of the current research findings and the used methods. The most important issues will be discussed. After that, the focus is on future research and potential of taking this research and the sound absorption by additive manufacturing a step further.
7.1. Results and research

Differences between measurements and simulations An interesting and important part of the results is the mismatch between the measurements and the numerical and analytical solutions. The significant and constant amplitude difference shows that there are parts of the process that we do not fully understand yet. On the one hand it is a positive result that measured results turn out better performing than the analytical solution would suggest. On the other hand, the measurements have been conducted in a safe and small environment, which will result in different findings than when using it in practice, since having normal or random incidence has an effect on the absorbing behaviour of the sample. Before finding a possible explanation to the difference, it is key that we define what should be taken as the truth. Both measurement tubes (Delft University of Technology, Eindhoven University of Technology) give higher results than the simulations, thus giving confidence in the prepared samples. If we assume the measured results to be true, a possible explanation for the amplitude mismatch is linked to the material properties. We have been using a lightweight plastic with some unknown properties. More importantly, the effect of additive manufacturing on the material is not researched into great detail. This might influence the heat capacity and the way energy is transferred into heat which could be of influence for the isothermal assumptions near the boundary. These material properties are not used in both the analytical and numerical solution. However, the good match between simulations and analytical solutions shows that it is very likely that the simulations confine with the known and used theoretical background and therefore adds value to these results. When we take these results to be true, the high measured results can be caused by the measurement tube. The overall shape of the measurements are relatively high. Note for example the values in low frequencies, but also higher frequencies have a constant absorption coefficient of around 0.5. The range for accurate measurements is between 500 and 6400 Hz, but this vertical 'shift' adds to the uncertainty of the measurements.

Computational capacity and 3D simulations One of the limitations noticed during the research is related to the computational capacity. Due to various reasons, it was not always possible to use a more powerful computer for the Comsol Multiphysics simulations. The 2D axisymmetrical calculations were possible to perform with a sufficiently detailed mesh, but 3D models (such as the finned structures and curved orifice resonators) reached the limits. Due to this reason the mesh size of the elements was not always heading towards the recommended tenth of a wavelength, leaving room for errors in both the volumes and boundary layer regions. The mesh size appeared to have a large influence on the height of the results. The resonating frequency did show stability when comparing different mesh sizes, but the absorption coefficient varied quickly. For the 3D models it has not been possible to do a full sensitivity analysis. Therefore it is suggested to not trust these numerical results with full certainty and see them as an indication for the performance of the Helmholtz resonator.

Quality and accuracy of the samples and measurements Part of the research has been focussed on boundary layer resistance. This has been a challenge due to the accuracy of the additive manufacturing techniques. The accuracy of a lot of available and accessible printing techniques vary around 0.1 mm. With geometrical features and a boundary layer that are in the same order of magnitude, the effect can be disturbed. The quality of the print can have a large influence on the performance and it is not possible to know if the effect is caused by the geometrical adjustments or the printed layer roughness. This adds uncertainty to the results with 0.1 and 0.25 mm geometry. In addition to this, the measurement tube that is used has an accuracy between 500 and 6400 Hz. A lot of the samples have a resonating frequency between 500 and 650 Hz. This results in measuring close to the range and, for bandwidth calculations, using data that is below the 500 Hz boundary. A last note on the measurements has to be made about the different samples with additional geometries. Due to the added fins, squares and triangles, the resonating frequency changes. As seen from previous research, this not only changes the resonating frequency, but also the absorption coefficient peak height and steepness of the curve. Therefore it is hard to draw conclusions on differently tuned resonators. It is difficult to know if the increase in absorption coefficient is due to resistance or due to geometrical adjustments. For future research, it is advised to adjust geometries without changing the main geometry of the resonator, thus keeping the resonating frequency similar.
7.2. Future research and potential

This research has been focused on a potential and innovation, which is something that is always ongoing and leaves questions. If the project will be continued, the following points and questions should be addressed.

**Resistance with surface area increase**  In order to show a significant trend on the increase of surface area for performance improvement, additional research on this topic should be conducted. It is most likely that there is an optimum or trend between the amount of surface area and the cross section or mass of the Helmholtz resonator. This relation can be found by additional research on different and more structured orifice shapes. Within that research it is important to contain the same resonating frequency, mass and main dimensions. The surface area related research can look into different shapes and geometries that might work best for this purpose. From the results within this research we have seen that the outer region of the curved orifice dissipates the most energy. It will be worth looking into a guideline that states the radius or curvature that still allows for this dissipation but does increase the surface area. This can be done analytically (looking into the impedance equations), numerically (using Comsol Multiphysics) and/or empirically (printing and testing).

**Numerical approximation**  As discussed earlier, the amplitude mismatch between the numerical and measured results is something that should be discussed on a physical and material property level. The way the energy is transferred to heat within the boundary layer and the material might possibly be influenced by the additive manufacturing process. First steps within this research could be focused on literature, to examine the exact isothermal process at resonators and how to model this properly using fluid dynamics. Also, the printing process on a material property scale should be examined. What are the exact material properties of a resin based polymer, and how are these influenced after heating and treatment? A next step in research could be using fluid-structure interaction models and modelling the material along with the acoustical behaviour. The good correlation between the analytical equivalent (or in case of the normal resonator - not equivalent but exact solution) and the numerical solution shows that using the fundamentals of physical acoustics we get accurate results. It is interesting to further investigate this potential since it could provide us with the analytical solutions of not common Helmholtz resonator shapes. If this validation can be proven in the future, Comsol Multiphysics provides us with a great tool for resonator analyses.

**Other absorbing principles and implementation**  The results of this research might be applicable for other resonators, such as quarter- or half wavelength tube resonators. Additional research on surface area increases in combination with different resonators could show the potential for a broader range of sound absorption methods. Knowing more about this potential in other mechanisms opens up new possibilities for combinations of absorbing principles and thus implementation within practice. One of the main advantages that require this knowledge is the integration with other disciplines and the application in day to day situations. For additive manufacturing, future research should focus on the complexity and integration with other disciplines in combination with using cheaper and faster techniques such as FDM with lower accuracy to discuss the topic of accuracy.
Bibliography


Appendix A - Acoustical Background

Within the research the focus is on the use of Helmholtz resonators. This has been a choice based on potential of resistance based improvements and was the result of a short research on the different types of sound absorption. This research is included in this appendix.

A.1. Additional room acoustic parameters

D50

Reflections within 50 ms after the direct sound are not noticed as separate sound events. These reflections are adding loudness to the direct sound, and can therefore be very useful in for example theaters or lecture rooms. This is defined in the quantity of the ‘definition’, which is defined by:

$$D_{50} = \frac{\int_0^{50ms} [g(t)]^2 dt}{\int_0^\infty [g(t)]^2 dt} \cdot 100\%$$  \hspace{1cm} (A.1)

With \( g(t) \) being the impulse response between certain time limits. The \( D_{50} \) is the ratio of early energy to the total energy in the impulse response.

C50

The clarity can be defined as the ratio of the impulse response of the room between 0 and 50 ms, to the impulse response of the room between 50 ms and infinity. This equation is often used to show the clarity of speech.

$$C_{50} = 10 \cdot \log_{10} \left[ \frac{\int_0^{50ms} [g(t)]^2 dt}{\int_0^\infty [g(t)]^2 dt} \right]$$  \hspace{1cm} (A.2)

With \( g(t) \) being the impulse response between certain time limits. The \( C_{50} \) is the ratio of early energy to late energy in the impulse response. For musical representations the limit is set at 80 ms.

G

The parameter G is the strength of the sound, measured in decibels and compared to the strength of sound in free field or anechoic room, 10 meter distant from the source.

$$G = L_p - L_{p,\text{anechoic;10m}} = 10 \log \frac{\int_0^\infty [g(t)]^2 dt}{\int_0^{50ms,\text{anechoic}} [g(t)]^2 dt}$$  \hspace{1cm} (A.3)
A.2. Porous absorbers
Porous absorbers are most efficient on mid and high frequencies. This is caused by the wavelength, which is small, and therefore there is less material needed to create an effective absorber. To get optimal results of the friction from the porous material, the sound should not be reflected by the material. The porosity should be large and the airflow resistance of the material should be low, because the transition from air to material has to be smooth. However, it should not be too low, because that would decrease the amount of friction and therefore the amount of absorption.

Important when using porous materials is the thickness of the material. Absorption should take place where the particle speed is largest, i.e. at $\frac{1}{4} \lambda$ from the wall.

![Figure A.1: The particle speed is largest at $\frac{1}{4} \lambda$ from the wall (van der Linden and Zeegers, 2006)](image)

This means the thickness depends on the frequency it should absorb. A logical consequence is that with a thickness as large as possible, most frequencies can be absorbed by the porous material. With placing it a certain distance from the wall, the amount of material can be reduced.

![Figure A.2: Placing the porous material from the wall (van der Linden and Zeegers, 2006)](image)

When a sheet is placed in front (for protection) the following guideline is often implemented. It should have a perforation percentage of at least 20% to not affect the absorbing qualities of the porous material. Besides that, the perforations should have a maximum spacing of 20 mm to prevent the plate from acting as a wall. (van der Linden and Zeegers, 2006) However, this guideline can differ for different applications. Sound loss in a porous material is mainly due to the effect of the viscous and thermal boundary layer. The friction along the sides of the pores cause a layer that is of submillimetre size. Other important causes for sound loss are loss in momentum due to changes in sound path and thermal conduction, which mainly has effect on the low frequencies. Losses due to the vibration of the material are usually less important. The material properties of porous materials are defined by four main quantities: flow resistivity, porosity, tortuosity and characteristic length. (Cox and D’Antonio, 2009)

A.2.1. Flow resistivity
The flow resistivity is a measure of how easily air can enter a porous absorber and the resistance that air flow meets through a structure. (Cox and D’Antonio, 2009, pg 169).

$$\sigma = \frac{\Delta P}{ud} \left[ \frac{N}{m^3 s} \right]$$ (A.4)

With $\Delta P$ being the pressure drop, $u$ the mean steady flow velocity and $d$ the thickness of the material.

The flow resistivity varies most between porous absorbers, and can therefore be of great influence to the final absorption coefficient of the material. Different empirical relations are found for different materials.
A.2. Porous absorbers

A.2.2. Porosity
The porosity $\epsilon$ is the rate of air volume within the total volume, and can therefore have a maximum of 1. For good porous absorbers, such as mineral wool, this value typically is in the order of 0.98. Closed pores do not contribute to the porosity, since they are practically unreachable for sound waves. It is an important parameter, but for most porous materials the value is close to 1 and does not differ a lot. Besides the before mentioned importance of the flow resistivity and the porosity, other quantities such as the characteristic length and pore shape factor can play a significant role in sound absorption.

A.2.3. Characteristic lengths
The characteristic length $\Lambda$ is the weighted ratio of the pore volume to surface area of the pores. (Cox and D’Antonio, 2009, pg 177) For simple pore shapes, the characteristic length can be found with the following equation.

$$\Lambda = \frac{1}{s} \sqrt{\frac{8\nu k_s}{\epsilon \sigma}}$$  \hspace{1cm} (A.5)

With $s$ being a constant between 0.3 and 3, depending on the type of pores, $\nu$ the kinematic viscosity of air, $k_s$ the tortuosity, $\epsilon$ the porosity and $\sigma$ the flow resistivity. Note that these last three are most determining for the effect of this parameter.

The characteristic length $\Lambda$ is the viscous characteristic length. When materials exist of complicated internal structures, it is often necessary to use the thermal characteristic length $\Lambda'$, given by:

$$\Lambda' = \frac{2V_p}{S_p}$$  \hspace{1cm} (A.6)

where $S_p$ and $V_p$ are the surface area of the pores and the volume of the pores. The determination often relies on measurements, since conventional materials have a complex pore shape.

A.2.4. Tortuosity
The tortuosity takes into account the orientation of the pores in relation to the incident sound field, which has an effect on the sound propagation. With a more complex propagation path, the absorption is higher. This complexity is denoted by the tortuosity. A general description of this parameter is given by:

$$k_s = 1 + \frac{1 - \epsilon}{2\epsilon}$$  \hspace{1cm} (A.7)

Because almost no material is perfectly ordered, the tortuosity is a quantity that should be measured. With the possibilities of additive manufacturing, this could change. Knowing the exact geometry, including the porosity of the element, the tortuosity can be calculated.

A.2.5. Modelling porous absorbers
Over time several models were created that describe the theoretical absorption of porous materials. The Biot model is widely accepted as the most accurate, but also the most complex model for predicting the sound absorption by porous materials. Other, slightly simplified models are from Delany-Bazley (1970) or the rigid frame model by Zwikker and Kosten (1949). (Kidner and Hansen, 2008)

The well used Delany-Bazley approximation is based on flow resistivity. During their research they found an empirical fit for the impedance and effective wavenumber. Cox and D’Antonio (2009) describe a semi-analytical approach for modelling porous absorbers. Several difficulties occur, because not all parameters are analytically approachable, such as pore shape related parameters. This could however be possible with additive manufacturing, because of the computer modelling of the material. The parameters that are known are the shape, sizes, volumes and possibly even the exact propagation path.

For predicting $\alpha$ one needs to know the characteristic impedance and (complex) wavenumber. According to the model by Delany and Bazley, the characteristic impedance is given as:

$$z_c = \rho_0 c_0 \left(1 + 0.0571X^{-0.754} - j0.087X^{-0.732}\right)$$  \hspace{1cm} (A.8)

And the wavenumber, $k$:

$$k = \frac{\omega}{c_0} \left(1 + 0.0978X^{-0.700} - j0.189X^{-0.595}\right)$$  \hspace{1cm} (A.9)
Where $\rho_0$ is the density, $c_0$ is the sound speed in air, i.e. 343 $m/s$, $\omega$ is the angular frequency and $X$ is given by:

$$X = \frac{\rho_0 f}{\sigma} \quad (A.10)$$

With $f$ is the frequency and $\sigma$ is the flow resistivity of the fibrous material. The same equation is proven to be useful for other materials as well.

There are however some important restrictions on the application of these empirical relations. The equations are only applicable when the porosity $\epsilon$ is close to 1 and $0.01 < X < 1.0$. This automatically limits the frequency range. Also, the limits of the flow resistivity are $1000 \leq \sigma \geq 50,000$ MKS $\text{rayl}$.

The figure shows the impedance and absorption in porous materials, when determined with the non-dimensional parameter by Delany-Bazley. An increase in $X$ shows an increase in absorption coefficient $\alpha$. An interesting point is that the absorption cannot reach unity over the range for which the equations are valid. During the experiments a change in sound speed was noted. The graph shows that within the material, the sound speed is less than half of the sound speed in air. Combined with the frequencies, this implies that the wavelengths in the material are shorter. Also worth noting is that the model does not fully describe the dynamics of the materials. It has limited accuracy when used for detailed design or optimization of absorbing properties.

The structure of porous materials has great influence on the way it should be modelled. The most important structures are the rigid frame and the elastic frame. The first one is often modelled as a series of small cylindrical pores with radius $r$, according to the theory of Rayleigh (Rayleigh, 1896). The first tube-like structure was in a later stage altered to a possible random orientation by adding the structural factor $\phi$.

When looking at these approaches, we must consider whether we are dealing with an adiabatic or isothermal process. After researches from Crandall (1927) and Zwikker and Kosten (1949), it was concluded that the process of flow in narrow tubes was changing from isothermal at low frequencies to adiabatic at the high frequencies.

As mentioned, the process depends on the fluid dynamics within the tubes. To describe this, the non-dimensional parameter $\mu$ was introduced. Be careful not to mix this up with the dynamic viscosity.

$$\mu = \sqrt{\frac{\omega \rho_0 r^2}{\nu}} \quad (A.11)$$
With $\nu$ being the kinematic viscosity. When $\mu$ is small, the motion and thus the process is related to the inertial changes. However when $\mu$ is large, the motion is dominated by viscous processes.

To determine the sound speed of the acoustic waves within the material, the effective density and bulk modulus are needed.

$$\rho = \frac{\phi \rho_0}{\epsilon} + \frac{R_1}{j \omega} \quad \text{(A.12)}$$

With $\phi$ is the structural factor (tortuosity) and $\epsilon$ the porosity.

Here again, the structural factor is taken into account. The value is greater than 1, and often in the range of 2-3. However, the choice is arbitrary, making it less accurate. The determination of the flow resistivity is highly depending on the type of process and needs the pre-calculation of $\mu$.

$$R_1 = 8 \frac{\phi \omega \rho_0}{\epsilon \mu^2} \quad \mu \ll 1 \quad \text{(A.13)}$$

$$R_1 = \sqrt{2} \frac{\phi \omega \rho_0}{\epsilon \mu} \quad \mu \gg 1 \quad \text{(A.14)}$$

$\mu < 1$ for low frequencies or very narrow pores.

Zwikker and Kosten (1949) derived a model valid for all frequencies and derived expressions for the density and bulk modulus. For cylindrical pores in air.

For $\mu < 1$:

$$\rho = \frac{\phi}{\epsilon} \left( 4 \rho_0 + \frac{6}{\mu^2} \right)$$

$$K = \frac{1}{\epsilon} \left( 1 + 0.028 \frac{j \mu^2}{\mu} \right)$$

(A.15)

For $\mu > 10$:

$$\rho = \frac{\phi}{\epsilon} \rho_0 \left( 1 + \frac{2}{\sqrt{\mu}} \right)$$

$$K = \frac{1}{\epsilon} \kappa P_0 \left( 1 - \frac{\sqrt{0.92}}{j \mu} \right)$$

(A.16)

With $\mu$ being the dimensional parameter that was introduced earlier, $\kappa$ the thermal conductivity of air and $\epsilon$ the porosity. For increasing frequency and pore size, the effective density decreases and the bulk modulus increases. The sound speed in porous material is slower than in air, causing the process often being closer to an isothermal than adiabatic process.

These definitions for rigid frames do not apply for more elastic structures, something that could possibly be of interest when using additive manufacturing techniques. Kosten and Janssen (1957) researched the acoustic properties of flexible and porous materials, based on the previous mentioned work by Zwikker and Kosten. To combine the frame and fluid motion, a coupling coefficient was determined and applied in the form of a transition matrix. The coefficient itself is a function of the structural factor and the effective and real fluid densities.

$$\begin{bmatrix}
  -ik & j \omega \rho_1 + \tau & -\tau & p_1 \\
  0 & -j k & -\tau & j \omega \rho_2 + \tau & p_2 \\
  j \omega & 0 & -j k G & -j k (1 - \phi) K_2 & v_1 \\
  0 & j \omega & -j k H & -j k \phi K_2 & v_2
\end{bmatrix} = \begin{bmatrix}
  0 \\
  0 \\
  0 \\
  0
\end{bmatrix}$$

With $G = K_1 + \frac{(1 - \phi)^2}{\phi} (K_2 - P_0)$ and $H = (1 - \phi) (K_2 P_0)$.

The coupling coefficient $\tau$ is described according to following equation:

$$\tau = j \omega (\phi \rho - \rho_0) \quad \text{(A.17)}$$

Allard and Champoux (2009) described several phenomenological models. They succeeded in describing the viscous and thermal boundary layer thickness by using the effective density of the porous material and bulk modulus of air. The formulae for the boundary layers are obviously similar to the ones we encountered in chapter 2.
\[ \rho_e = k_s \rho_0 \left[ 1 + \frac{\sigma \epsilon}{j \omega \rho_0 k_s} \sqrt{1 + \frac{4 j k_s^2 \nu \rho_0 \omega}{\sigma^2 \Lambda^2 \epsilon^2}} \right] \] (A.18)

\[ K_e = \frac{\gamma P_0}{\gamma - (\gamma - 1)/\left( 1 + \frac{8 \nu}{j \Lambda'^2 \rho_0 \sigma \omega^2} \right)} \] (A.19)

\[ \delta_{BL_{visc}} = \sqrt{\frac{2 \nu}{\rho_0 \omega}} \] (A.20)

\[ \delta_{BL_{therm}} = \sqrt{\frac{2 \kappa}{\rho_0 C_p \omega}} \] (A.21)

Combining these formulas for the effective density and bulk modulus, we can derive the specific impedance and propagation wavenumber by using:

\[ z_c = \sqrt{K_e \rho_e} \] (A.22)

\[ k = \omega \sqrt{\frac{\rho_e}{K_e}} \] (A.23)

\( \gamma \) is the ratio of the specific heat capacities, \( P_0 \) is the atmospheric pressure, \( Pr \) is the Prandtl number, \( \delta_{BL_{visc}} \) is the viscous boundary layer, \( \delta_{BL_{therm}} \) is the thermal boundary layer, with \( \kappa \) being the thermal conductivity of air and \( C_p \) the specific heat capacity of air at constant pressure.

These formulas can be used in general, making them the perfect way to calculate the specific impedance and propagating wavenumber when knowing the effective density and bulk modulus. Comparisons with the Delany and Bazley model show a very good correlation. For complicated pore geometries, the correlation at mid frequencies turns weak.

### A.3. Resonant absorbers

Resonance related sound absorption is based on the natural frequency of the material. These absorbers work as a mass-spring system, which has a natural frequency (resonance frequency). There are two main types of resonant absorber: the Helmholtz resonator and the membrane or panel absorber.

#### A.3.1. Helmholtz resonator

The Helmholtz resonator has an enclosed air volume and a neck which contains vibrating air. The resonance frequency of the Helmholtz resonator is given by the following equation (Rienstra and Hirschberg, 2004).

\[ \omega_0^2 = \frac{S \eta_0^2}{(l + 2 \delta)V} \] (A.24)

With \( \omega \) is the angular frequency, \( S \) the cross sectional area of the neck of the Helmholtz resonator, \( l \) is the length of the neck, \( \delta \) is the end correction and \( V \) is the volume of the air void. Here we see that the resonance frequency is depending on the surface area, length and volume of the Helmholtz resonator. Lowering the air volume or the length will increase the frequency for which the resonator is efficient. The end correction (\( \delta \)) is dependent on the surface area, therefore that is not making any difference. Also worth noting is that the volume of the air void is important, and the exact dimensions only have small influence. See chapter 6 for research by Godbold (2008).

The resonance is created by the Helmholtz volume. Note that the actual absorption is provided by damping. The sound energy is removed due to for example porous material within the system, which is again based on friction. As we have seen earlier, this should be placed at the location where the particle velocity is highest, being in and/or right behind the neck of the Helmholtz resonator. Viscous losses within or right behind the neck of the resonator are beneficial for the sound absorption. These kinds of absorbers we see at perforated
A.3. Resonant absorbers

panels, where the small openings give room for the viscous losses.

The mass of the neck can be described with the following formula (Cox and D’Antonio):

\[
m = \frac{\rho}{\epsilon} \left[ 1 + 2\delta a + \sqrt{\frac{8\nu}{\omega} \left( 1 + \frac{l}{2\delta a} \right)} \right]
\]  
(A.25)

With \( l \) being the length of the neck of the resonator, i.e. the thickness of the perforated panel.

The end correction is depending on the geometry of the neck. A large variety of equations has been provided for this correction factor. A first estimation is often done by using:

\[
\delta = 0.85 \left( \frac{\pi}{S} \right)^{1/2}
\]  
(A.26)

For a circular orifice, this leads to an end correction of \( 0.85 \times r \). However, this does not take into account the effect of multiple orifices next to each other. Therefore, Ingard (1953) provided us with a more accurate formulation, valid for all porosities \( \epsilon < 0.16 \). Eventually this turned to the following equation by Rschevkin, allowing porosities up to 1.

\[
\delta = 0.8(1 - 1.47\epsilon^{1/2} + 0.47\epsilon^{3/2})
\]  
(A.27)

Allard and Atalla (2009) do not use the end correction in general, but they use a different end correction for the internal and external side of the aperture. All combined, (pg 203-205) this gives an impedance:

\[
Z = \frac{1}{s_e} \left[ -j s_i Z_c \cot(k e) + \frac{2d}{R} + 4 \right] R_t
\]  
(A.28)

Implicitly we can find the resonance frequency \( f_0 \) for which the absorption coefficient is maximum, by \( \text{Im} (Z) = 0 \):

\[
\cot \frac{2\pi f_0 \epsilon}{c_0} = \frac{1}{s_i c_0} [\epsilon_i + \epsilon_e + d] 2\pi f_0
\]  
(A.29)

Note that \( f_0 \) entered this equation by \( k = \frac{f_0}{c_0} \). This eventually leads to the resonance frequency:

\[
f_0 = \frac{c_0}{2\pi} \left( \frac{S}{V(d + \epsilon_i + \epsilon_e)} \right)^{1/2}
\]  
(A.30)

Which is again closely related to the overall known formula for the resonance frequency, as given in equation 4.22. In the equations above we have \( s_e \) is the surface of the aperture on the external side, \( s_i \) the surface on the inside, \( k \) is the wave number, \( \epsilon_i \) and \( \epsilon_e \) the porosity on the inside, respectively outside, \( d \) the thickness of the plate, \( R \) the radius of the aperture, \( V \) the volume of the resonator void.

According to Cox and d’Antonio (2009, page 180), viscous boundary layer effects will occur in perforations if they are small enough. The boundary layer are in the range of sub millimeter size, like we have seen earlier.

Extra peaks that occur in absorption coefficient graphs can most likely be assigned to these viscous boundary layer effects.

**Perforated panel absorber** The perforated panel works exactly by the same principle as the Helmholtz resonator, but without the closed volumes. It absorbs efficiently between 300-1500 Hz, but then needs a perforation degree of 5-10%. (van der Linden and Zeegers, 2006) Combined with the porous material, the effect of the latter might be decreased by the lower degree of perforation, but in return it gains some absorption in the lower and mid frequencies. (Micro)perforated panels actually work as a combined Helmholtz resonator. When dividing the panel in smaller volumes, the same principle is found.

A.3.2. Panel absorber

By using a massive plate, without perforation, the sheet forms a mass-spring system with the air layer behind. The effective absorption is between 50-500 Hz. By filling the air cavity with porous material, the effectiveness is reduced to max \( \alpha = 0.5 \), but the frequency range is wider. (van der Linden and Zeegers, 2006)
A.4. Other mechanisms

An other interesting mechanism is the tube resonator by dr. ir. Y.H. Wijnant, also known as the low reduced frequency model. This model describes the wave propagation in narrow tubes or layers, where the wavelength is large the viscous boundary layer thickness (Wijnant, 2006). The wave equation for these circumstances is:

\[
\nabla^2 p(x^{pd}) - k^2 \Gamma^2 p(x^p d) = 0 (A.31)
\]

With \( \Gamma \) being the propagation constant:

\[
\Gamma = \sqrt{\frac{\gamma}{\eta(s) B(s)}} (A.32)
\]

Where

\[
n(s \sigma) = 1 + \left[ \frac{\gamma - 1}{\gamma} D(s \sigma) \right]^{-1} (A.33)
\]

\[
D(s \sigma) = \frac{1}{S} \int_S C(s \sigma, \vec{x}^{cd}) dS (A.34)
\]

\[
B(s) = \frac{1}{S} \int_S A(s, \vec{x}^{cd}) dS (A.35)
\]

Where \( n \) is the polytropic constant, \( S \) the cross-sectional area. If viscous and thermal effects can be neglected, the propagation constant is equal to \( i \), giving the general wave equation. Van der Eerden (2001) showed that viscous effects in narrow tubes can provide an efficient sound absorber. Coupling different sizes gives opportunity for a more broader absorber.

Figure A.4: Calculated normal absorption coefficient for a wall with resonators (van der Eerden, 2001)

A.5. Conclusions

The literature study gives us several guidelines for the design. This creates a bridge between the theory and more practical part of the research. The different models and methods have showed us how the formulae and parameters can be used. This will be reviewed for friction, porous material, helmholtz resonators and the tube resonators.

Friction We have seen that the effect of the friction highly depends on the frequency. An important aspect that may give interesting results is the use of obstacles within the mean flow of tubes. We have seen from the boundary-layer separation at cylinders and spheres that these edges cause eddies and therefore turbulence which equals loss of acoustical energy. The enlargement of the boundary layer is worth investigating and can be an interesting starting point for the design. The viscous boundary layer might be bigger when the roughness of the wall is adjusted. However, one must keep in mind that the order of magnitude of the roughness can be of great importance. Friction based normative parameters are all traced back to the viscous boundary layer. The roughness and geometry are important.
Porous material  For the porous material we have seen several equations for determining the acoustical properties and therefore influencing the absorption coefficient of the element. We can look at the characteristic lengths, tortuosity, porosity and flow resistivity. Using models from Allard and Champoux (2009) and Zwikker and Kosten (1949), we can determine the specific impedance with equation A.15, A.16 and A.22. The goal for better absorption would be to get the impedance towards the specific impedance of air. The characteristic lengths are mainly determined by the tortuosity, porosity and flow resistivity. The other elements within the equations are based on geometry (related to porosity) and are constants that vary within a small range. The tortuosity is completely dependent on the porosity of the material. Both the porosity and the tortuosity will be in the same range, with the porosity close to unity and the tortuosity between 1 and 4, assuming the porosity will not drop below 25%. (Cox and D’Antonio, 2009) The only material dependent property is again the flow resistivity. However, using the effective density and bulk modulus with the model from Zwikker and Kosten (1949) it is possible to determine the impedance of the porous material. Different densities can create a combination of multiple frequency range absorbers.

Helmholtz resonators  Normative parameters for Helmholtz resonators are mainly geometrical parameters, that influence the resonating frequency of the element. When looking at the general resonating frequency equation, we find the following parameters.

\[ f_0 = \frac{C_0}{2\pi} \sqrt{\frac{S}{V(\ell + 2\delta)}} \]  

- Surface of aperture (\(S\))
- Volume (indirectly: geometry) of the cavity (\(V\))
- Length of the neck (\(\ell\))
- End corrections (\(\delta\))

The effect of some of these parameters on the resonating frequency is shown in the following figures. Adjusting a single parameter is obviously done with other parameters being constant. The first figure shows the effect of a changing neck radius. A smaller neck lowers the resonating frequency.

Figure A.5: Effect of orifice radius on resonating frequency

The following figure shows the effect of the volume depth to the resonating frequency. The total depth was kept constant at 5 cm. A changing neck length therefore automatically means a changing cavity depth, since the sum of these two is kept constant. An interesting point was from 25 to 30 mm cavity depth, the resonating frequency rose again. This means there are two solutions for resonating frequencies between the range of 150-300 Hz. Also interesting is the clear potential for low frequency absorption. The effect of volume depth is calculated with a neck radius of 2 mm and a volume width of 50 mm. The m-file of the following graph can be found in Appendix G.2.
Tube resonators The tube resonator shows the effect of coupling several tubes. Important parameters for this technique are mainly geometrical. Besides that, we should look at the roughness of the material. However, the thickness of the entire element will be the key issue in this situation. Interesting about these resonators is that they can be used sideways, as we see often in duct acoustics.

A.5.1. Design constraints
To set a clear goal for the absorber, several design constraints are determined. According to the NEN-EN-ISO 11654:1997 and the (new) NEN-EN-ISO 11654:2015, the final element will receive several absorption parameters. First the old, general parameters \( \alpha_w \) and the SA Class. The new norm has different coefficients for the low, mid and high frequencies. These are, respectively, \( \alpha_L \), \( \alpha_M \), \( \alpha_H \). The average value \( \alpha_a \) is also determined according ISO 354. The goal is to create a broadband absorber, meaning a fairly good to good absorption for all 1/3 octave bands.

| Thickness | 5 cm |
| Frequency range | 125-4000 Hz |
| \( \alpha \) | 0.75 minimum for \( \alpha_L \), \( \alpha_M \) and \( \alpha_H \) |
| | 0.80 for \( \alpha_a \) and \( \alpha_w \) |

Note: an \( \alpha_w \) of 0.80 or 0.85 corresponds to SA Class: B. Higher absorption coefficients equal SA Class: A.

A.5.2. Opportunities and disadvantages

Helmholtz resonator

+ Easy to tune to frequency
+ Different parameters that are adjustable and influence the frequency
+ Effect of the end correction when designed accordingly
+ Combinations and coupling possible
+ Size of volume adjustable with relative small influence
  - High resonance peaks
  - Maybe not the most potential for friction
  - Too much friction in neck might block the mass spring system
Tubes

- Used as quarter or half wavelength
- Small tubes have large contribution of friction for damping
- Coupling possible
- Curvature
  - Large lengths for low frequencies
  - Curvature reduces effect

The porous material is not taken into account as a starting point. Research has been done on possible ways of creating porous material with additive manufacturing. The most common way to produce this is by printing a grid-like structure, which preferably is as small as possible. Current research by F. Setaki will most likely be used for implementation within this design process. These results will be discussed in a later stage. Other possibility is to implement the porous material designed by Godbold (2008), who created a grid-like structure which is rotated in layers.

Friction  The way friction is approached within this research, is on the scale of the viscous boundary layer, resulting in elements with sizes not larger than tenths of millimetres. This automatically brings up possible disadvantage of certain types of additive manufacturing, since small sizes may be difficult or too fragile to print. However, research also showed that abrupt changes are effective, for which very small sizes may not be needed.
Appendix B - Additive Manufacturing

Additive manufacturing is one of the techniques of rapid prototyping. Creating prototypes can be done in three ways, and is therefore often separated in three categories.

- Subtractive: starting from a block of material, the material is removed until the desired shape is obtained.
- Additive: materials or elements are combined to create the desired shape.
- Formative: a specimen is shaped with the use of forces.

Most techniques within rapid prototyping are additive techniques. This also includes the additive manufacturing that is used in this research. From this we can conclude that additive manufacturing is a type of rapid prototyping. However, this does not hold the other way around (Liou, 2008).

A lot of techniques can be seen as a form of rapid prototyping. There are three conditions that the technique has to meet before calling itself a rapid prototyping technique.

- The technique should not contain a lot of skillful human intervention.
- No special tooling is needed to create a specific element. For this the focus is on post-processing of the element.
- There are very little geometric limitations.

Apart from the technique specific processes, the following basic operating principle holds for almost every technique of additive manufacturing. One thing to look at is the data flow from and to different kind of software.

1. Construct the CAD model
2. Convert the CAD model to STL format
3. Check and fix STL file
4. Generate support structures if needed
5. Slice the STL file to form layers
6. Produce physical model
7. Remove support structures
8. Post-process the physical model

B.1. Techniques
The most commonly used techniques, their (dis)advantages and applications are explained below. (3DPI, 2014)

B.1.1. Liquid-based techniques
With liquid-based processes, the initial material is in a liquid state. Results often have a quality finish, because it is possible to print a very smooth surface. One of the most known and widely used liquid-based technique is stereolitography (SLA).
**Stereolithography (SLA)**
Stereolithography works with photopolymer resins. When it gets in contact with a laser, the liquid forms a solid. Within the bed of resin, a moveable platform lowers the object layer by layer. A very accurate object can be printed with this technique. However, it needs supporting structures when the printed object contains overhangs or fragile parts. These supporting structures can be removed afterwards. This is something to take into account, in case the object has closed or hard to reach volumes. Before the final product is ready, it needs to be cured in an oven-like machine where it is fully hardened. For using this technique, often the printer Form 1/Form 1+ or Form 2 is used.

**Digital light processing (DLP)**
Digital light processing is a quite similar process to SLA. It differs mainly in the light source, which in this case is a conventional source. The technique is faster than SLA, and still contains the same accuracy. The post processing is similar; DLP also needs curing.

**B.1.2. Solid-based techniques**

**Fused deposition modelling (FDM)**
This technique is the most known, and one of the fastest. A plastic filament is melted and then layer by layer deposited. Each layer bonds to the previous one. For stability it requires support structures, that are removed afterwards. A rough surface is the first result, but there are treating options that can also make the product watertight.

**Inkjet: material jetting**
With this technique the actual material is printed in melted state, and is hardened by curing it with UV light. Advantage of this type of technique is that it is possible to use multiple materials at the same time. The results are accurate and very smooth.

**Selective deposition lamination (SDL)**
The selective deposition lamination uses ordinary copier paper that is layered and bonded with an adhesive. A more dense adhesive is used for places that should stay, other parts will be removed afterwards.

**B.1.3. Powder-based techniques**

**Laser sintering/laser melting (LS/LM)**
The first powder based technique is the laser sintering. The laser causes the powder to fuse together in a solid material. The powder bed should be tightly compacted. Again, a moveable platform lowers and allows for a new layer of powder to be sintered. This all happens in a chamber, because it is necessary to contain a specific temperature. Afterwards, the excess power can be removed. This technique does not require temporary structures, but it should be possible to remove the powder with brushes and high pressure air blowers. The accuracy and finish is not as good as with SLA and DLP, but the parts are much stronger.

**Inkjet: binder jetting**
In this case a binder is sprayed in a powder bed, that consequently fuses together. Like with laser sintering, the powder bed is then lowered to create the next layer. This type of technique does not need support structures. The result product is not that strong as with laser sintering, and post-processing is needed to make the product durable.

**Electron beam melting (EBM)**
This technique creates parts from metal powder, by using an electron beam. This should be conducted in a vacuum environment. The results are very accurate and designed for high-end use such as the medical industry.

**B.2. Added value**
Additive manufacturing is a fairly new process, dating from the end 20th century and starting to show its potential to the fullest in current practice. Within the building industry and therefore also the acoustics industry, the implementation is still minimal and the added value can be found in a lot of different topics. Some of the added values of additive manufacturing in general are mentioned below.
B.3. Characteristic demands

For this specific project, there are several characteristics that should turn out perfect for the optimal condition of the material and printing technique.

- The ratio of smoothness/roughness
- Acoustically hard
- High accuracy
- Strength
- Removal of support structure or powder

B.4. The use of additive manufacturing in acoustics

The small amount of research that has been done on the combined field of additive manufacturing and acoustics is discussed here. The pictures in appendix A are results from the PhD research done by Godbold (2008). These graphs show the effect from dimensional changes to a helmholtz resonator and the absorption changes of porous materials with different porosity. For the helmholtz resonator the results show that:

1. A smaller cavity depth gives a higher resonance frequency. Half of the depth heightens the resonant frequency with around 100 Hz. The width of the resonant peak is not much influenced by the depth.
2. Lengthening the orifice gives a lower resonance frequency and reduces the width of the resonant peak.
3. A smaller orifice width lowers the resonant frequency drastically. This is combined with a smaller peak width.

Another research by A. Selamet and I. Lee (2003) showed that extending the neck indeed lowers the resonance frequency of a Helmholtz resonator. The following figure also shows the effect of perforation of the extended part of the neck.

Figure B.1: Comparison of BEM predictions with experiment for a Helmholtz resonator with perforated neck extension ($\ell_1 = 8.5$ cm and $\ell_2 = 10$ cm). Source: Selamet and Lee, 2003
B.5. Conclusions
The techniques that can be used are determined by the possibilities at the Technical University of Delft. The printlab at the Faculty of Architecture contains a SLA printer, which has the advantage of having an extremely high accuracy (0.05 mm), but does have to print support structures. Other printers use FDM technique, that is not able to give such a high resolution. Even though the use of support structures is not ideal, the first samples will be printed using the Form 1, the SLA printer available at the Faculty of Architecture. Eventually use has been made of external companies, such as Shapeways in Eindhoven, to obtain the needed accuracy and have good post processing possibilities.
Appendix C - Derivations

C.1. Wave equation

In this section, the derivation of the wave equation is given. We start with defining an uniform and stagnant fluid.

\[ p = p_0 + p' \]
\[ \rho = \rho_0 + \rho' \]
\[ v = v_0 + v' \]
\[ s = s_0 + s' \]

Both \( p_0, \rho_0 \) and \( s_0 \) are constant and \( v_0 \) is assumed to be zero.

We define the linearized mass and momentum conservation laws and the linearized equation of state. With the linearization we neglect second order terms and the time derivative of a constant is zero. This gives us:

\[ \frac{\partial \rho'}{\partial t} + \rho_0 (\nabla v') = q \quad (C.1) \]
\[ \rho_0 \frac{\partial v'}{\partial t} + \nabla p' = \nabla \tau + f \quad (C.2) \]
\[ \frac{dp'}{dt} = c_0^2 \frac{\partial \rho_0}{\partial s} \frac{ds'}{dt} \quad (C.3) \]

After subtracting the divergence of the momentum linearization from the time derivative of the mass linearization, we obtain:

\[ \frac{\partial^2 \rho'}{\partial t^2} - \nabla^2 p' = \frac{dq}{dt} - \nabla \cdot \nabla \tau - \nabla f \quad (C.4) \]

Using the linearized equation of state with vanishing mean flow, we can define the wave equation:

\[ \frac{\partial^2 p'}{\partial t^2} - c_0^2 \nabla^2 p' = c_0^2 \frac{dq}{dt} - c_0^2 \nabla \cdot \nabla \tau - c_0^2 \nabla f + \left( \frac{\partial p}{\partial s} \right) \rho \frac{\partial^2 s'}{\partial t^2} \quad (C.5) \]

In a more general form, the wave equation is known as:

\[ \frac{\partial^2 \varphi'}{\partial t^2} - c_0^2 \nabla^2 \varphi' = 0 \quad (C.6) \]
C.2. Two microphone method

The complex sound pressure propagation in both the incident and reflected direction are:

\[ p_I = \hat{p}_I e^{(-j k_0 x + \phi_I)} e^{j \omega t} \] (C.7)
\[ p_R = \hat{p}_R e^{(j k_0 x + \phi_R)} e^{j \omega t} \] (C.8)

With \( p_I \) and \( p_R \) being the amplitudes of the incident and reflecting wave, \( k_0 \) the real wave number, \( \phi_I \) and \( \phi_R \) the phase shift of the incident and reflecting wave and \( x \) the distance from the sample.

At the two microphones, the pressures can be described as:

\[ p_1 = p_I(x_1) + p_R(x_1) = \hat{p}_I e^{-j k_0 x_1} + \hat{p}_R e^{j k_0 x_1} \] (C.9)
\[ p_2 = p_I(x_2) + p_R(x_2) = \hat{p}_I e^{j k_0 x_2} + \hat{p}_R e^{-j k_0 x_2} \] (C.10)

To correct for the phase and amplitude mismatch, the transfer function method uses the microphone interchange method. The measurement must be conducted in switched position, to get the transfer function between the microphones. The transfer functions of the separate waves are defined as:

\[ H_I = \frac{p_I(x_2)}{p_I(x_1)} = \frac{\hat{p}_I e^{-j k_0 x_2}}{\hat{p}_I e^{-j k_0 x_1}} = e^{j k_0 s} \] (C.11)
\[ H_R = \frac{p_R(x_2)}{p_R(x_1)} = \frac{\hat{p}_R e^{j k_0 x_2}}{\hat{p}_R e^{j k_0 x_1}} = e^{-j k_0 s} \] (C.12)

With \( s = x_2 - x_1 \), the distance between the two microphones. Combining these two transfer functions we obtain the transfer function of the two microphone method:

\[ H_{12} = \frac{p_1}{p_2} = \frac{\hat{p}_I e^{-j k_0 x_2} + \hat{p}_R e^{j k_0 x_2}}{\hat{p}_I e^{-j k_0 x_1} + \hat{p}_R e^{j k_0 x_1}} = e^{-j(k_0 x_2 + \phi)} + Re^{j(k_0 x_2 + \phi)} \] (C.13)

With \( R \) being the reflection coefficient, equal to the ratio of the complex reflected and incident pressure. Rewriting this equation and rephrasing this for the reflection coefficient at the sample surface (\( x=0 \)) we get an expression for the reflection coefficient.

\[ R = \frac{H_{12} - H_I}{H_R - H_{12}} e^{j k_0 x_1} \] (C.14)

From this we can calculate the absorption coefficient (Wolkesson, 2013; Chung and Blaser, 1980).

\[ \alpha = 1 - |R|^2 \] (C.15)
Appendix D - Additional results

In addition to the results in Chapter 4 the velocity plots of the triangular shapes are presented here.

Figure D.1: Velocity plots for Helmholtz resonator with triangular geometry at resonating frequency 625 Hz

The figures show a very low velocity within the voids of the triangular shapes, with very similar response as for the squared geometries.
Appendix E - Case Study
Reverberation time calculation

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| Room dimensions, X Y Z [m]          | 6.6    | 6.9 | 5.8    |       |     |     |     |     |     |     |     |     |     |
| Room constants                      | 4.0    | 265.8| 265.4 |       |     |     |     |     |     |     |     |     |
| Schroeder frequency [Hz]            | 106    |     |       |       |     |     |     |     |     |     |     |     |
| Attenuation parameters              | 343    | 21  | 50%    |       |     |     |     |     |     |     |     |     |

| Sabine                              |        |     |       |      |     |     |     |     |     |     |     |     |
| Eyring                              |        |     |       |      |     |     |     |     |     |     |     |     |
| Fissory                             |        |     |       |      |     |     |     |     |     |     |     |     |
| Millington-Settle                   |        |     |       |      |     |     |     |     |     |     |     |     |
| Arau-Puchades                       |        |     |       |      |     |     |     |     |     |     |     |     |
| Norm - lower limit                  | 0.7    |     |       |      |     |     |     |     |     |     |     |     |
| Norm - upper limit                  | 0.9    |     |       |      |     |     |     |     |     |     |     |     |

| T60 - a                             | 0.785s | Sabine |
| T60 - b                             | 0.777s | Sabine |

Reverberation time calculation with Sabine and Eyring methods.

Reverberation time curves for Sabine, Eyring, Fissory, Millington-Settle, Arau-Puchades, and T60A and T60B methods.
Appendix F - Previous Research

A very inspiring and related research has been the one from Godbold (2008). The most relevant results are presented in this appendix.

Figure F.1: Effect varying cavity depth (Godbold, 2008)

The figure above shows that the effect of varying the cavity depth is not of great influence on either the frequency or bandwidth of the absorption.
The length and width of the orifice of a Helmholtz resonator can have an effect on the absorbing frequency and bandwidth. A longer orifice lowers the peak frequency and makes the bandwidth of the frequency range smaller. A wider diameter higher the peak frequency and broadens the frequency range.

For the helmholtz resonator with an implemented perforated panel, the figure above shows the result of the change in perforation separation. One can clearly see that a larger separation lowers the peak frequency and the bandwidth.
Using the same porosity, the thickness of the samples show a broadening of the bandwidth for the thicker samples. This is as expected, since more friction causes for broadening of the absorption bandwidth. With a thickness of 20 mm, a reasonable absorption is obtained from 2000Hz upward.

The created dual layer broadband absorber has a predicted absorption with peaks at 700 and 1600 Hz. The measured sample shows some small differences.
G.1. Boundary layer thickness

%%%% Boundary layer as function of frequency
%% matlab code for plotting the thickness of the total boundary layer as a function of the frequency

clear all;
close all;

v = 15.11e-6; %kinematic viscosity (nu) of air at 20 degrees celsius
f = 0:100:4000; %frequency range
w = f.*(2*pi); %angular frequency
delta_BL_visc = sqrt((2.*v)./w)*1000; %formula for viscous boundary layer thickness [mm]

delta_BL_therm = delta_BL_visc./(sqrt(Pr));

figure(1)
plot(f,delta_BL_visc,'g',f,delta_BL_therm,'b')
title('Boundary layer thickness for different frequencies')
ylabel('Boundary layer thickness [mm]')
xlabel('Frequency [Hz]')
legend('Viscous boundary layer','Thermal boundary layer')
grid on

figure(2)
plot(f,alfa_1,'g',f,alfa_2,'b',f,alfa_3,'r')
title('Absorption of boundary layer with different orifice radii')
ylabel('Absorption boundary layer')
xlabel('Frequency [Hz]')
% This script calculates the resonance frequency of a helmholtz resonator with varying depth of the volume. The other parameters are held constant on the following values:
% Volume: depth=x, width50x50mm
% Length neck: 50-xmm
% Sound speed: 343 m/s
% Radius orifice: 4 mm
% clear all;
% close all;

c = 343;  % [m/s]
r = 0.004;
S = pi*r.^2;  % [m^2]
e = 0.85*(S./pi).^(1/2);  % end correction estimation cox d'antonio [m]
R = 0.0135;  % volume radius

for i=1:1:50
  d(i)=0.001*i;  % depth of volume
  V(i)=d(i)*(pi*R^2);  % [m^3]
  l(i)=0.05-d(i);  % [m]
  f(i)=(c./(2*pi)).*sqrt((S)/(V(i).*l(i)+2.*e));  % [Hz]
end

disp(f(i));
hold on
plot(d*1000,f);  % *1000 for a plot in mm
title('Effect of volume depth on resonating frequency');
xlabel('Depth of the volume [mm]');
grid on

% This script calculates the resonance frequency of a helmholtz resonator with varying volume radius. The other parameters are held constant on the following values:
% Volume: depth=x, width50x50mm
% Length neck: 50-xmm
% Sound speed: 343 m/s
% clear all;
% close all;

c = 343;  % [m/s]
r = 0.004;
S = pi*r.^2;  % [m^2]
G.4. Effect of neck length

```matlab
%This script calculates the resonance frequency of a helmholtz resonator
%with varying length of the neck. The other parameters are held constant
%on the following values:
%Length: 50 mm
%Volume depth: 50-x, radius: 13.5 mm
%Length neck: x
%Radius neck: 4 mm
%Sound speed: 343 m/s

clear all;
close all;
c=343; %[m/s]
r=0.004;
S=pi*r.^2; %[m^2]
e=0.85*(S./pi).^(1/2); %endcorrection estimation cox d'antonio [m]
R=0.0135;

for i=1:1:50
    l(i)=0.001*i; %length of neck
    d(i)=0.05-l(i);
    V(i)=d(i)*R^2; %[m^3]
    f(i)=(c./(2*pi)).*sqrt((S)/(V(i).*(l(i)+2.*e))); %[Hz]
end

disp(f(i));
hold on
plot(l*1000,f); %*1000 for a plot in mm
axis([0 15 0 5000]);
title('Effect of neck length on resonating frequency');
ylabel('Resonance frequency of the Helmholtz resonator [Hz]');
xlabel('Length of the neck [mm]');
grid on
```

G.5. Effect of neck radius

```matlab
%This script calculates the resonance frequency of a helmholtz resonator
%with varying length of the neck. The other parameters are held constant
%on the following values:
%Length: 50 mm
%Volume depth: 39 mm, radius: 13.5 mm
%Length neck: 11 mm
%Radius neck: x mm
%Sound speed: 343 m/s

clear all;
close all;
c=343; %[m/s]
r=0.004;
S=pi*r.^2; %[m^2]
e=0.85*(S./pi).^(1/2); %endcorrection estimation cox d'antonio [m]
R=0.0135;

for i=1:1:50
    l(i)=0.001*i; %length of neck
    d(i)=0.05-l(i);
    V(i)=d(i)*R^2; %[m^3]
    f(i)=(c./(2*pi)).*sqrt((S)/(V(i).*(l(i)+2.*e))); %[Hz]
end

disp(f(i));
hold on
plot(l*1000,f); %*1000 for a plot in mm
axis([0 15 0 5000]);
title('Effect of neck length on resonating frequency');
ylabel('Resonance frequency of the Helmholtz resonator [Hz]');
xlabel('Length of the neck [mm]');
grid on
```
clear all;
close all;
c = 343; % [m/s]
l = 0.011; % [m]
d = 0.039; % [m]
R = 0.0135; % [m]
V = d*pi*R^2; % [m^3]
for i = 1:1:25
    r(i) = 0.0005*i; % radius of neck
    S(i) = pi*(r(i).^2); % [m^2]
    e(i) = 0.85*((S(i)./(pi).^(1/2))); % end correction estimation cox d’antonio [m]
    f(i) = (c./(2*pi)).*sqrt((S(i))/(V.*(l+(2.*e(i))))); % [Hz]
end
plot(r*1000,f); % *1000 for a plot in mm
grid on

G.6. Analytical equation and two microphone method

%% TWO MICROPHONE METHOD
% matlab file to calculate the absorption coefficient with wave equation in an impedance tube.
% created by: Bettine Gommer, September 2016
% sources: Beekman, 2012; Tenpierik, 2016; Cox and d’Antonio, 2009;
% "Theoretical (Cox and d’Antonio)
% model for calculating the sound absorption curve of a Helmholtz resonator sample
% copyright M.J. Tenpierik, TU Delft
% edited by B. Gommer, TU Delft
% September, 2016

close all
clear all

T_t = 293.15; % temperature during measurements in Kelvin
C_t = 343.3; % [m/s] speed of sound in air
rho_t = 1.22; % density of air
Z0_t = C_t*rho_t;
viscosity_t = 15.1e-6; % kinematic viscosity of air
sigma_t = 0; % Flow resistivity of filler material
t_t = 11e-3; % plate thickness
l2_t = 39e-3; % backing thickness air
l1_t = 0.000000000001; % backing thickness porous absorber
a_t = 0.004; % hole radius
Sa_t = pi*a_t^2; % hole area
b_t = 0.0135; % volume radius
Sb_t = pi*b_t^2; % volume area
r_t = 0.029/2; % sample radius
Sr_t = pi*r_t^2; % sample area
f_t = [100:25:6400]; % Frequency
f = [100,125,160,200,250,315,400,500,630,800,1000,1250,1600]; % Frequency in 1/3rd octave bands
nf_t = length(f_t);
kair_t = 2*pi*f_t/C_t;
w_t = 2*pi*f_t;
n_t = 1; % way of mounting resonator (0: in free field; 1: flush mounted on surface; 2: at junction of two walls; 3: in 3 surface corner)
% impedance at top of air layer
z1_t = -j*20.*cot(kair_t);% Additional coefficient for surface area
Z1_t = -j*20.*cot(kair_t+11); % additional coefficient for surface area
k_t = 0.044; % calculate impedance of porous material (Delany and Bazley)
X_t = rho_t*w_t/sigma_t;
G.6. Analytical equation and two microphone method

\[
\begin{align*}
\% \text{characteristic impedance} \\
Z_c &= \rho_0 c \left( 1 + 0.0571 (X^{-0.754}) - j 0.087 (X^{-0.732}) \right) S_a / S_b \\
\% \text{wavenumber} \\
k_t &= (2 \pi / c_t) \cdot f_t \cdot (1 + 0.0978 (X_t^{-0.700}) - j 0.127 (X_t^{-0.595})) \\
\% \text{Impedance at top of porous absorbent} \\
z_2_t &= \frac{(-j z_1_t \cdot Z_c_t \cdot \cot(k_t l_2_t) + Z_c_t^2)}{z_1_t - j Z_c_t \cdot \cot(k_t l_2_t)} \\
\% \text{Loop over different open areas} \\
\eta_t &= S_a / S_r \\
ne_t &= \text{length}(\eta_t) \\
for \ m = 1:ne_t \\
D_t &= (8/3 \pi) \cdot a_t \cdot \eta_t(m) \% \text{Hole spacing} \\
\% \text{end correction} \\
\delta_t &= (8/3 \pi) \cdot a_t \\
\% \text{surface resistance} \\
\eta_m &= (\rho_0 / \eta(m)) \cdot \sqrt{8 \cdot \text{viscosity} \cdot w \cdot (1 + \pi / (2 \cdot \alpha))} \\
r_m &= \frac{\rho_0 c_t \cdot (1 + \alpha_t)}{(2 + \alpha_t) \cdot S_a} \\
\% \text{including viscosity terms} \\
z_4_t &= \frac{z_3_t}{\eta_t(m)} \\
\% \text{impedance of resonant absorber} \\
R_t &= \frac{z_4_t - Z_0_t}{z_4_t + Z_0_t} \% \text{reflection factor} \\
\alpha_t &= 1 - |R_t|^2 \% \text{absorption coefficient} \\
fpeak_t &= \frac{c_t}{2 \pi} \sqrt{\frac{S_a}{(S_b \cdot (l_1 + l_2) + \delta_t)}} \\
\lambda_{\text{peak},t} &= \frac{c_t}{f_{\text{peak},t}} \\
R_t &\text{rad} = \frac{2 \cdot S_a \cdot Z_0}{\lambda_{\text{peak},t}^2} \\
A_t &= \frac{2^n \cdot Z_0 \cdot S_a \cdot (\eta_m + R_t \cdot \text{rad})}{|z_4_t|^2} \\
\% \text{Insert first data set - Comsol} \\
\% \text{define starting parameters} \\
T &= 293.15 \% \text{temperature during measurements in Kelvin} \\
c &= 20 \cdot \sqrt{T} \% \text{[m/s] speed of sound in air} \\
L &= 0.055 \% \text{[m] distance from sample face to first mic (furthest from sample)} \\
s &= 0.02 \% \text{[m] distance between two mics} \\
L &= 0.637 \% \text{[m] total length of impedance tube} \\
\rho_0 &= 1.2 \% \text{[kg/m}\text{^3]} \text{ density of air} \\
\rho_0 &= 1.2 \% \text{[Pa-s/m]} \text{ specific impedance of air} \\
ID_d &= 0.029 \% \text{[m] diameter of the impedance tube} \\
\% \text{import csv file} \\
filename &= \text{'Normal_Finaldata.csv'} \\
M1 &= \text{csvread}(filename,5,1) \\
[b1,c1] &= \text{size}(M1) \\
f1 &= \text{csvread}(filename,5,0,[5 0 (b1+4) 0]) \\
u2 = M1(:,1) \% \text{pressure microphone 4} \\
u1 = M1(:,2) \% \text{pressure microphone 5} \\
f1 &= f1(1) \% \text{lower boundary frequency} \\
fh &= f1(\text{end}) \% \text{upper boundary frequency} \\
for \ i = 1:length(f1) \\
f1(i) \% \text{wavenumber} \\
\end{align*}
\]
ko_1(i) = -1.94*0.01*sqrt(f1(i))./(c*ID_d);
k0(i) = complex(k1(i),ko_1(i));
u=[u1_1; u2_1]; % pressure matrix
H12_1(i)=u2_1(i)./u1_1(i); % transfer function H12
H1_3(i)=exp(-i*i*k1(i)*s);
Hr_1(i)=exp(1i*k1(i)*s);
R_1(i)=exp(1i*2*k1(i)*L)*(H12_1(i)-H1_1(i))./(Hr_1(i)-H12_1(i));
Z_1(i)=(1+R_1(i))/(1-R_1(i));
alpha_1(i)=1-(abs(R_1(i)).^2); % absorption coefficient

%% Insert second data set - This can be repeated for multiple entries
% filename = 'Normal_onlysmalltube.csv';
%M_2 = csvread(filename,5,1); % pressure matrix
[b2,r2]= size(M_2);

%f2 = csvread(filename,5,0,[5 0 (b2+4) 0]);
%u2_2=M_2(:,1); % pressure microphone 4
%u1_2=M_2(:,2); % pressure microphone 5

%f1=f2(1) % lower boundary frequency
%fh=f2(end) % upper boundary frequency

% for i=1:length(f2)
% k2(i)=(2 *pi*f2(i))./c; % wavenumber
% ko_2(i)=-1.94*0.01*sqrt(f2(i))./(c*ID_d);
k0(i) = complex(k2(i),ko_2(i));
u=[u1_2; u2_2]; % pressure matrix
H12_2(i)=u2_2(i)./u1_2(i); % transfer function H12
H1_2(i)=exp(-i*i*k2(i)*s);
Hr_2(i)=exp(1i*k2(i)*s);
R_2(i)=exp(1i*2*k2(i)*L)*(H12_2(i)-H1_2(i))./(Hr_2(i)-H12_2(i));
Z_2(i)=(1+R_2(i))/(1-R_2(i));
alpha_2(i)=1-(abs(R_2(i)).^2); % absorption coefficient
% end

%% Insert measurement data
% filename = 'Masterfile3.xlsx';
% fm = xlsread(filename,'06.16','A3:A6403'); % normal helmholtz resonator
% alpha_m1 = xlsread(filename,'06.16','C3:C6403');
% alpha_m2 = xlsread(filename,'F20:F807'); % alpha_m3 = xlsread(filename,'U20:U807');
% alpha_m4 = xlsread(filename,'V20:V807');

%% Performance parameters
closest_1 = min(abs(f1-250));
bl_1 = find(f1 == 125+closest_1);
fl_1 = f1(bl_1);
R_1 = f1(end);

% closest_m = min(abs(fm-250));
% bl_m = find(fm == 125+closest_m);
% fm_m = fm(bl_m);

% f_t = f1(end);
% bndleft_t = find(f_t == 250);
% bndright_t = find(f_t == 2000);

alpha_t_tot = trapz(f_t(bndleft_t:bndright_t),alpha_t(bndleft_t:bndright_t)); % Get the area underneath the graph for quantitative analysis
alpha_1_tot = trapz(f1(bndleft_1:bndright_1),alpha_1(bndleft_1:bndright_1));
alpha_2_tot = trapz(f2,alpha_2);
alpha_3_tot = trapz(f3,alpha_3);
alpha_4_tot = trapz(f4,alpha_4);
alpha_m1_tot = trapz(fm(bndleft_m:bndright_m),alpha_m1(bndleft_m:bndright_m));
alpha_t_avg = alpha_t_tot.\/(f_t(bndright_t)-f_t(bndleft_t)); % Get the average absorption coefficient for the entire frequency range
alpha_1_avg = alpha_1_tot.\/(f1(bndright_1)-f1(bndleft_1));
alpha_2_avg = alpha_2_tot.\/(f2(end)-f2(1));
alpha_3_avg = alpha_3_tot.\/(f3(end)-f3(1));
alpha_4_avg = alpha_4_tot.\/(f4(end)-f4(1));
alpha_m1_avg = alpha_m1_tot.\/(fm(bndright_m)-fm(bndleft_m));

%% Plot reflection coefficient and impedance
figure
subplot(2,1,1)
semilogx(f_t,real(R_t),'k:',f1,real(R_1),f2,real(R_2))% ,f3,real(R_3))% ,f4,real(R_4))
hold on
title('Real part of reflection coefficient')
xlim([0 fh])
legend('Theoretical curve','Normal','0.5mm Triangles Down','0.5mm Triangles Up','Location','northwest')
axis([0 fh -1.1 1.1])

subplot(2,1,2)
semilogx(f_t,imag(R_t),'k:',f1,imag(R_1))% ,f2,imag(R_2),f3,imag(R_3))% ,f4,imag(R_4))
hold on
title('Imaginary part of reflection coefficient')
xlim([0 fh])
legend('Theoretical curve','Normal','0.5mm Triangles Down','0.5mm Triangles Up','Location','northwest')
axis([0 fh -1.1 1.1])

figure
subplot(2,1,1)
semilogx(f_t,real(Z_4_t),'k:',f1,real(Z_1))% ,f2,real(Z_2),f3,real(Z_3))% ,f4,real(Z_4))
hold on
title('Real part of the impedance');
legend('Theoretical curve','Normal','0.5mm Triangles Down','0.5mm Triangles Up','Location','northwest')
xlim([0 fh])

subplot(2,1,2)
semilogx(f_t,imag(Z_4_t),'k:',f1,imag(Z_1))% ,f2,imag(Z_2),f3,imag(Z_3))% ,f4,imag(Z_4))
hold on
title('Imaginary part of the impedance');
legend('Theoretical curve','Normal','0.5mm Triangles Down','0.5mm Triangles Up','Location','northwest')
axis([0 fh -500 500])

%% Plot absorption coefficient curve and bandwidth calculation
figure
do = semilogx(f_t,alpha_t,'--'); %different sets for properties
set(do,'Color',[.3 .3 .3]);
set(do,'LineWidth',1);
hold on
dq = semilogx(f1,alpha_1);
set(dq,'Color',[.3 .3 .3]);
set(dq,'LineWidth',1);
hold on
dp = semilogx(fm,alpha_m1);
set(dp,'Color',[1 0 0]);
set(dp,'LineWidth',1);
hold on
legend('Analytical','Numerical','Measured','Location','northwest')
axis([100 fh 0 1])

% Find peak and bandwidth for measured data
[pks,locs] = findpeaks(alpha_m1,fm,'minpeakdistance',500,'minpeakheight',0.2,'Annotate','extents');
```matlab
hold on
plot(locs, pks, 'ko', 'MarkerSize', 8)
f_max = locs(2);
a_max = pks(2);
str_max = [', num2str(f_max) ', ' [Hz] ; ', num2str(a_max) ', ' [-] '];
text(f_max, a_max, str_max, 'VerticalAlignment', 'bottom');
hold on
a_BW = a_max / 2;
alpha_m1_left = alpha_m1(1:a_max_index);
tmpl1 = abs(alpha_m1_left - a_BW);
[c index1] = min(tmpl1);
closest1 = alpha_m1(index1);
f_BW1 = fm(index1);
plot(f_BW1, a_BW, 'k.');
str_BW1 = [', num2str(f_BW1) ', ' [Hz] '];

alpha_m1_right = alpha_m1((a_max_index + 1):1000);
tmpl2 = abs(alpha_m1_right - a_BW);
[c index2] = min(tmpl2);
closest2 = alpha_m1(index2);
f_BW2 = fm(index2 + a_max_index);
plot(f_BW2, a_BW, 'k.');
str_BW2 = [', num2str(f_BW2) ', ' [Hz] '];

BW_m1 = f_BW2 - f_BW1;
Q_m1 = BW_m1 * a_max;

% Calculates the Q-factor used in quantitative analysis

% Find peak and bandwidth for numerical data
[pks1, locs1] = findpeaks(alpha_1, f1, 'Annotate', 'extents');
hold on
plot(locs1, pks1, 'ko', 'MarkerSize', 8)
f1_max = locs1(1);
a1_max = pks1(1);
str1_max = [', num2str(f1_max) ', ' [Hz] ; ', num2str(a1_max) ', ' [-] '];
text(f1_max, a1_max, str1_max, 'VerticalAlignment', 'bottom');
hold on
a1_BW = a1_max / 2;
alpha_1_left = alpha_1(1:a1_max_index);
tmpl1 = abs(alpha_1_left - a1_BW);
[c index1] = min(tmpl1);
closest1 = alpha_1(index1);
f1_BW1 = f1(index1);
plot(f1_BW1, a1_BW, 'k.');
str1_BW1 = [', num2str(f1_BW1) ', ' [Hz] '];

alpha_1_right = alpha_1((a1_max_index + 1):1000);
tmpl2 = abs(alpha_1_right - a1_BW);
[c index2] = min(tmpl2);
closest2 = alpha_1(index2);
f1_BW2 = f1(index2 + a1_max_index);
plot(f1_BW2, a1_BW, 'k.');
str1_BW2 = [', num2str(f1_BW2) ', ' [Hz] '];

BW_1 = f1_BW2 - f1_BW1;
Q_1 = BW_1 * a1_max;

% Calculates the Q-factor used in quantitative analysis
```

clear all % Clears all previously defined variables in matlab
close all % Closes all open figure windows

%% Distance from microphone to sample surface
z1 = 1.1938;
z2 = 0.8518;
z3 = 0.5520;
z4 = 0.3106;
z5 = 0.1903;
z6 = 0.0593;

z_values = [z1; z2; z3; z4; z5; z6]; % Vector with all z values for convenience

fs = 192000; % sample frequency [Hz] The number of samples taken per second by the audio device (sampler)

fvector = [200:2500]; % Frequency vector [Hz] Defines the frequency range of interest.

zt = 1.3690; % Distance from the source to the specimen surface [m] The distance is used to determine the amount of samples used for cross correlation of the signals and can be measured using for example a tapemeasure

%% calculation of sound speed
[speed]=audioread('sound_speed.wav');
[V,I]=findpeaks(speed); % V-peak value; I-index of peak value
[B,II]=sort(V,'descend'); % B-value; I-index

c=2*z1.*fs./(I(II(2))-I(II(1)));

%%% DATA INPUT & INPUT PARAMETERS

% load IRmic1.txt % Loads the impulse responses measured by mic1, 2, 3 and 5 into matlab. In order to be able to do so, the filenames of the impulse response text files should be the same as the file names in the load command and the header should be removed
% load IRmic2.txt
% load IRmic3.txt
% load IRmic4.txt
% load IRmic5.txt
% load IRmic6.txt

% load IRmic1,2,3,4,5 into matlab

[IRmic1,fs]=audioread('fins4ext3_1.wav'); % Loads the impulse responses measured by mic1, 2, 3 and 5 into matlab. In order to be able to do so, the filenames of the impulse response text files should be the same as the file names in the load command and the header should be removed
[IRmic2,fs]=audioread('fins4ext3_2.wav');
[IRmic3,fs]=audioread('fins4ext3_3.wav');
[IRmic4,fs]=audioread('fins4ext3_4.wav');
[IRmic5,fs]=audioread('fins4ext3_5.wav');
[IRmic6,fs]=audioread('fins4ext3_6.wav');

% DATA INPUT & INPUT PARAMETERS

IR1 = IRmic1(1:fs,1); % Impulse response measured by microphone 1
IR2 = IRmic2(1:fs,1); % Impulse response measured by microphone 2
IR3 = IRmic3(1:fs,1); % Impulse response measured by microphone 3
IR4 = IRmic4(1:fs,1); % Impulse response measured by microphone 4
IR5 = IRmic5(1:fs,1); % Impulse response measured by microphone 5
IR6 = IRmic6(1:fs,1); % Impulse response measured by microphone 6
IR = [IR1 IR2 IR3 IR4 IR5 IR6]; % Matrix with all IRs for convenience

c = 346.76187616746467; % Determined by measurement with rigid termination

ki_emp = -0.02*sqrt(fvector)./c.*0.04); % Calculate the attenuation
% constant for correcting plane wave attenuation by the impedance tube according to ISO 10534-2

L = z1; % Relevant tube length [m]
x1 = 0; % Distance between mic1 and mic1 [m]
x2 = L-z2; % Distance between mic2 and mic1 [m]
x3 = L-z3; % Distance between mic3 and mic1 [m]
x4 = L-z4; % Distance between mic4 and mic1 [m]
x5 = L-z5; % Distance between mic5 and mic1 [m]
x6 = L-z6; % Distance between mic6 and mic1 [m]

N = round((zt/c)*fs)*2; % The number of samples before the direct sound reaches the back wall of the tube. This number is used for the cross correlation of the signals in which only the direct sound is taken into account

% CROSS CORRELATION

% The impulse responses are measured separately, this may cause a phase difference. To correct this, the difference in sample number of the first peak in the impulse response is compared to the expected difference in sample number caused by the increase in distance from the source.

% Cross correlation for IR2 with IR1 as reference
C2 = xcorr(IR1(1:N+1),IR2(1:N+1)); % Crosscorrelation function of IR2 with IR1 as reference over samples 1 to N+1
M2 = max(C2,[],1); % Maximum value in cross correlation C2

for n = 1:2*N;
    if C2(n) == M2; % "If row n in the range 1 to 2N in C2 equals M2" then A2 = n;
end

% Cross correlation for IR3 with IR1 as reference
C3 = xcorr(IR1(1:N+1),IR3(1:N+1)); % Cross correlation function of IR3 with IR1 as reference over samples 1 to N+1
M3 = max(C3,[],1); % Maximum value in cross correlation C3

for n = 1:2*N;
    if C3(n) == M3; % "If row n in the range 1 to 2N in C3 equals M3" then A3 = n;
end

% Cross correlation for IR4 with IR1 as reference
C4 = xcorr(IR1(1:N+1),IR4(1:N+1)); % Crosscorrelation function of IR4 with IR1 as reference over samples 1 to N+1
M4 = max(C4,[],1); % Maximum value in cross correlation C4

for n = 1:2*N;
    if C4(n) == M4; % "If row n in the range 1 to 2N in C4 equals M4" then A4 = n;
end
for n = 1:2*N;  % The function finds the row number in which M4 occurs
    if C4(n) == M4;  % "If row n in the range 1 to 2N in C4 equals M4
        A4 = n;  % than A4 = n"
    end
end

% Cross correlation for IR5 with IR1 as reference
C5 = xcorr(IR1(1:N+1),IR5(1:N+1));  % Crosscorrelation function of IR5
    % with IR1 as reference over samples 1 to N+1
M5 = max(C5,[],1);  % Maximum value in cross correlation
    % C5

for n = 1:2*N;  % The function finds the row number in which M5 occurs
    if C5(n) == M5;  % "If row n in the range 1 to 2N in C5 equals M5
        A5 = n;  % than A5 = n"
    end
end

% Cross correlation for IR6 with IR1 as reference
C6 = xcorr(IR1(1:N+1),IR6(1:N+1));  % Crosscorrelation function of IR6
    % with IR1 as reference over samples 1 to N+1
M6 = max(C6,[],1);  % Maximum value in cross correlation
    % C6

for n = 1:2*N;  % The function finds the row number in which M6 occurs
    if C6(n) == M6;  % "If row n in the range 1 to 2N in C6 equals M6
        A6 = n;  % than A6 = n"
    end
end

S2 = N+1-A2;  % Number of samples the first peak in the impulse
S3 = N+1-A3;  % response differs from the first peak in IR1
S4 = N+1-A4;
S5 = N+1-A5;
S6 = N+1-A6;

D2 = round((x2/c)*fs);  % Number of samples the first peak in the
D3 = round((x3/c)*fs);  % impulse response should differ from the
D4 = round((x4/c)*fs);  % first peak in IR1
D5 = round((x5/c)*fs);
D6 = round((x6/c)*fs);

T1 = 0;  % Number of samples the impulse response should be
T2 = D2 - S2;  % shifted forward to correlate with microphone 1
T3 = D3 - S3;  % (negative means it should be shifted backwards)
T4 = D4 - S4;
T5 = D5 - S5;
T6 = D6 - S6;

IR1 = IR1;
IR2 = zeros(length(IR1),1);
IR3 = zeros(length(IR1),1);
IR4 = zeros(length(IR1),1);
IR5 = zeros(length(IR1),1);
IR6 = zeros(length(IR1),1);

if T2 >= 0
IR22(T2+1:end) = IR2(1:end-T2);
else
IR22(1:end+T2) = IR2(-T2+1:end);
end
if T3 >= 0
IR33(T3+1:end) = IR3(1:end-T3);
else
IR33(1:end+T3) = IR3(-T3+1:end);
end
if T4 >= 0
IR44(T4+1:end) = IR4(1:end-T4);
else
IR44(1:end+T4) = IR4(-T4+1:end);
end
if T5 >= 0
IR55(T5+1:end) = IR5(1:end-T5);
else
IR55(1:end+T5) = IR5(-T5+1:end);
end
if T6 >= 0
IR66(T6+1:end) = IR6(1:end-T6);
else
IR66(1:end+T6) = IR6(-T6+1:end);
end
% ________________________________________________________________________
% FAST FOURIER TRANSFORM
% ________________________________________________________________________
FFT1 = fft(IR11);  % Fourier Transform of the correlated impulse
FFT2 = fft(IR22);  % response
FFT3 = fft(IR33);
FFT4 = fft(IR44);
FFT5 = fft(IR55);
FFT6 = fft(IR66);
% ________________________________________________________________________
% COMPUTATIONS
% ________________________________________________________________________
% The function is repeated for each frequency within the range of interest
% The plane wave propagating through the tube is modeled. The ideal
% propagation of the wave over the length traveled without obstructions is
% modeled and compared to the wave reflected by the specimen and back wall
for ff = 1:length(fvector)
k = (2*pi*fvector(ff)./c)+(1i*ki_emp(ff));
e11 = exp(-1i*k*x1);
e12 = exp(-1i*k*(2*L-x1));
e21 = exp(-1i*k*x2);
e22 = exp(-1i*k*(2*L-x2));
e31 = exp(-1i*k*x3);
e32 = exp(-1i*k*(2*L-x3));
e41 = exp(-1i*k*x4);
e42 = exp(-1i*k*(2*L-x4));
e51 = exp(-1i*k*x5);
e52 = exp(-1i*k*(2*L-x5));
e61 = exp(-1i*k*x6);
e62 = exp(-1i*k*(2*L-x6));
M = [e11 e12;e21 e22;e31 e32;e41 e42;e51 e52;e61 e62];
Pmeas = [FFT1(fvector(ff)+1);FFT2(fvector(ff)+1);FFT3(fvector(ff)+1);FFT4(fvector(ff)+1);FFT5(fvector(ff)+1);FFT6(fvector(ff)+1)];
Ptemp = M\Pmeas;
R(ff) = Ptemp(2,1)./Ptemp(1,1);  % Reflection coefficient
end
Z = -(R+1)./(R-1);  % Surface impedance
a = 1-abs(R).^2;  % Absorption coefficient
% ________________________________________________________________________
% OUTPUT
% $Z_{\text{measured}} = \{\text{fvector; real}(Z); \text{imag}(Z); \text{abs}(Z); Z\}$;
% $a_{\text{measured}} = \{\text{fvector}; a\}$;
% save('Z1_measured.mat','Z_measured')
% save('a1_measured.mat','a_measured')

% FIGURES

figure
semilogx(fvector,a)
title('Absorption coefficient')
legend('Measurement')
ylabel('Frequency [Hz]')
ax = gca;
set(ax,'XTick',
    [200 250 315 400 500 630 800 1000 1250 1600 2000 2500]);
set(ax,'XTickLabel',
    {'200','250','315','400','500','630','800','1k','1.25k','1.6k','2k','2.5k'});
grid
F = transpose(fvector);
A = transpose(a);
filename = 'EindhovenData.xlsx';
xlswrite(filename,F,1,'A2');
xlswrite(filename,A,1,'B2');