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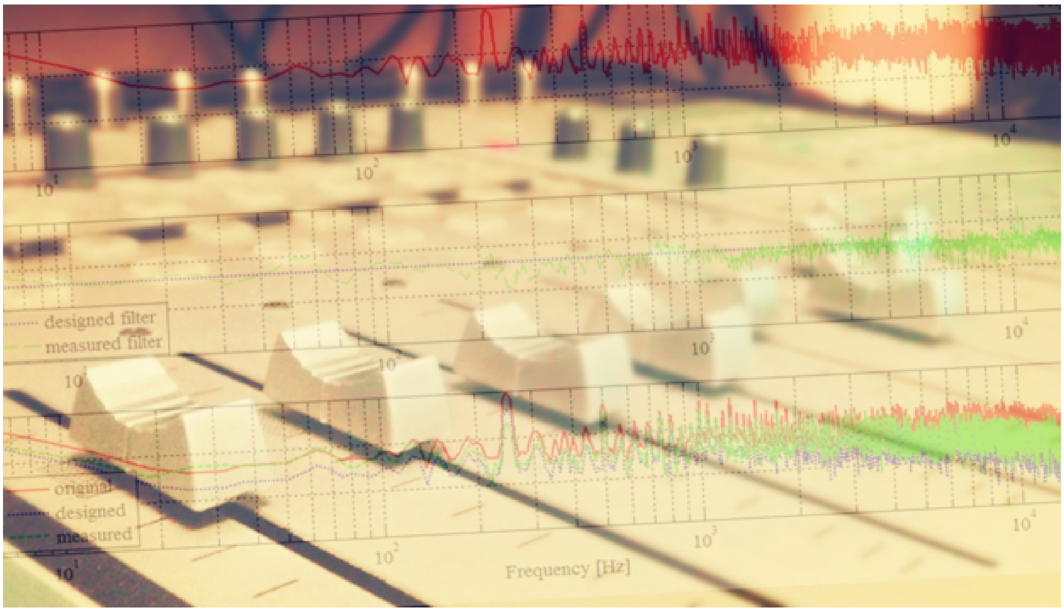

TOWARDS A METHODOLOGY FOR CONSEQUENTIAL PRODUCT SOUND DESIGN

Master's thesis in the Master's programme in Sound and Vibration

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Abstract

Product sound design is attracting more and more attention due to the fact that product acceptance is affected directly by the product sound quality. There are two categories of product sound: consequential and intentional sounds. Intentional sounds are the sounds created by electroacoustic transducers mounted in the product. These sounds need to be designed. Consequential sounds, on the other hand, are sounds that are radiated as a consequence of the vibroacoustic properties of the mechanical components such as engines and pumps, and their coupling to surrounding substructures.

The main purpose of this Master's thesis is to focus on consequential sound design. Using a coffee machine, the possibility to design or predict the product sound of a modified product based on recordings of the product and basic vibroacoustic assumptions is explored.

The mechanics of the coffee machine is studied to get a correlation between the radiated sound and the mechanical components. Furthermore, a target sound is designed based on the knowledge of both technical acoustics and sound design. The last step is to modify the real machine and obtain a good match between the radiated sound and the target sound. Jury group evaluations are conducted in order to evaluate and iterate the sound designs and compare the designed sounds to the actual product sounds.

Keywords: Consequential product sound design, sound quality, technical acoustics, sound sketch, jury group evaluation.

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Introduction

Over the past few decades, the importance of product sound design has gained considerable interest within the industry and design institutions due to the fact that the acceptance of products can be affected directly by the product sound quality. People are immersed in various product sounds during daily life: rings from a cellphone, beeping sound from a microwave oven when the heating is done, the sounds we can hear when we are making a cup of coffee, using a blender, or when we shut the door. From the examples above, product sound can be divided into two categories: consequential and intentional sounds.

Intentional sounds are the sounds created by electroacoustic transducers mounted in the product. They are required to be well designed as an informative auditory icon or earcon, most of the time, to provide feedback to users. Using a typical intentional sounds, people can even distinguish different products or brands. Well-designed intentional sounds are quite helpful for easier access to multifunctional products and can be acquainted as a symbol of a company brand.

Consequential sounds, apart from intentional sounds, are sounds that are radiated as a consequence of the vibro-acoustic properties of the mechanical components and their coupling to surrounding substructures. Consequential product sound design might involve vibration and noise control which has been widely used to handle product noise problems. Treatments such as damping, enclosures and isolations, are based on the knowledge of sound sources and structure-borne radiation. In most cases, consequential product sound design is carried out as an improvement according to customers' complaints or desires after the production has been launched in the market.

Sound design is often vital for maximizing the acceptance of products. For some products, the sound has a strong connection to the company brand and is therefore of highest importance.

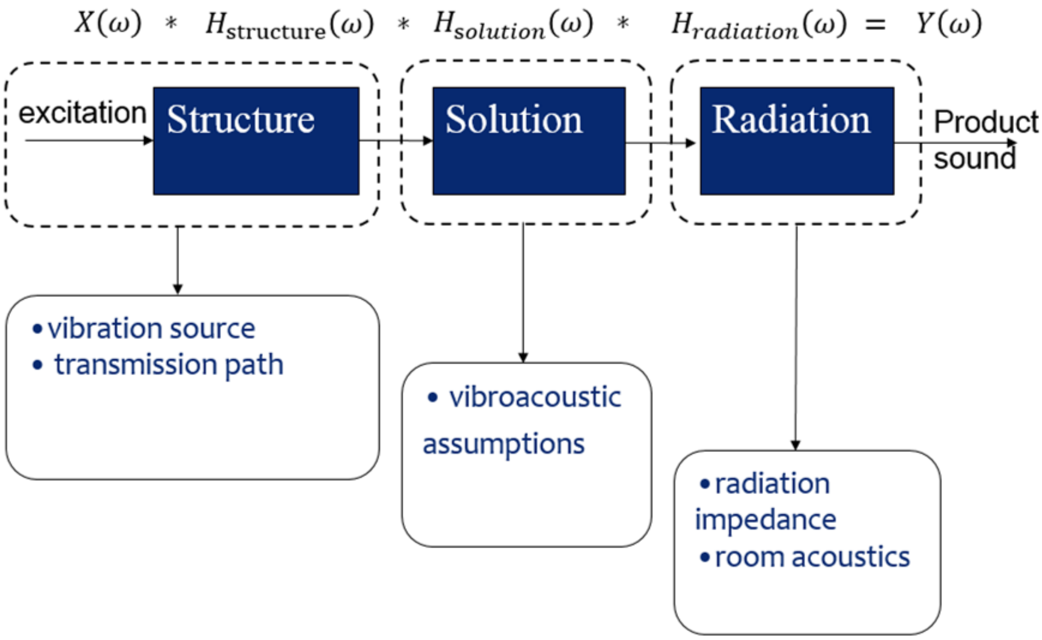


Figure 1.1: General methodology.

1.1 Problem description

For product sounds, especially consequential product sounds, it's generally hard to design or to improve the sound quality. As stated before, consequential product sounds are mostly experienced as undesirable noise. In general, it is difficult to describe the problems of product sound in a structured manner, which impedes communications between designers and users. Furthermore, the complex mechanical structures within products allow for several sources, multiple transfer paths, but it is usually difficult to disassemble the structure or modify the real machine. Consequently, there are difficulties to improve consequential sound quality even if responses or complaints from market are obtained. Hence, it is of significance to provide a methodology that could help the designer find efficient and applicable solutions to predict and create target product sounds of higher acceptance.

The general methodology is modeled in fig. 1.1, which contains three dashed blocks that stand for three elements in obtaining the target product sound. Before exploring the possible solutions, one should first get the knowledge of the machine itself as the machine vary a lot, including the sources, the transfer path, and the radiation from the surfaces. This thesis work mainly focus on the aspect of vibro-acoustics due to the study object– a coffee machine. Sources introduce the interaction of material at a location in an environment with a certain impact caused by the excitation power. The excitation power is electricity. While vibro-acoustic energy is delivered to the product at a desired moment for a limited time, the excitation from the vibro-acoustic sources is taken as 'input' $X(\omega)$.

Then, the energy transfers within the structure. Structure-borne sound transmission is carried out by the coupling components, while air and liquid sound transmissions are carried out by the mediums. Multiple transfer paths are quite common depending on the product layout. The transfer paths constitutes $H_{structure}(\omega)$. Based on this information, reasonable assumptions for a possible solution $H_{solution}(\omega)$ can be proposed based on knowledge in technical acoustics. More than one solutions can be applied on the product for the same purpose, different solutions introduce different product sound, which will be evaluated to find the final solution. In practice, this means noise reduction treatments such as damping material, absorber, attached mass etc. These methods allow us to reduce the undesirable noise or change the spectrum character of the radiated sound. $H_{radiation}(\omega)$ takes sound radiation and room acoustics into consideration. The sound radiation is the excitation of air by vibrating surfaces. After the vibro-acoustic energy is transferred through different transfer paths, it is radiated from different surfaces and the sound waves propagate in the air. The properties of the room such as geometry, the absorption, and the position in the room are significant as they affect the perceived sound directly. Hence, for all the recordings of the product sound, there are several aspects that should be noticed:

a) Suitable equipment that will not introduce unwanted disturbing noise;

b) Sound should be recorded at the typical ear position of the user;

c) Measurement position must be fixed to keep consistence during the whole design work;

d) Room acoustic properties should be unchanged;

In summary, the general methodology can be modeled as

$$Y(\omega) = H_{structure}(\omega) \times H_{solution}(\omega) \times H_{radiation}(\omega) \times X(\omega), \tag{1.1}$$

where $X(\omega)$ and $Y(\omega)$ stand for the excitation from vibration source and the recorded production sound respectively.

1.2 Purpose of the thesis

The purpose of this Master's thesis was to explore a methodology to predict the consequential product sound of a modified product by using recordings of the product and different vibroacoustic assumptions.

On the application level, several papers have focused on the evaluation and description of the various aspects of SQ perception in both overall vehicles and specific car elements [1, 2, 3, 4, 5, 6]. In addition to the studies concerning vehicle SQ, researches on household equipment such as vacuum cleaners, washing machines and fridges have been performed where the perception and characteristics of the SQ and specific machinery elements have been studied [7, 8, 9, 10]. In these studies, statistical analysis illustrating the relationship between the perceived qualities and the frequency characteristics of the sound signals are performed. These SQ descriptions are helpful during analyses, benchmarking and defining targets. The radiated sound can give us hints about the physics of how the noise is generated and transmitted, which is helpful when designing and optimizing the products to ensure high acceptance from customers.

However, the methods applied in the cited studies above either involve extensive modifications on real machines or computer aided simulations for structural analyses and product sound predictions, both of which are quite time consuming. This thesis is an effort to develop a design methodology that requires less experimental and numerical simulations to enable efficient target sound predictions and evaluations.

1.3 Thesis goals and tasks

There are two aspects included in this thesis work: one concerns the real physics of the machine, and the other deals with sound sketching and evaluation.

A coffee machine (Model no: ELM3100) by Electrolux was studied in this thesis. The product sound together with the structure of the machine were studied to identify vibro-acoustic problems and obtain potential target sound. After the target sound is created, we need to modify the machine to achieve a sound which matches the target sound. The physical feasibility should be analyzed for further adjustments to the target sound and the machine.

Sound sketching was applied to create target sounds. This method has an advantage over potential target sounds evaluation and physical modification estimation because one can predict and evaluate the potential target sounds based on the original product sound and knowledge on technical acoustics before any prototype is manufactured.

The crucial goals and tasks involved during this Master's thesis are:

a) Carry out a literature survey on sound quality, sound design, sound sketching, jury group evaluation and technical acoustics;

b) Study the vibro-acoustic properties of the coffee machine. The character of the potential target sounds and possible vibro-acoustic treatments are also defined based on the he vibro-acoustic study of the machine;

c) Sketch potential target sounds using a digital audio mixing software, evaluate different vibro-acoustic treatments such as damping layers, absorbers, attached mass;

d) Evaluate the sound sketches by using a jury group and obtain feedback from the participants with regards to the product sound quality and acceptance;

e) Perform vibroacoustic treatments on the coffee machine according to the feedback obtained during jury group evaluations, until the desirable sounds are designed;

f) Test the designed sound sketches by conducting the second listening test.

1.4 Delimitations

The product used in this thesis work is a coffee machine by Electrolux (Model no: ELM3100), therefore, all the vibro-acoustic assumptions based on the original recorded sound of the real product were limited to this specific machine, for instance, the frequency characteristics.

The values of reduction using each method were calculated using relative equations assuming longitudinal wave incident, therefore, the results were different from those of bending wave. The filters which simulate the treatments of damping layers and attached mass (method 3 to method 12) were designed for frequencies between 1 kHz to 3 kHz, thus, the sound characteristics of the designed sounds deviated from the real sounds of the coffee machine with same treatments. This might introduce biasing during the sound sketching and evaluations. All materials are relatively heavy, but the effect of attached mass was not included in the filters that simulate the treatment of absorbers due to the fact that the weight of absorbers was light compared to the plastic structure of the coffee machine. While calculating the value of reduction for damping layers and mass blocks, the mass per area was calculated as the sample mean value of mass per area of the damping layer or mass block and that of the ABS plastic structure of the coffee machine. However it was not possible to remove the left chassis as the pump was fixed on the left side, hence, the mass per area of the left chassis was approximately calculated according to the thickness that one can see.

2

Literature

In this section, a brief literature review is presented relevant to the main topics in this thesis: product sound quality and technical acoustics (TA). The concept of product sound design (PSQ) includes the definition, the application, and the general approaches for product sound quality engineering, sound sketching and jury group evaluation. An introduction to TA is also given in this chapter to provide the theory behind vibro-acoustic treatments used in this work.

2.1 Product sound quality

PSQ is defined as "the perceptual reaction to the sound of a product that reflects the listener's reaction to the acceptability of that sound for that product; the more acceptable, the greater the SQ" [11].

The definition illustrates that PSQ is product specific as the perception is certainly affected by a particular product. Listeners can rate the PSQ using different scales, and this scale might be correlated with the product sound. For example, listeners will answer questions about how well made the product seems to be, how well it seems to be working, and about the perceived sound power and loudness [11]. Depending on various situations, the general term of acceptability may refer to any or all of these.

PSQ has become a critical design aspect in fields such as acoustic performance and noise comfort optimization. As a result, considerable research has been focused on analyzing the characteristics of PSQ and sound design.

2.1.1 sound design steps

The basic steps in sound design engineering can be divided into the following parts [12]:

a) PSQ assessment;

b) Sound quality diagnosis and problem identification;

c) PSQ solution engineering.

The whole process of PSQ engineering can be explained more detailed as fig. 2.1:

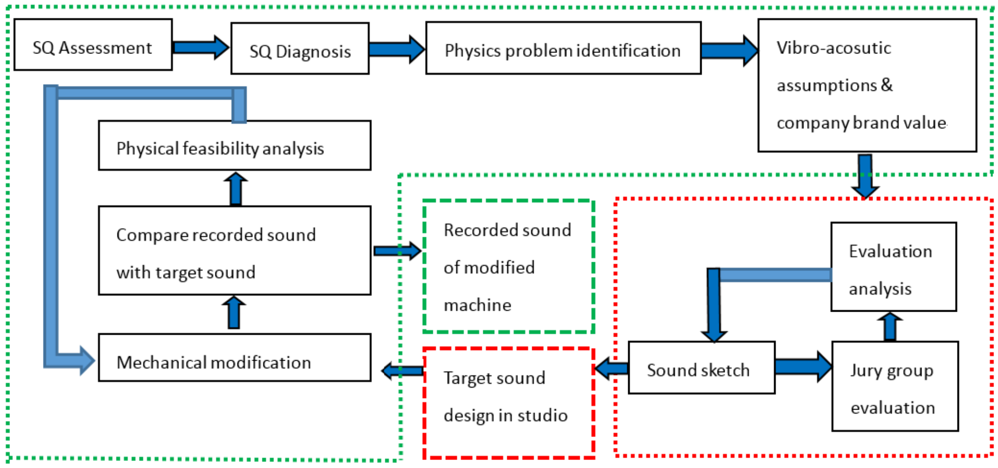


Figure 2.1: General process of PSQ engineering.

Step 1: Product sound assessment

In order to analyze the sound statistically, obtaining an appropriate understanding and description of the SQ is the first step in product sound design.

Depending on the aim of the design, SQ assessment can be performed by binaurally replaying sounds to a group of listeners while using questionnaires for evaluation. The crucial perceptual characteristics should be properly described and defined before final evaluations, whenever a quantitative analysis is required.

Step 2: Sound quality diagnosis and problem identification

After obtaining basic descriptors of the sound quality, the second step is to analyze the sound signal in detail to reveal the correlation between different sound components and sound quality descriptors. A combination of several methods is usually used in this step.

Signal processing techniques such as time-frequency analysis, time analysis, angular domain processing, order structure and order value analysis are the most common methods for analyzing the intrinsic temporal and spectral characteristics of sounds.

Another method is to synthesize sounds containing selected features from the signal under analysis. This approach can be used for parametric evaluation of these features without being affected by disturbing sources [12].

By using all the methods mentioned above, the SQ characteristics can be related to specific sound properties. Aiming at identifying the SQ problem in terms of sources, components or vibroacoustic parameters, the correlation between the acoustic response signal and physical mechanisms is required.

Step 3: Physical problem identification and modeling
Several techniques are available to achieve the physical problem identification. They can be classified into the following three methods:

a) Correlation analysis is the first widely applied method. It considers the relation between and effects of a particular source or component and the radiated sound. By applying this method, the critical features of the radiated sound are related to other measurable signals which are more directly related to the physical explanation of the problem.

b) The second method is transfer path analysis, where the partial contribution of each transmission path as a function of a specific parameter is quantified through the contribution of different airborne or structure borne sources to the sound response.

c) Numerical and experimental modal analysis is the last method which considers the inherent system features of structures and acoustic spaces [12].

Step 4: Possible solution engineering
Once the SQ problems are pinpointed and the underlying structural or acoustic phenomena is clear, we can utilize the above approaches to model sources, transmission paths and mechanic-acoustical system properties to design and validate our solutions.

After modifying some elements, the sound can be re-measured subjectively through jury group sessions or objective metrics. In an iterative process, the modified results are obtained through experiments, or numerical modeling, or both [12]. The new sounds are then analyzed and modified based on the jury group responses. In real cases, these two steps are carried out iteratively until there is significant convergence towards the target.

2.1.2 Sound sketching

In fig. 2.1, the dashed red rectangle shows the basic approach to the iterative design procedure. A more detailed explanation of the sound design procedure applied in this thesis work can be found in the next section, including sound sketching and jury group evaluation that are applied to obtain the final target sound in this thesis work.

The most traditional but still effective method used for developing new design ideas is pencil and paper. It is a direct way in which one is capable to explore bunches of sketches and also a easy way to decide which ones are more valuable to save based on the evaluations. Hence, designers are provided the chance to improve the initial sketches and produce a set of final designs later on.

The iterative sound design procedure consists of jury group evaluations and analyzing their responses. The final results are obtained after a series of iterations and improvements of the initial sound sketches.

2.1.3 Jury group evaluation

The features of product sound that give an impression of the operation or quality of the product are not necessarily defined by the usual metrics such as loudness or annoyance. Although these metrics are of use in prediction, the general method of forming sound design goals and evaluating product sounds is having people listen to sounds and ask them to rank the sounds using various scales.

This kind of evaluations must be carefully designed to avoid biasing. This experimental design consists of the following steps: establishing the population to be tested, defining the scales, building the stimulus set and establishing the questionnaire and the procedures for data analysis and presentation.

The main goal of the jury group evaluations is to figure out which sound characteristics are more favorable to the product acceptance, and which are less favorable. The second goal is to determine the corresponding design features or degree of modification in the manufacturing process.

The jury group evaluation including both the preparation work and the analyses of the results can be described as:

Step 1: Selecting jury group
The selection of appropriate jury group members is the first step. One thing that should be noticed during selection is a person with specific knowledge of the product and much experience of using it is more likely to associate a sound with a particular mechanism, but he may also have biased opinions on the sound due to his experiences with the product. Hence, the jury group consisting of this kind of listeners might not reflect the attitudes and reactions of most of the average consumer. In the jury group evlations of this thesis work, 2 or 3 jury group members were acousticians. The participation of professional jury group members helped to explain their selections, and also inspirited the response of other participants.

Step 2: Training for jury group members
The information provided during the test should be explained carefully to the jury group so that they know what they are expected to responsed to. Meanwhile, there is also a risk if the information was described too much. In that case, the participants might try to find the "right" answers.

Step 3: Stimulus set
The stimulus set is the set of sounds the jury group are going to listen to. The stimulus set has many factors to consider, such as the duration of sounds, the time between presentations, the sequencing of sounds, the loudness, and so on.

Step 4: Conduct the listening test
An important first issue when conducting listening tests is how to present sounds to the jury. The system playing the sounds should be carefully designed in order to avoid biasing effects due to sound coloration or poor spatial representation. Headphones are

convenient but often suffer from front-back confusion when it is reproducing binaural material. In this thesis work, the sound pieces were played using two loudspeakers to simulated the condition when the users are operating the coffee machine.

Step 5: Data analysis and interpretation

It is important to relate the result to characteristics that the sound designers can focus on. Hence, an open discussion is necessary for the jury group members to explain their choices and describe their feeling towards the sound sketches. One can ask for the reasons why participants like or dislike a sound to gather useful information, e.g. what to focus on or to avoid.

2.2 A brief introduction to TA

Once the product sound has been characterized and the target sound has been established, the actual physical design of the product should be modified to meet the goals as shown in the dashed green rectangle in fig. 1.1.

2.2.1 Noise reduction strategy

Different kinds of noise from different parts of the product can be reduced by applying various noise reduction strategies. In this thesis project, the pump is the most important noise generator. The vibration energy is transferred to the plastic surfaces through multiple paths, and radiated into the surrounding air from the outer plastic surfaces.

Step 1: Modifying the generator

In this thesis work, the generator is the pump placed in the back left side of the coffee machine. Radiated sound is a result of vibration energy being transformed into forces and motions. Since the energy is concentrated at the source, we should apparently consider modifying the generator as an important method in noise reduction. However, such a strategy is sometimes problematic, for instance, if the modification of the generator affects the product performance, efficiency or has an unfavorable effect to the environment.

Step 2: Modifying the transmission paths

The force and motions of the generator can both 'share' energy locally near the excitation point as well as transmit it remotely through air or structure. The transmission path plays a significant role in modifying the amplitude and the frequency content of the noise energy, and therefore, can be of importance for consequential sound design. In most of the situations, the transmission path consists of a combination of mechanical components. Consequently, the treatments for different elements can vary considerably.

As a rule of thumb, two traditional methods can be implemented for transmission component modification. The first one is dissipation such as damping or absorption, while the other one is reflection of the energy. Both methods are discussed in detail in subsection 2.2.2.

Properly designed damping treatment can be effective if the structure is somewhat light and flexible. On the other hand, traditional damping treatments are less effective for stiffening panel structures. This is attributable to the phenomenon that a small portion of the potential energy of vibration is stored in the attached damping component, and the treatment only dissipates a fraction of the energy that it stores.

Step 3: Modifying the radiating surfaces

If the noise energy reaches the external surface or openings of the product, it can be radiated into the air. We can modify the radiating sound by modifying the structure or the surrounding air, or both of them.

The radiation efficiency can be reduced by adding a commercially available decoupling component to the product, or by making openings on the structure. Such a structure may shake, but with reduced radiation efficiency.

2.2.2 Structure borne sound reduction

In the following text, some basic methods to attenuate structure borne sound are presented. The quality of the vibration isolation is described through different quantities, for instance, the transmission loss, the insertion loss, the transmissibility, or reduction index.

The transmission loss is defined as the ratio between transmitted power and incident power. The insertion loss is defined as the ratio between the transmitted power with insulator and transmitted power without insulator. The transmissibility is defined as the level difference on both side of the insulator. The relationship between the reduction index and the transmission factor can be described by the logarithm function.

Impedance change along the propagation path

The majority of noise control treatments are based on the fact that partial reflection of waves takes place when the impedance of the medium suddenly changes. However, the changes in impedance only reflect energy instead of reducing it, such as dissipating the energy into heat by damping.

The modification of impedance can be implemented mainly in two ways: sudden changes in the geometry of the product such as cross sections or in the material properties, for instance, Young's modulus and density.

Method 1: Change of the cross section

The most conventional way to adjust the impedance is to vary the cross section. However, a dramatic change of thickness is required to obtain a significant reduction as illustrated in fig. 2.2, which is obviously ineffective and lack applicability. In practice, changes of the cross section will be no less than a factor of 5, which results in a reduction index of around 3 dB. Coupling a number of beams with different cross sections can achieve a further improvement of the reduction index.

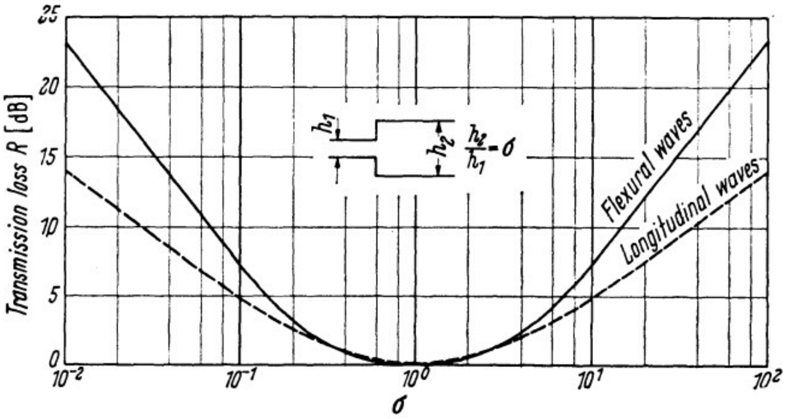


Figure 2.2: Reduction index with the ratio of impedance of both beams [13].

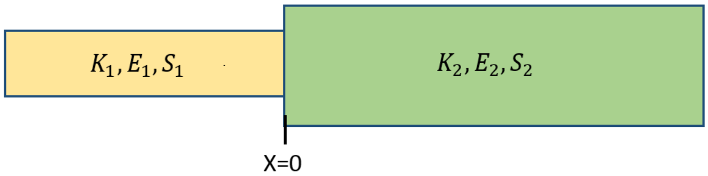


Figure 2.3: Two semi-infinite beams.

The derivation of the reduction index is shown in the following text for the interested reader:

Figure 2.3 shows two semi-infinite beams coupled together at $x = 0$. The longitudinal sound wave comes from the left side, and will be partially reflected at the interface between the beams.

The displacement at the left side on the interface($x \leq 0$) can be expressed as:

$$\xi_1(x = 0,\omega) = \xi_{1+}(\omega) + \xi_{1-}(\omega), \tag{2.1}$$

where $\xi_{1+}(\omega)$ and $\xi_{1-}(\omega)$ are the incident and reflected waves respectively, expressed in meters.

The displacement on the right side of the interface can be written as:

$$\xi_2(x = 0,\omega) = \xi_{2+}(\omega), \tag{2.2}$$

where $\xi_{2+}(\omega)$ is the displacement when waves propagate on the side $x \geq 0$, expressed in meters.

The displacement and forces at the boundary on both sides must be identical to fulfill the boundary condition. Hence, we can obtain the boundary conditions as:

$$Displacement : \xi_{1+}(\omega) + \xi_{1-}(\omega) = \xi_{2+}(\omega), \tag{2.3}$$

$$Force: jK_1E_1S_1[\xi_{1+}(\omega) - \xi_{1-}(\omega)] = jK_2E_2S_2\xi_{2+}(\omega), \tag{2.4}$$

where K is wave number, E is Young's modulus, S is the area of cross section.
It is obvious to derive the equation as below:

$$jK_1E_1S_1[2\xi_{1+}(\omega) - \xi_{2+}(\omega)] = jK_2E_2S_2\xi_{2+}(\omega), \tag{2.5}$$

From eq. (2.5), we derive the transmission factor:

$$t = \frac{\xi_{2+}}{\xi_{1+}} = \frac{2K_1E_1S_1}{K_1E_1S_1 + K_2E_2S_2} = \frac{2\rho_1c_1S_1}{\rho_1c_1S_1 + \rho_2c_2S_2} = \frac{2Z_1}{Z_1 + Z_2}, \tag{2.6}$$

where ρ is the density of the material, c is the speed of sound in the material, Z_n is the impedance, $Z_n = \rho_n c_n S_n$.
The reflection factor can be derived as:

$$r = \frac{\xi_{1-}}{\xi_{1+}} = \frac{K_1E_1S_1 - K_2E_2S_2}{K_1E_1S_1 + K_2E_2S_2} = \frac{\rho_1c_1S_1 - \rho_2c_2S_2}{\rho_1c_1S_1 + \rho_2c_2S_2} = \frac{Z_1 - Z_2}{Z_1 + Z_2}, \tag{2.7}$$

The ratio of the transmitted power to the incident power is

$$\tau = \frac{W_2}{W_{in}} = \frac{|\xi_{2+}|^2\Re(Z_2)}{|\xi_{1+}|^2\Re(Z_1)} = |t|^2\frac{\Re(Z_2)}{\Re(Z_1)} = \left|\frac{2Z_1}{Z_1 + Z_2}\right|^2\frac{\Re(Z_2)}{\Re(Z_1)}, \tag{2.8}$$

where W_1 is the incident power, W_2 is the transmitted power.
The reflection coefficient is

$$\rho = \frac{W_{1-}}{W_{in}} = \frac{|\xi_{1-}|^2\Re(Z_1)}{|\xi_{1+}|^2\Re(Z_1)} = |r|^2 = \left|\frac{Z_1 - Z_2}{Z_1 + Z_2}\right|. \tag{2.9}$$

The ratio between the impedance of both beams is expressed as

$$\alpha = \frac{\rho_2c_2S_2}{\rho_1c_1S_1}. \tag{2.10}$$

Using eqs. (2.7) to (2.9), the reduction index then can be expressed as:

$$R = -10\log(\tau) = 20\log\left[\frac{\alpha^{-1/2} + \alpha^{1/2}}{2}\right]. \tag{2.11}$$

For bending waves, a similar reduction index can be obtained as:

$$R = 20\log\left[\frac{0.5\sigma^{-2} + \alpha^{-1/2} + 1 + \alpha^{1/2} + 0.5\sigma^2}{\sigma^{-5/4} + \sigma^{-3/4} + \sigma^{5/4} + \sigma^{3/4}}\right], \tag{2.12}$$

where σ stands for the ratio of thicknesses of both beams [13].

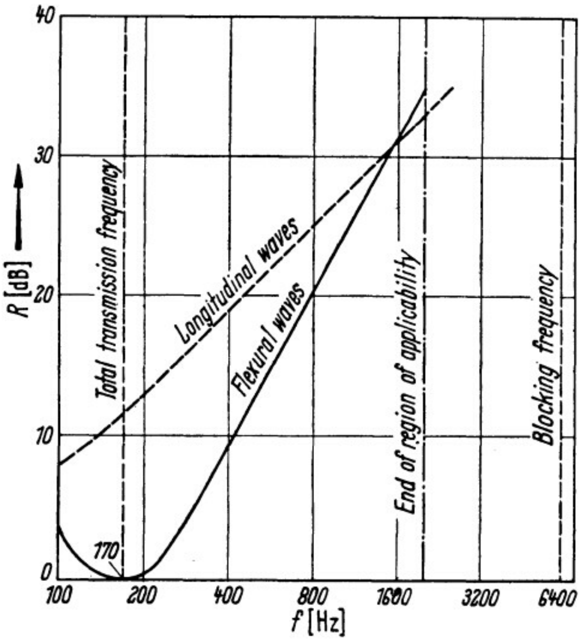


Figure 2.4: Reduction index of elastic interlayer [13].

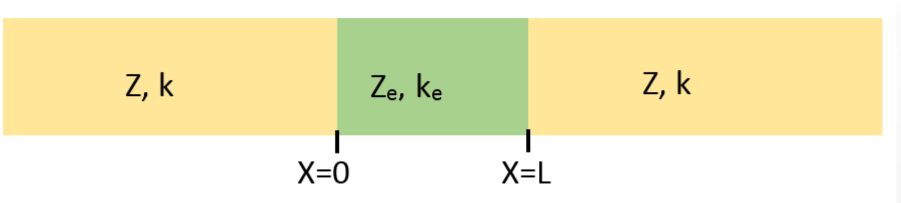


Figure 2.5: Elastic interlayer.

Method 2: Elastic interlayer

For the method of elastic interlayer, the reduction index is dependent on the ratio of stiffness between the interlayer and the material that vibrates in phase on both sides of the interlayer.

Under the condition of same elastic interlayer material and size for the longitudinal wave, the bending wave undergoes less attenuation than the longitudinal waves. This is illustrated in fig. 2.4. Therefore it is important to shift the frequency of total transmission as low as possible, for example, we can use a thick interlayer to get a low transmission, which also guarantees low total transmission frequency.

The derivation of the reduction index is shown in the following text for the interested reader:

The elastic interlayer illustrated in fig. 2.5 is characterized as the middle layer material ($0 \leq x \leq L$) with smaller impedance in comparison to the material on either side $Z_e \ll Z$. The transmission ratio is determined by the stiffness of the interlayer and the stiffness of the material, which vibrates in phase on both sides of the interlayer.

When only longitudinal waves are taken into consideration, the boundary conditions at $x = 0$ and $x = L$ are the same as discussed in the case of sudden change in cross section. Hence, the displacement at $x = 0$ have the relation of

$$\xi_{1+}(\omega) + \xi_{1-}(\omega) = \xi_{e+}(\omega) + \xi_{e-}(\omega). \tag{2.13}$$

At $x = L$,

$$\xi_{e+}(\omega) + \xi_{e-}(\omega) = \xi_{2+}(\omega) + \xi_{2-}(\omega). \tag{2.14}$$

The forces excited by the propagating wave at $x = 0$ have the relation:

$$jK_1E_1S_1[\xi_{1+}(\omega) + \xi_{1-}(\omega)] = jK_2E_2S_2[\xi_{e+}(\omega) + \xi_{e-}(\omega)]. \tag{2.15}$$

where K is the wave number, E is the Young's modulus, and S is the area of cross section.

The forces at $x = L$ have the relation:

$$jK_eE_eS_e[\xi_{e+}(\omega) + \xi_{e-}(\omega)] = jK_2E_2S_2[\xi_{2+}(\omega) + \xi_{2-}(\omega)], \tag{2.16}$$

From eqs. (2.13) to (2.16), the transmission coefficients is derived as:

$$\tau = \frac{W_2}{W_{in}} = \frac{|v_{2+}|^2\Re(Z_2)}{|v_{1+}|^2\Re(Z_1)} = \frac{|\xi_{2+}|^2\Re(Z_2)}{|\xi_{1+}|^2\Re(Z_1)} = \frac{1}{\cos^2(K_eL) + \frac{1}{4}\left|Z/Z_e + Z_e/Z\right|^2 \sin^2(K_eL)}, \tag{2.17}$$

where W is the sound power, Z is the wave impedance of the semi-infinite beam and Z_e is the wave impedance of the infinite beam. According to eq. (2.17), the total transmission is obtained when

$$\cos^2(K_eL) + \frac{1}{4}\left|Z/Z_e + Z_e/Z\right|^2 \sin^2(K_eL) = 1, \tag{2.18}$$

which means exactly half a wavelength or multiples of half a wavelength is equal to the length of the finite beam. In this case, the transmission loss depends only on the damping of the beam in the middle.

As mentioned above, the elastic layer has a much smaller impedance than the other materials ($Z_e \ll Z$), and in practice the elastic layer is often much thinner, which enables us to simplify the transmission coefficient as:

$$\tau = \frac{1}{1 + \frac{1}{4}\left|\frac{Z}{Z_e} + \frac{Z_e}{Z}\right|^2 (k_eL)^2} = \frac{1}{1 + \frac{1}{4}\left|\frac{ES}{E_eS_e}\frac{L}{\lambda/2\pi}\right|}, \tag{2.19}$$

where λ is the wavelength, E is the Young's modulus, S is the area of cross section. We can reform eq. (2.17) as:

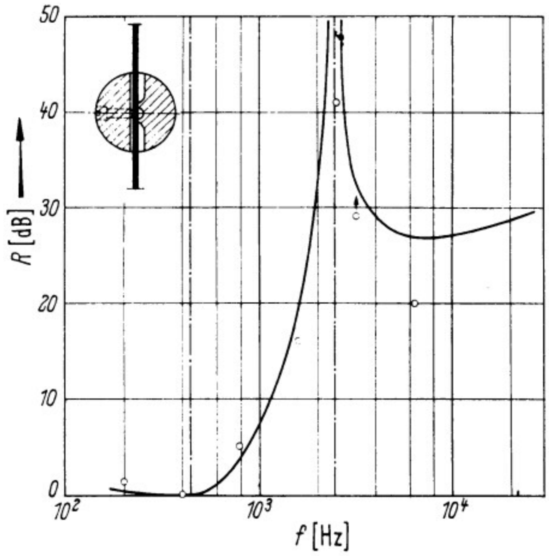


Figure 2.6: Reduction index of blocking mass [13].

$$\tau = \frac{1}{1 + \frac{1}{4}\left|\frac{ES}{E_e S_e}\frac{L}{\frac{2\pi f}{c}/2\pi}\right|} = \frac{1}{1 + \frac{1}{4}\left|\frac{ES}{E_e S_e}\frac{L_c}{f}\right|}, \tag{2.20}$$

Method 3: Blocking mass

The reduction is dependent on the ratio between the mass of the blocking element and the mass that vibrates in phase with the blocking mass on its both sides under the condition that the center of the mass is in the middle line of the beam.

Rather than total transmission, total reflection of certain frequencies is obtained as shown in fig. 2.6, which leads to significant attenuation. This happens only when a bending near field is excited on the receiving side as the bending near field helps to fulfill the boundary conditions but is of no effort to the radiation.

If the blocking mass is not located symmetrically, the analysis would be more complicated. For example, placing the blocking mass on one side of the beam means that the rotation centers of the mass and the beam are different. As a result, the mass will have a tangential movement along the beam. This forces the beam to also move in the tangential direction at this point because the beam and mass are coupled rigidly. Such a rotation introduces a moment to the structure and thus causes bending waves. The moment impedance comprises the impedance of the mass and the impedance of the beam. If the moment impedance is high compared to the wave impedance of longitudinal waves, the transmission from bending waves to longitudinal waves is high. This is also true if the center of the mass is far from the neutral axis of the beam.

The derivation of the reduction index is shown in the following text for interested

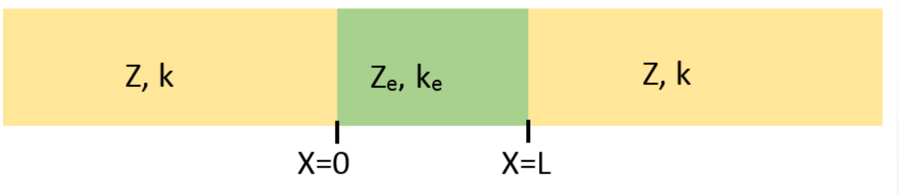


Figure 2.7: Blocking mass

reader:

In the case of blocking mass as fig. 2.7, we require that the impedance of the middle part of beam to be greater than the surrounding impedance. Consequently, assuming $Z_e \gg Z$, the transmission coefficient can be derived as:

$$\tau = \frac{1}{1+\frac{1}{4}\left|\frac{Z_e K_e L}{Z}\right|^2} = \frac{1}{1+\frac{1}{4}\left|\frac{\omega\rho_e S_e L}{\rho c S}\right|^2} = \frac{1}{1+\frac{1}{4}\left|\frac{\rho_e S_e}{\rho S}\frac{L}{\lambda/2\pi}\right|^2}, \tag{2.21}$$

where Z is the impedance, K is wave number, ρ is density, and S is the area of cross section, c is the speed of sound.

Frequency response function of impedance changing method

Figure 2.8 illustrates the frequency response function of the velocity on a semi-infinite beam under longitudinal wave incident, calculated using equations eq. (2.7), eq. (2.19), and eq. (2.21). The models are calculated under the following condition:

a) Semi-infinite beam with loss factor $\eta = 0.02$; Young's modulus $E = 3.2 \times 10^9$; Poisson's ratio $\nu = 0.35$; density $\rho = 1190\ kg/m^3$; thickness $t = 15$ mm;

b) Two semi-infinite beams with the same material and cross section ratio of 4/3;

c) Elastic layer foam with loss factor $\eta = 0.08$; density $\rho = 300\ kg/m^3$, Young's modulus $E = 3.3 \times 10^8$; thickness $t = 15$ mm;

d) Blocking mass with loss factor $\eta = 0.0004$; Young's modulus $E = 2.8 \times 10^{11}$; Poisson's ratio $\nu = 0.33$; density $\rho = 7850\ kg/m^3$; thickness $t = 15$ mm.

Figure 2.9 illustrates the frequency response function of displacement on a semi-infinite beam under bending wave incident. The models are calculated using the following conditions:

a) Finite beam of length 100 meters with loss factor $\eta = 0.02$; Young's modulus $E = 3.2 \times 10^9$; Poisson's ratio $\nu = 0.35$; density $\rho = 1190 kg/m^3$; thickness $t = 15$ mm;

b) Two finite beams of length 50 meters with the same material and cross section ratio of 4/3;

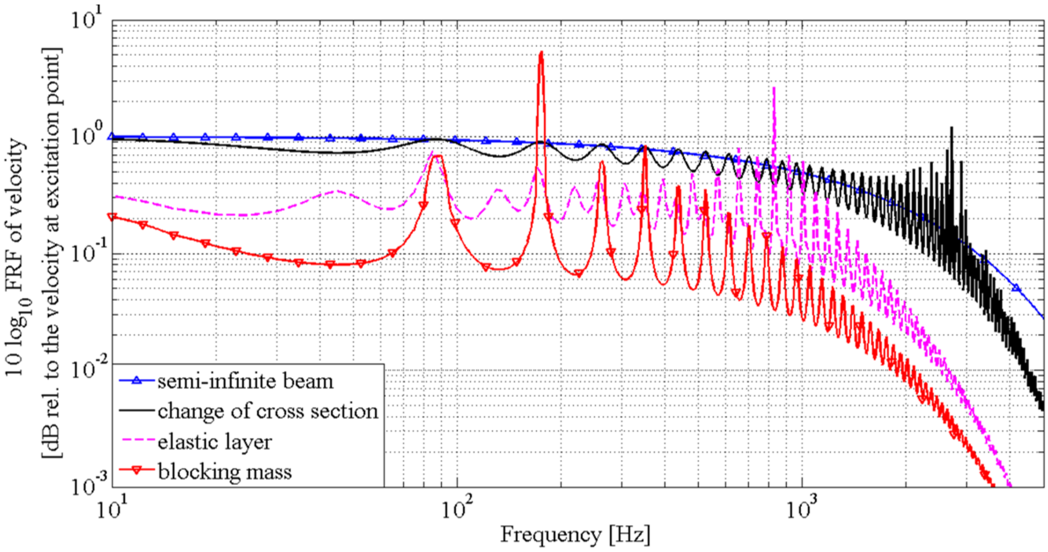


Figure 2.8: FRF of longitudinal wave incident in semi-infinite beams.

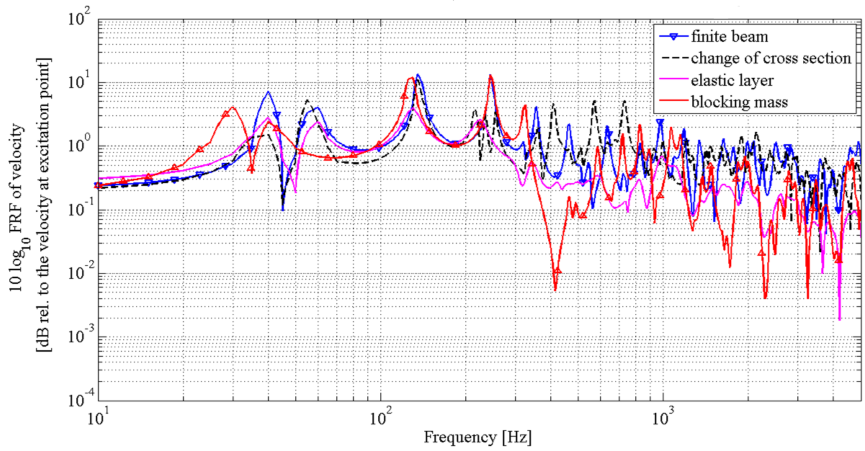


Figure 2.9: FRF of bending wave incident in finite beams

c) Elastic layer foam with loss factor $\eta = 0.08$; density $\rho = 300\ kg/m^3$, Young's modulus $E = 3.3 \times 10^8$; thickness $t = 15$ mm; length of the whole structure is 100 meters;

d) Blocking mass with loss factor $\eta = 0.0004$; Young's modulus $E = 2.8 \times 10^{11}$; Poisson's ratio $\nu = 0.33$; density $\rho = 7850\ kg/m^3$; thickness $t = 15$ mm; length of the whole structure is 100 meters;

In fig. 2.8, all three methods introduce more reduction at higher frequencies which is reasonable due to the fact that more periods in propagation at high frequencies than at low frequencies. The method of sudden change in cross section is of the lowest reduction among all methods as discussed before. The method of blocking mass has higher reduction compared to the method of elastic layer at low frequencies, but the difference gets lower with increasing frequencies, which can also be proved by eqs. (2.20) and (2.21). The cases of bending incident waves are more complicated. Meanwhile, the boundary causes more cancellation during wave propagation, especially for the case of elastic layer and blocking mass, however, the trends seen in the reduction are similar to the case of longitudinal wave.

Damping of structure borne sound

In the previous sections, we discussed some commonly applied techniques for attenuating structure borne sound. In all these methods, energy was only reflected at impedance discontinuities. However, energy is not consumed but still exists somewhere in the structure in the form of vibrational energy. In this section, various damping mechanisms which transform vibration energy into another form of energy like heat are presented.

The damping mechanisms can be classified as:

a) non-material damping such as acoustic radiation, and viscous friction between surfaces

b) material damping, for instance, viscoelastic effects

Method 1: Non-material damping mechanisms

Non-material damping mechanisms are cheap and economical methods for noise reduction. Although this method is not applied in this thesis work, some basic theories will be presented here.

a) Gas pumping

   Let us first look at the medium between two independently vibrating plates, which is pumped along the plate because of the vibration of the plates.

   At the surfaces of the plates, the tangential velocity of the medium is equal to that of the plates. Thus, in the process of pumping gas through the gap between the plates, the energy is dissipated by viscous friction. The loss factor is approximated by:

$$\eta = \frac{\rho\delta}{m''}\frac{\lambda_p}{2\pi d}, \tag{2.22}$$

   where m'' is the mass per unit area of the plates, λ_p is the wavelength on the plate and d is the distance between the plates, which is supposed to be small. For thin double plates, this damping mechanism can be dominant due to a large loss factor caused by a small mass per unit area.

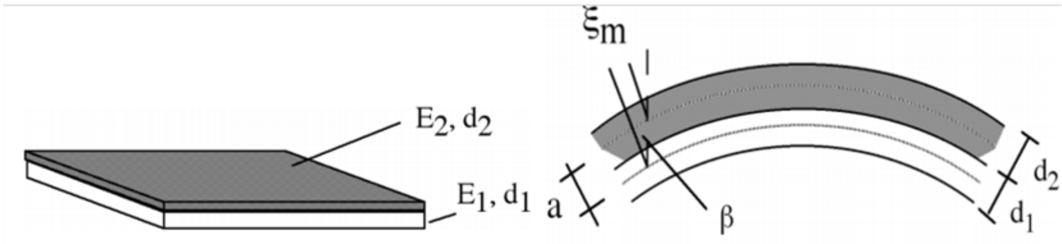


Figure 2.10: Attached damping layer for longitudinal and bending waves [13].

b) Damping at connectors

Different parts of machines and structures are usually connected by bolts, screws, welds or other types of connectors. In practice, these connections are imperfect, and therefore allow relative movement of structures, despite the fact that the displacement might have small-scale amplitudes.

At the contact area, material such as oil, dust, air or even surface roughness acts as springs with damping. Hence, the relative movement between different parts will introduce supplementary damping effect.

Method 2: Material damping

Material damping is widely applied in this thesis to reduce the radiation from the pump and the energy transferred by different paths. Different materials result in different damping, for example, steel has a relatively small loss factor while for polymer materials the loss factor increases.

Ignoring the density, Young's modulus and Poisson's number of the material, the loss factor is related to the "history" of the material, for example, previous deformation and ageing processes. The material damping is also dependent on the frequency and environment temperature. Moreover, small changes in the molecular structure of the materials result in significant variation of the loss factor.

a) Damping layers

The total loss factor is proportional to the thickness times the Young's modulus of the attached damping layer. A larger Young's modulus enables us to utilize a thinner layer while achieving the same damping. Physically, this is because the losses are proportional to the potential energy stored in each layer. If the damping layer is very soft, there is almost no potential energy stored and the losses are very small. The loss factors achieved by this arrangement are obviously not very high.

The derivation of the total loss factor is shown in following text for interested reader:

The concept of damping layers is to add polymer material layers to structures to increase the damping.

The loss factor is defined as:

$$\eta = \frac{E_{loss}}{E_{rev}}, \tag{2.23}$$

where E_{loss} is the lost energy, E_{rev} is the reversible energy. For longitudinal waves, the reversible energy can be calculated as:

$$E_{rev} = \frac{1}{2}(E_1 d_1 + E_2 d_2) \left| \frac{d\xi}{dx} \right|^2, \tag{2.24}$$

where E_1 and E_2 are the Young's modulus, d_1 and d_2 are the thickness for both material, and ξ is the displacement of the neutral axis line of the attached layer.

And the loss energy in both plates can be expressed as:

$$E_{loss} = \frac{1}{2}(\pi\eta_1 E_1 d_1 + \pi\eta_2 E_2 d_2) \left| \frac{d\xi}{dx} \right|^2, \tag{2.25}$$

where E_1 and E_2 are the Young's modulus, d_1 and d_2 are the thickness, η_1 and η_2 are the loss factor for both material, ξ is the displacement of the neutral axis line of the attached layer.

Assuming $\eta_1 E_1 d_1 \ll \eta_2 E_2 d_2$, the loss factor for longitudinal waves with a damping layer attached can be simplified as

$$\eta_1 = \eta_2 \frac{E_2}{d_2}(E_1 d_1 + E_2 d_2), \tag{2.26}$$

For bending waves, the loss energy is slightly different compared to that of longitudinal waves,

$$E_{loss} = \pi\eta_2 E_2 d_2 \left| \frac{d\xi_m}{dx} \right|^2, \tag{2.27}$$

where ξ_m stands for the displacement of the middle layer of the structure, and $\xi_m = a\beta$.

Thus the stored energy thus can be calculated as:

$$E_{rev} = \frac{1}{2} B \left| \frac{d\beta}{dx} \right|^2, \tag{2.28}$$

where B is the bending stiffness, β is the bending angle.

The loss factor can be expressed in the form of

$$\eta_B = \eta_2 \frac{E_2 d_2 a^2}{B}, \tag{2.29}$$

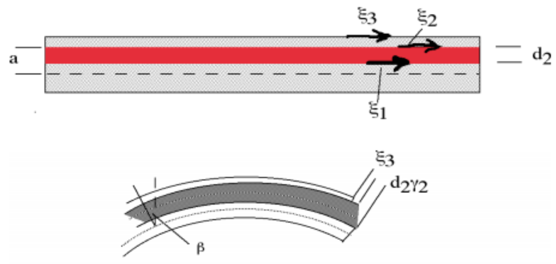


Figure 2.11: Sandwich structure for longitudinal waves and bending wave [13].

where a is the distance between the neutral axis of two layers ($a = (d_1 + d_2)/2$) as shown in Fig. 2.7.

The bending stiffness can be calculated as:

$$B = E_1 \int_{-h_2}^{h_1} y^2 dy + E_2 \int_{h_1}^{h_2} y^2 dy = \frac{E_1}{3}(h_1^3 + h_2^3) + \frac{E_2}{3}(h_2^3 - h_1^3), \tag{2.30}$$

where $h_3 = \frac{d_1}{2}$, $h_2 = \frac{d_1}{2} + d_2$, $h_1 = \frac{d_1}{2}$.

Hence,

$$B \approx \frac{E_1 d_1^3}{12} + E_2 d_2 a^2, \tag{2.31}$$

Similar to the situation with longitudinal waves, a higher Young's modulus leads to higher stiffness.

b) Sandwich constructions

Sandwich constructions enable increased total stiffness which results in a higher bending stiffness at low frequencies and a lower bending stiffness at high frequencies compared to the simple damping layers mentioned above.

The derivation of the total loss factor is shown in following text for interested reader:

Sandwich constructions are essentially analogous to the simple elastic layer. Its major difference is that the movements on both sides of the layer are constrained by the attached plates. Thanks to these constraints and the fact that both plates move distinctly in the tangential direction, the layer is forced to have shear motions.

The shear motion results in loss of energy, which can be expressed by

$$E_{loss} = \pi \eta_2 G_2 d_2 \left| \frac{\xi_2}{d2} \right|^2, \tag{2.32}$$

where G is the shear modulus.

The reversible energy is introduced by the potential energy because of the bending of the structure, which can be written as:

$$E_{rev} = \frac{1}{2} B \left| \frac{d\beta}{dx} \right|^2 . \tag{2.33}$$

The loss factor can be obtained as:

$$\eta_s = \eta_2 \frac{G_2 d_2 \gamma^2}{B K^2 \beta^2}, \tag{2.34}$$

where γ stands for the shear angle.

c) Stiff plate with thin cover plate

In the case of a very thin and stiff damping layer, for instance a damping tape, associated with a thin cover plate, it is obvious that the total stiffness of the construction is only dependent on the stiffness of the lower plate.

Applying the same procedure as in eqs. (2.32) to (2.34), the loss factor can be derived as:

$$\eta_s = \eta_2 \frac{g_d E_3 d_3 a^2}{B \left| 1 + j\eta_2 g_d \right|^2}, \tag{2.35}$$

where the shear parameter $g_d = \frac{G_2 \lambda^2}{4\pi^3 E_3 d_3 d_2} = \frac{G_2 (2\pi f/c)^2)}{4\pi^3 E_3 d_3 d_2}$.

Equation 2.34 shows that the loss factor can be determined by wave number (by frequency), and thus introduce a maximum damping for each case discussed in Method 2. Therefore, additional considerations regarding the construction design is necessary in order to maximize damping in the frequency range of interest.

From eq. (2.34), we are able to conclude that the shear modulus only influences the value of the optimal frequency rather than the value of the optimal loss factor. Consequently, in reality, the shear modulus of the interlayer is not extremely important provided its loss factor is high. This fact allows the implementation of comparatively soft viscoelastic materials.

d) Two stiff plates with a thin visco-elastic interlayer

In practice, both the damping layer and sandwich construction are similar in terms of equivalent weight and the result damping [13]. One minor benefit of the sandwich design would be the frequency dependent bending stiffness. At low frequencies the structure acts as one plate with bending stiffness $B = \frac{1}{12} E_1 (d_1 + d_3)^3$. At high frequencies the plates are decoupled with the bending stiffness approximated by $B = \frac{1}{12} E_1 (d_1^3 + d_3^3)$.

The increased total stiffness results in a higher bending stiffness at low frequencies which decrease with increasing frequencies. The latter might be favorable with regard to sound radiation.

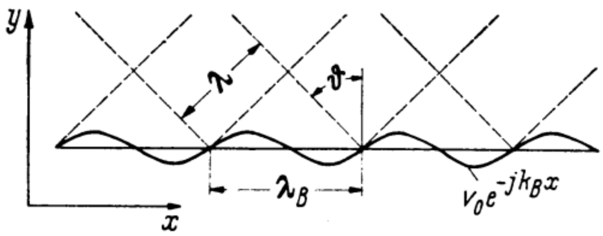


Figure 2.12: Radiation from infinite plates [13].

The efficiency of the sandwich design at low frequencies can be improved by dividing one of the beams (usually the cover beam) into smaller segments. By doing so, the shear motion can be enhanced even at lower frequencies.

Radiated sound from structures

In practice, the sound that is radiated to the surroundings is more important than the structural vibrations. Hence, it is of importance to apply proper methods for sound radiation reduction.

The critical frequency, which is a very important concept in sound radiation, is defined as the frequency for which the speed of sound or the wavelength of the waves in the radiating surface and waves in the surrounding medium are identical. Only an acoustic near field is produced at the surface of an infinite plate but the radiation efficiency is zero when it is below the critical frequency, which will be proved in different cases in the following text.

If the frequency is below the critical frequency for infinite plates, there would be a cancellation between areas with high pressures and low pressures. For finite plates, the two areas will not cancel each other completely. Moreover, the edges do not have such cancelling areas and thus radiate sound. Hence, the radiation efficiency on the baffled plates is dependent on both the size of the plate and the wavelength of the plate. Similar to that of an infinite plate, one can show that only a near field is created below the critical frequency. Above that, there exists only one angle that satisfies the boundary conditions, which is the propagation direction of the radiated sound waves.

The radiated sound power from external excitation is only related to the excitation force and the mass per area of the plate. In other words, one can suppress the radiated sound power using extra mass at the excitation point. In addition, we notice that the stiffness of the plate does not change the result.

The derivation of radiation efficiency is shown in the following text for the interested reader:

Case 1: Sound radiation from infinite plates

An external force can excite vibrations in a structure. The resulting vibration pattern of the structure is determined by the force distribution, the structural eigenfunctions, as well as the excitation frequency.

From this vibration pattern we can obtain a certain velocity distribution by applying the boundary condition to the interface between the structure and surrounding medium.

Figure 2.12 indicates a wave propagating on an infinite plate with the amplitude

$$v(x,\omega) = v_0 e^{-jK_B x}, \tag{2.36}$$

where K_B is the wave number for bending waves.

The wave propagating in the surrounding medium can be described as:

$$p(x,y,\omega) = p_0 e^{-jK_x x} e^{-jK_y y}, \tag{2.37}$$

where $K_x = K_0 \sin\nu$ and $K_y = K_0 \cos\nu$ [13].

At the interface between the plate and medium, the speed of sound must have the same direction and amplitude. This gives us the relation $K_x = K_B$.

At the boundary between the medium and the plate, the velocities in the y-direction must be the same for continuity. Hence,

$$v_y(x,\omega) = \frac{v_0 K_y}{\omega \rho_0} e^{-jK_B x} = v(x,\omega), \tag{2.38}$$

where $K_y = \sqrt{K_0^2 - K_B^2}$.

We can derive the amplitude of the sound wave as [13]:

$$p_0 = \frac{v_0(\omega)\rho_0 c_0 K_0}{sqrt K_0^2 - K_B^2} e^{-jK_x x} e^{-jsqrt K_0^2 - K_B^2 y}, \tag{2.39}$$

The velocity then can be expressed as:

$$v(x,y,\omega) = v_0(\omega) e^{-jK_x x} e^{-j\sqrt{K_0^2 - K_B^2} y}, \tag{2.40}$$

From eq. 2.39, we can see that pressure and velocity are in phase above the critical frequency as in this case $\sqrt{K_0^2 - K_B^2}$ is real. On the other hand, they are out of phase below the critical frequency, due to the imaginary root of $\sqrt{K_0^2 - K_B^2}$, which implies that only an acoustic near field is produced at the surface of the plate, and the radiation efficiency is zero.

Case 2: Sound radiation from finite plates in a rigid baffle

In the case of finite plates, we can use the previously obtained results for infinite plates only if the frequencies are considerably higher than the critical frequency. If the frequency is below the critical frequency, the finite dimensions and the boundary conditions will influence the radiation efficiency significantly.

The radiated pressure from an arbitrary vibration pattern on the baffled plate can be written as:

$$p(R,\omega) = \frac{j\omega\rho_0}{2\pi R}e^{-jK_0R}v(K_x,K_y), \tag{2.41}$$

where R is the distance to the source.

In the one dimensional case, we assume $K_y = 0$. The wave number in the x direction is $K_x = \frac{2\pi}{\lambda_B}$.

Similar to the discussion for infinite plates, one can show that when the wave number K_0 in the surrounding medium is less than K_x, only a near field is created. When $K_0 \geq K_x$, there exists only one angle that satisfies the relation $K_x = K_0 sin\nu$, where the angle ν is the propagation direction of the radiated sound waves.

Case 3: Sound radiation with external excitation

If both an external force and the resulting radiation are studied, the excitation will consist of two terms, the external excitation p_e and the pressure field due to radiation from the plate p_r.

$$p(x,y,\omega) = p_e(x,y,\omega) + p_r(x,y,\omega), \tag{2.42}$$

So the bending wave equation can be rewritten as

$$[K_x^4 + 2K_x^2K_y^2 + K_y^4 - K_B^4]v_p(K_x,K_y,\omega) = \frac{j\omega}{B}[p_e(x,y,\omega) + p_r(x,y,\omega)], \tag{2.43}$$

where $K_B^4 = (m^{''}\omega^2)/B$, $m^{''}$ is the mass per unit area of the radiating plate.

The excitation pressure is described as:

$$p_e(K_x,K_y,\omega) = P_0(\omega), \tag{2.44}$$

where P_0 is the pressure per unit area at $(x = 0, y = 0)$ caused by A point force.

For an infinite plate, we can express the radiated pressure as:

$$p_r(K_x,K_y,\omega) = Z_{rad}v_p(K_x,K_y,\omega), \tag{2.45}$$

where the radiation impedance $Z_{rad} = \frac{\rho_0 c_0 K_0}{\sqrt{K_0^2 - K_x^2 - K_y^2}}$.

And the total impedance can be calculated by:

$$Z = Z_p + Z_{rad}, \tag{2.46}$$

where Z_p is the plate impedance,

$$Z_p = \frac{B}{j\omega}[K_x^4 + 2K_x^2K_y^2 + K_y^4 - K_B^4], \tag{2.47}$$

The ratio between the external excitation and propagating velocity on excitation in eq. 2.44 can be written as:

$$v_p(K_x,K_y,\omega) = \frac{p_e(x,y,\omega)}{Z_p + Z_{rad}}, \tag{2.48}$$

From eq. (2.45), it is obvious that the radiation loading $\rho_0 c_0$ affects the propagating velocity only when the radiation impedance is large in comparison with the impedance of the plate.

Assuming that $K_B^2 \gg K_0^2$, and $m^{''}\omega^2 \gg \rho_0 c_0$, the radiated sound power is derived as:

$$W_{rad} = \frac{1}{8\pi^2}\int_0^{2\pi}\int_0^{K_0} Re(Z_{rad})\frac{|P_0|^2}{|Z_p + Z_{rad}|^2}K_r dK_r d\psi, \tag{2.49}$$

Where K_r stands for the wave number in cylindrical coordinates, and ψ is the angle in the cylindrical coordinates.

When considering frequencies below the critical frequency, for which the radiation loading is negligible ($m^{''}\omega^2 \gg \rho_0 c_0$), we can simplify the radiated sound power to:

$$W_{rad} = \frac{P_0^2 \rho_0}{4\pi m''^2 c_0}. \tag{2.50}$$

Frequency response function of the damping method

Figure 2.13 illustrates the frequency response function of displacement on a finite beam under longitudinal wave incident, calculated using eqs. (2.26), (2.34) and (2.50). The models are calculated using the following conditions:

a) Infinite plate with loss factor $\eta = 0.02$; Young's modulus $E = 3.2 \times 10^9$; Poisson's ratio ν; density $\rho = 1190\ kg/m^3$; thickness $t = 15$ mm.

b) Infinite plate attached with damping layer with loss factor $\eta = 0.08$; density $\rho = 300\ kg/m^3$, Young's modulus $E = 3.3 \times 10^8$; thickness $t = 15$ mm;

c) Sandwich structures with damping layer foam with loss factor $\eta = 0.08$; density $\rho = 300\ kg/m^3$, Young's modulus $E = 3.3 \times 10^8$; thickness $t = 15$ mm and a thin cover of the same material with the plate.

d) Attached mass with loss factor $\eta = 0.0004$; Young's modulus $E = 2.8 \times 10^11$; Poisson's ratio $\nu = 0.33$; density $\rho = 7850\ kg/m^3$; dimension of mass $t = 50$ mm.

Figure 2.14 illustrates the frequency response function of displacement on a finite beam under bending wave incident. The models are calculated under the following condition:

a) Finite plate with loss factor $\eta = 0.02$; Young's modulus $E = 3.2 \times 10^9$; Poisson's ratio $\nu = 0.35$; density $\rho = 1190\ kg/m^3$; thickness $t = 15$ mm; the dimension of the plate is $100 \times 100m^2$.

b) Finite plate attached with damping layer with loss factor $\eta = 0.08$; density $\rho = 300$ kg/m^3, Young's modulus $E = 3.3 \times 10^8$; thickness $t = 15$ mm; the dimension of the plate is $100 \times 100m^2$

c) Sandwich structures with damping layer foam with loss factor $\eta = 0.08$; density $\rho = 300$ kg/m^3, Young's modulus $E = 3.3 \times 10^8$; thickness $t = 15$ mm and a thin cover of the same material with the plate; the dimension of the plate is $100 \times 100m^2$.

d) Attached mass with loss factor $\eta = 0.0004$; Young's modulus $E = 2.8 \times 10^11$; Poisson's ratio $\nu = 0.33$; density $\rho = 7850$ kg/m^3; dimension of mass $t = 50$ mm; the dimension of the plate is $100 \times 100m^2$.

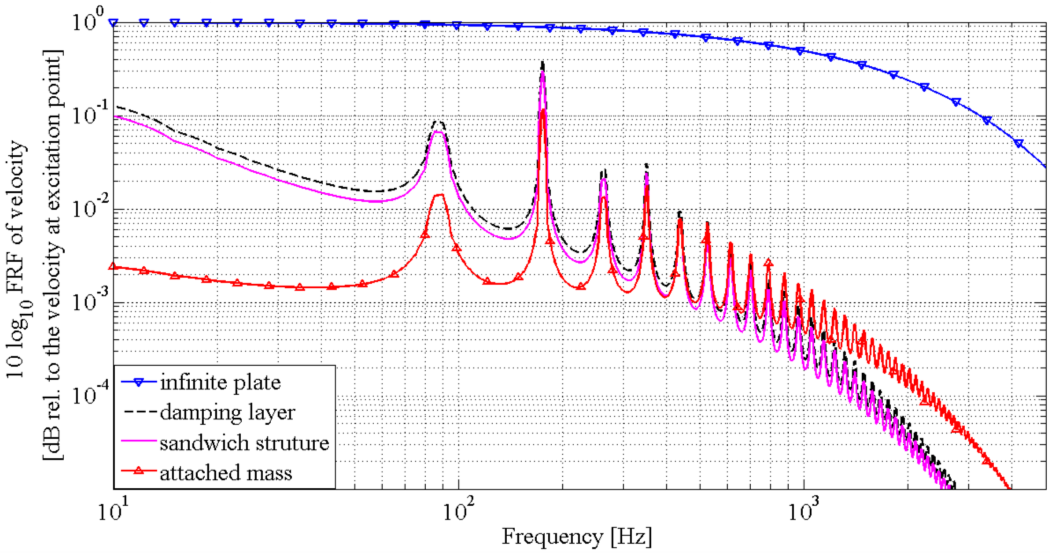


Figure 2.13: FRF of longitudinal wave incident in infinite plate.

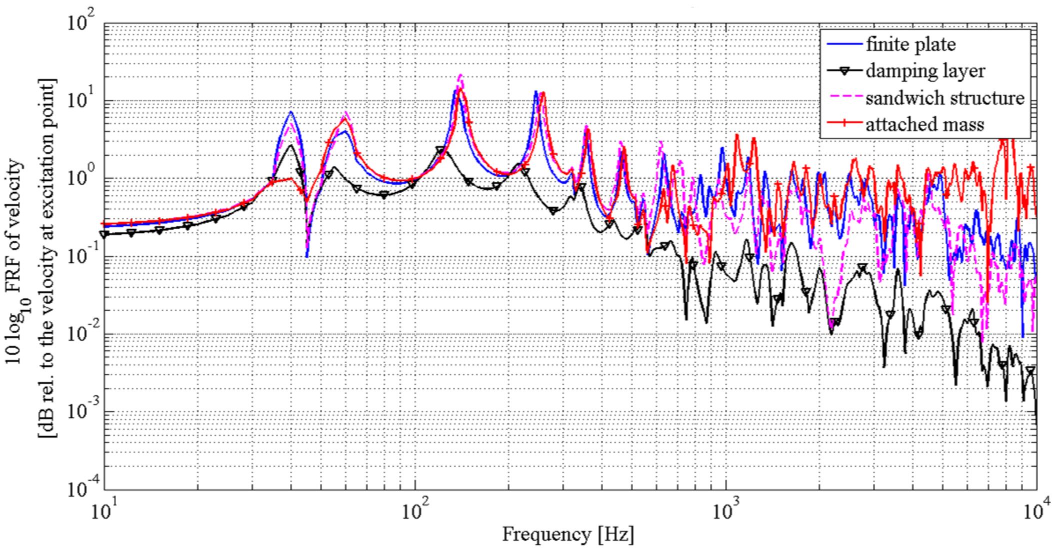


Figure 2.14: FRF of bending wave incident in finite plate.

From fig. 2.13 and fig. 2.14, one can see that the method of damping layer and sandwich structure introduce similar reduction, both are more effective at high frequencies. The method of attached mass introduce higher reduction at low frequencies compared to the other two methods, but lower reduction at high frequencies.

3

Methodology

To suggest reasonable mechanical improvements, knowledge about the structure-borne sound and the sound radiation is required. It is of great importance to study the correlation between the structural vibration and the radiated sound.

An iterative sound sketching method was implemented to create the target sound. The sound sketches were designed for the first four seconds of the radiated sound from the coffee machine. The sound sketches were evaluated by several jury groups and improved according to evaluation results to obtain an appropriate target sound that is structurally applicable as well.

The initially designed target sounds were improved using both the feedback obtained from the evaluations and the vibro-acoustic findings. A second jury group evaluation was conducted to investigate the improved designed product sound.

3.1 Vibro-acoustic study

The sound radiated from the coffee machine was recorded in mono and the power spectrum was analyzed in the time-frequency domain to obtain the characteristics and ranges of interest for sound quality engineering.

Furthermore, the coffee machine was disassembled to study the vibration source and to obtain the frequency response functions of different transfer paths. The correlation between sound and vibration was studied to get an impression of the machinery components we should focus on to improve the product sound quality.

3.1.1 Sound quality assessment and diagnosis

Sound recording implementation

The sound of the coffee machine was recorded in the studio of A2Zound, Semcon, with fixed positions for both the machine and microphone. The microphone was set to a position similar the ear height of a typical user (1.6 m height and 0.5 m to the user).

The equipment used for measurements in this thesis work is listed in Table 3.1:

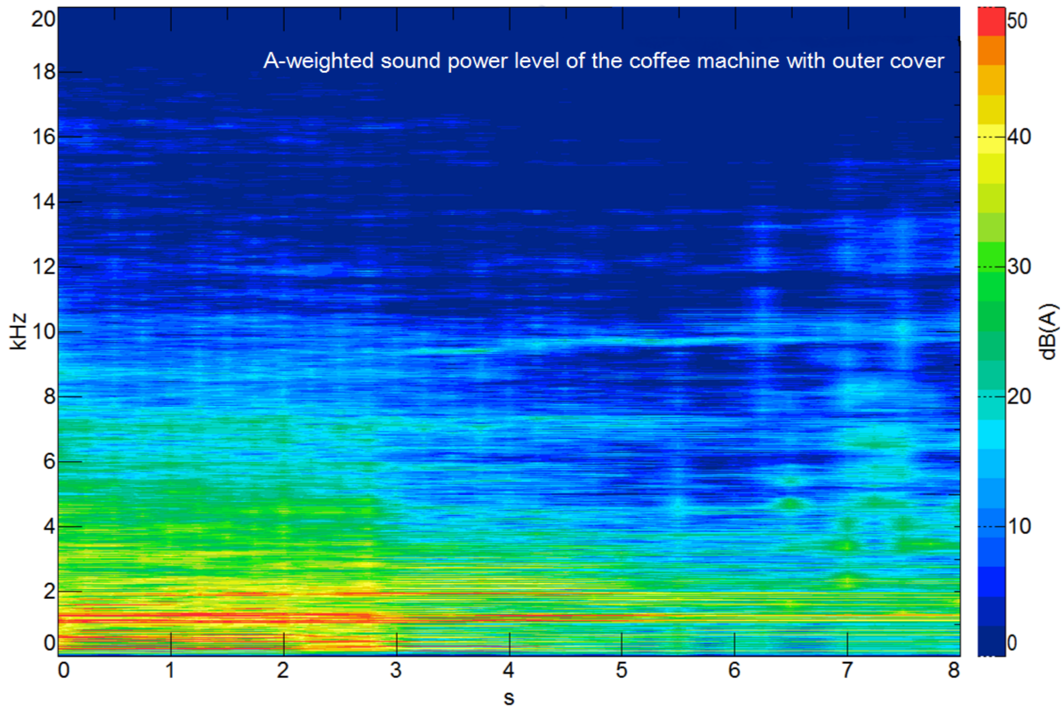
Table 3.1: The equipment used for measurement .

No	Name	Description	Manufactor	Series no.	Sensitivity	Point id
1	MPA205	mic	BSWA	440423	28.2 mV/Pa	ear
2	3031AC	accelerometer	PCB	518	10.0569 mV/g	1-1
3	3031AC	accelerometer	PCB	564	10.4502 mV/g	2-1
4	AC07-0003	accelerometer	B&K	10348	10.1201 mV/g	1-2
5	4507	accelerometer	B&K	2030918	10.0926 mV/g	2-2
6	AC07-0007	accelerometer	B&K	10889	96.9815 mV/g	1-3
7	352B10	accelerometer	PCB	83801	10.113 mV/g	1-4
8	AC06-0010	accelerometer	B&K	60260	10.1159 mV/g	1-5
9	3031AC	accelerometer	PCB	576	10.5018 mV/g	1-6
10	AC07-0005	accelerometer	B&K	10347	96.8039 mV/g	2-6
11	AC07-0004	accelerometer	B&K	30739	99.0393 mV/g	1-7
12	3031AC	accelerometer	PCB	572	10.4753 mV/g	2-7
13	SCADAS	analyze software	LMS	N/A	N/A	N/A

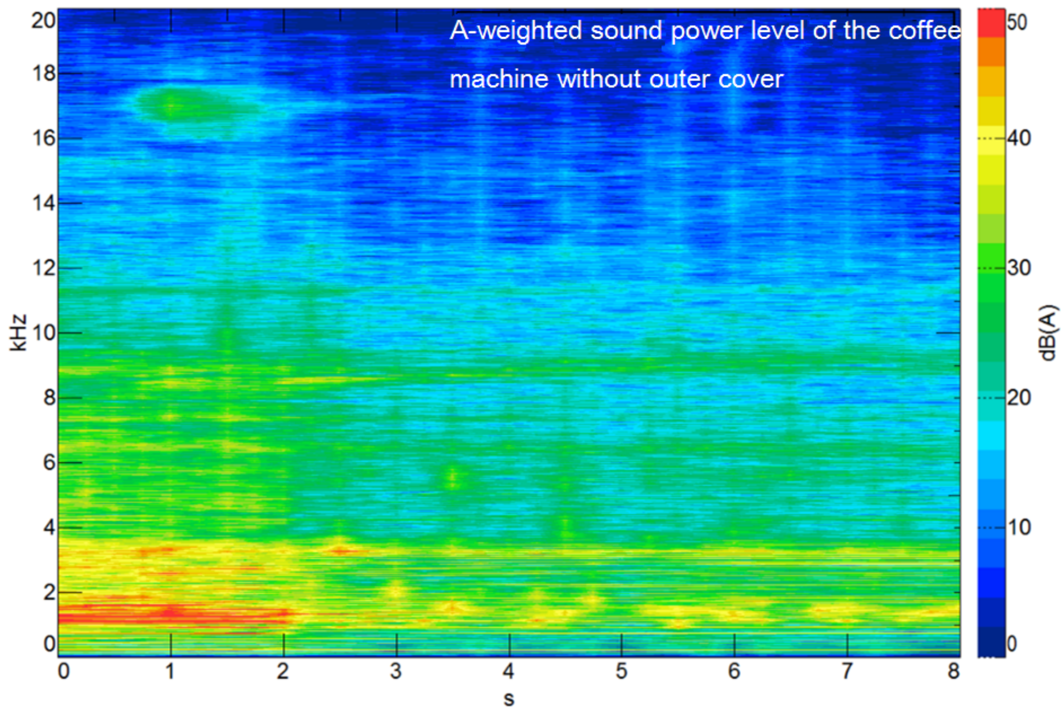
Sound quality assessment and diagnosis

The sound of the coffee machine can be divided into two parts from both hearing and functional aspects. The first part is when the pump acts as the main vibration source to supply the water from the water tank to the upper heating area. The following part is when the heated water flows through the coffee capsule. Hence, the first part was decided to be studied in this thesis work after discussion among two acousticians and me.

For convenience to analyze the sound radiated from each component of the coffee machine, the plastic outer cover was removed. Figure 3.1 illustrates the radiated sound power level in dB in the time-frequency domain, which enables us to analyze the annoying characteristics in the time-frequency domain and consequently get the following conclusion:



(a) covered



(b) uncovered

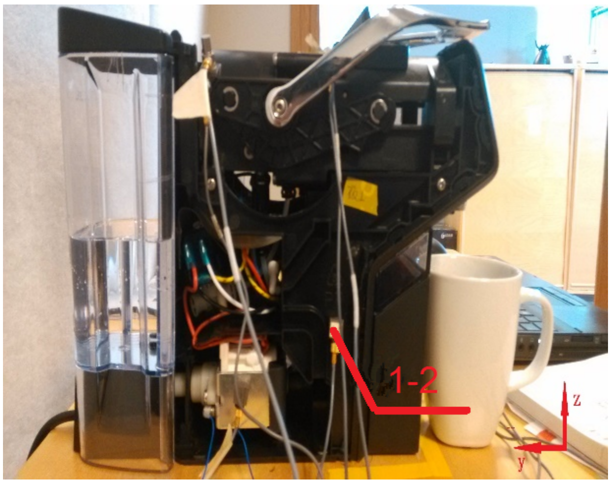
Figure 3.1: A-weighted sound power level of the original coffee machine with (a) and without (b) plastic covers.

a) The most problematic period is the start up period (0-4 s in fig. 3.1).

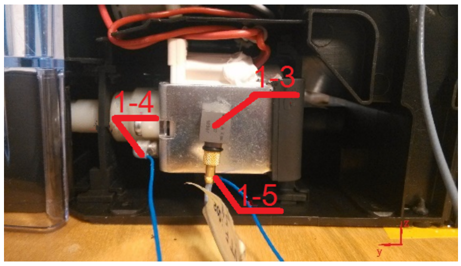
b) The most interesting frequency ranges for noise reduction are from 1 to 3 kHz, from 4 to 10 kHz, and from 15.8 to 17.8 kHz. Although 15.8 kHz is already high for users to perceived especially with the outer cover of the coffee machine, it is of meaning to show the methodology by making the sound sketching of reduced high frequency components and applying the same solution on the coffee machine.

c) The sound is too loud and sharp . The rattling sound is also annoying.

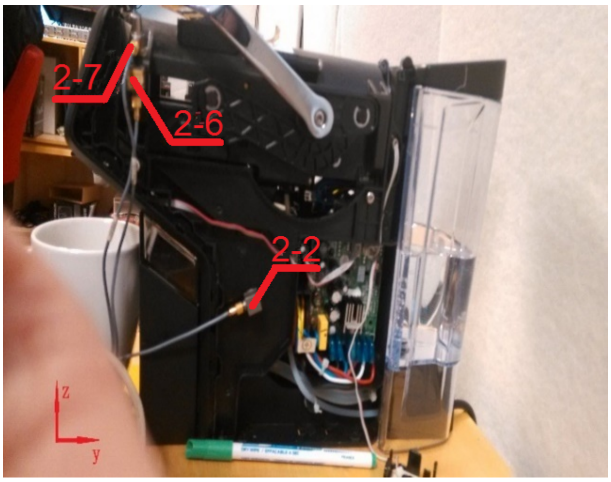
3.1.2 Physics problem identification



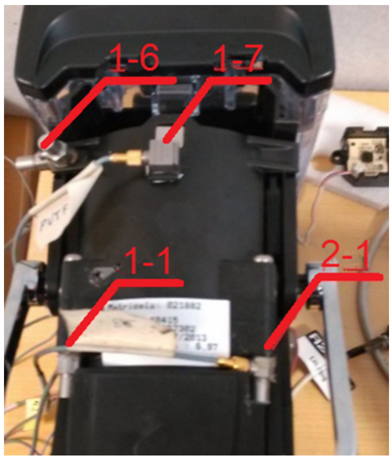
(a) left side



(b) pump



(c) right side



(d) top

Figure 3.2: Positions of accelerameters.

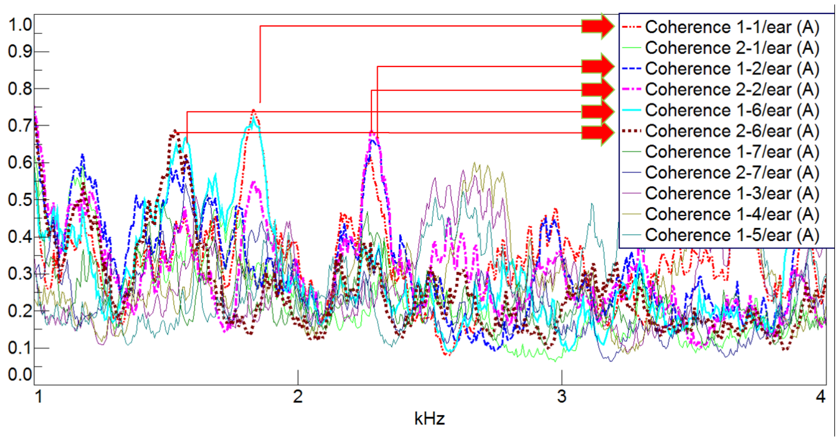
Once the relevant frequency ranges and sound properties are defined, we can move on to show the relation between the radiated sound and the vibration on different machine components. For convenience, we removed the outer covers of the coffee machine and defined the measurement positions as in fig. 3.2.

At the top of the machine, there are 4 accelerometers as shown in fig 3.2(d). 1-1, 2-1 are symmetrically placed at the connections of the top plate and the sliders. 1-6 is on the connection of the top plate and the side chassis. 1-7 is on the middle of the top heating chamber.

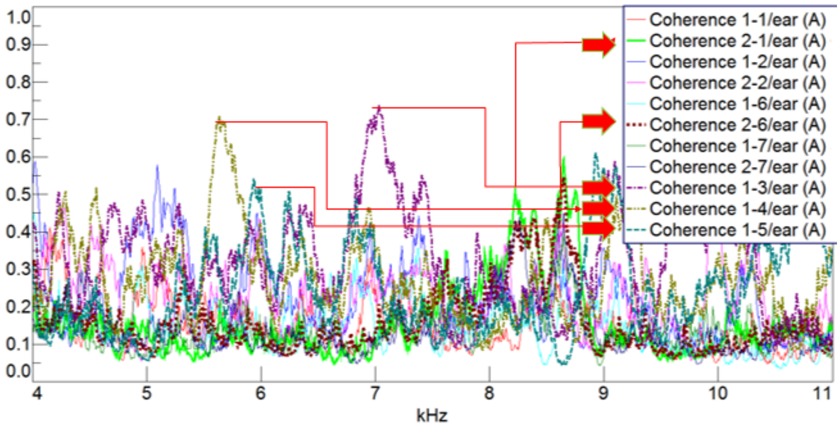
Figure 3.2(a)and fig. 3.2(b) shows the measurement positions on the left side. 1-2 is on the middle of the left chassis with 10 cm distance to the table; 1-3, 1-4, 1-5 are on the support frame of the pump along different directions.

2-2 is on the middle of the right chassis with 10 cm distance to the table as shown in fig. 3.2(c); 2-6 and 2-7 are on the connection of the top plate and the side chassis along different directions.

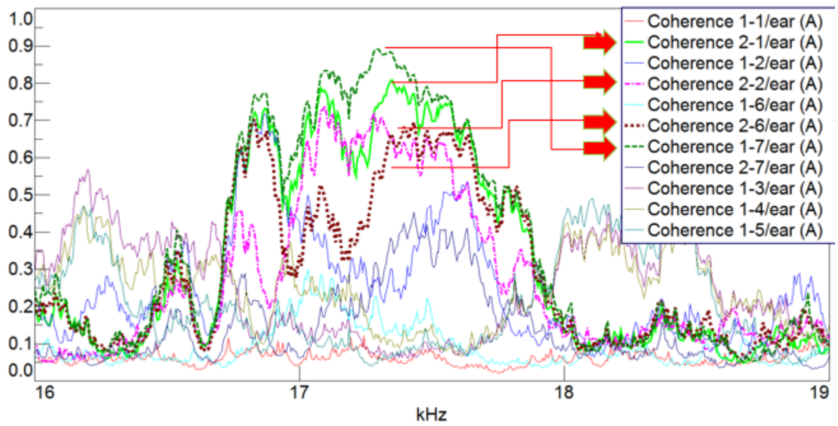
The coherence between the velocities in different measurement positions and the recorded sound power level are shown in fig. 3.3(a) to fig. 3.3(c).



(a) 1-3 kHZ.



(b) 4-10 kHz.



(c) 15.8-17.8 kHz.

Figure 3.3: Coherence of vibration and recorded sound power level.

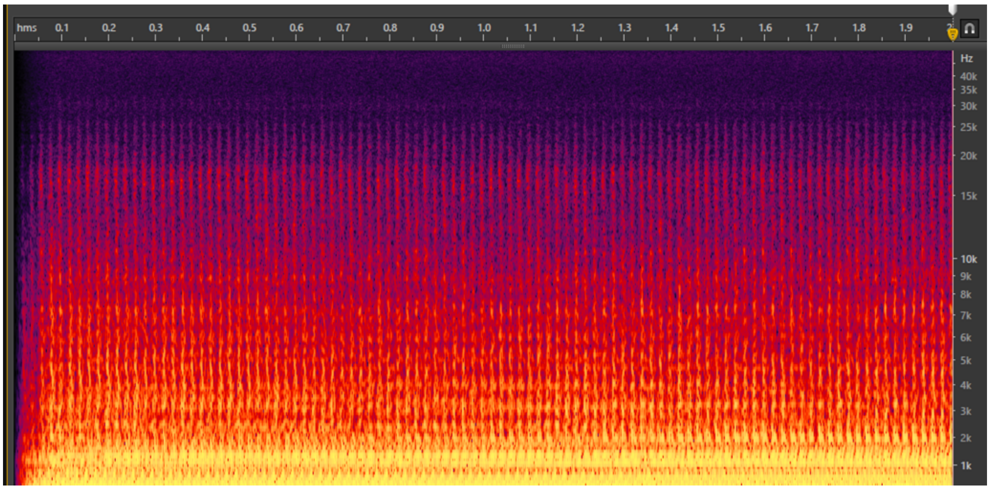


Figure 3.4: Pump rattling in time-frequency domain.

From fig. 3.3, we see that at low frequencies, the side chassis contributes most to the radiated sound, and the sound radiated from the top plate is the second greatest. The support frame of the pump shows higher correlation with increasing frequency. At the highest interesting frequency internal, the whole structure is dominating, especially the top plate and the side chassis.

The rattling pattern introduced by the impact excitation of the pump is shown in fig. 3.4.

In summary, the highest correlation of vibration at different positions and the recorded sound power level corresponding to three frequency ranges, and the structural problem which introduces the rattling sound are shown in table 3.2.

Table 3.2: Physical problem identification.

Frequency range [kHz]	highest correlated position
1-3	side chassis; top plate;
4-10	pump frame; top plate;
15.8-17.8	top plate; side chassis;
rattling	pump

In the discussion above, four problems were identified after analyses. For further studies of different solutions, the coffee machine was disassembled even more to configure the main vibration source and quantify how the source affects different components.

Modifying the generator

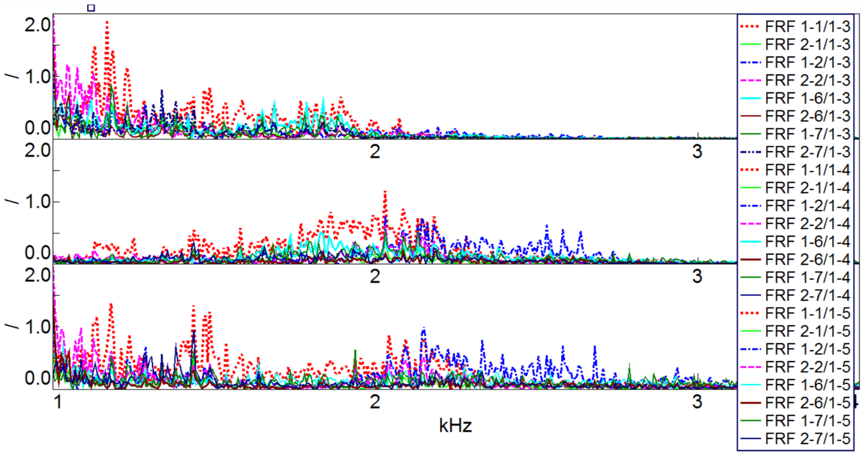
To reduce the rattling pattern discussed above and the radiated sound, the pump is the first thing taken into consideration. However, the pump is compactly installed in this structure, which results in difficulties for further analysis and treatment. Thus, only a potential target sound without rattling was created but no actual implementation was taken on the coffee machine to achieve the designed sound.

Meanwhile, the radiated sound can be suppressed by attaching extra mass at excitation as discussed in 2.2.2. In this case, extra mass can be attached to the pump supporting frame.

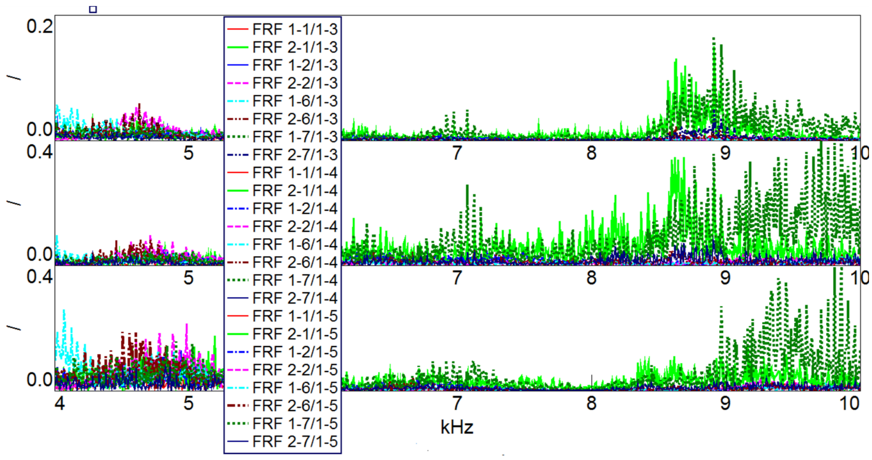
Modifying the transmission paths

Before one could choose the proper methods to reduce the energy transferred trough different paths, it is required to get the frequency response functions between the pump and the radiating surfaces. Hence, a new measurement was set up for this purpose to obtain the FRF of different surfaces with reference to the pump.

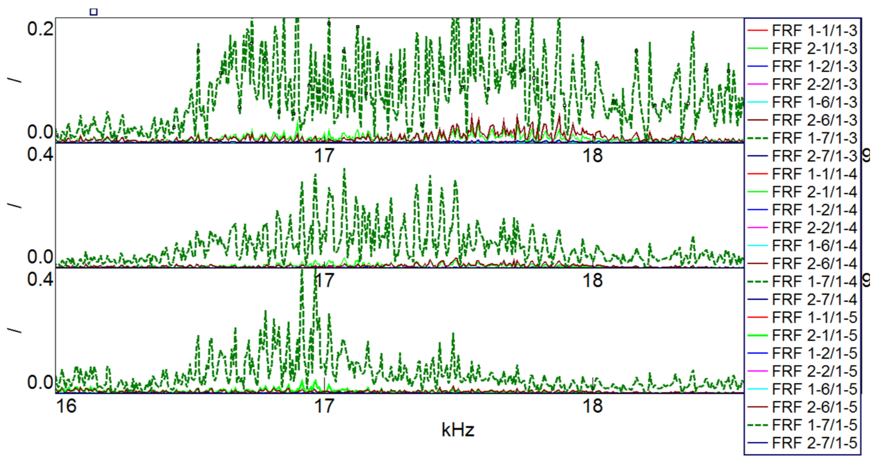
The side chassis and the pump supporting frame are critical for the energy transferred from the pump. Thus, damping materials can be used on the contact surfaces to smooth the peaks caused by impact excitation. One thing should be noticed is that the damping materials are relatively heavy compared to the plastic components of the coffee machine, hence, this method also add some extra mass at the excitation point. Some mass block of higher impedance compared to ABS plastic can be attached on the side chassis as blocking mass for a further reduction, especially at low frequencies.



(a) 1-3 kHz



(b) 4-10 kHz



(c) 15.8-17.8 kHz

Figure 3.5: FRF of velocities at different positions relative to that on the pump.

Modifying the radiating surfaces

The coupling between the metal handle and the plastic sliders, and the coupling between plastic sliders and side chassis are dominant to the sound radiated from the top plate. To suppress the high frequency components, absorbers provide a good reduction and allow easy application as well. Meanwhile, the metal handle can be changed to other materials, or the sliders on both sides can be attached by blocking mass to reduce the vibration.

From the analyses above, the following conclusions can be drawn:

Table 3.3: Conclusion of machine component identification and possible methods.

Character	Machine component	Method
Sound level	pump	extra mass
High frequency	side chassis; pump frame; top plate;	absorber; blocking mass;
Rattling	pump; side chassis;	delimited; damping;

3.2 Target sound design

Based on the results of section 3.1, the potential target sounds were designed.

Sound sketches were created in the sound design studio of A2Zound, Semcon. The equipment used for sound design are listed in table 3.4. The incoming audio was synthesized using Adobe Audition and Samplitude Pro X. These two combine with a library of sound effects and allow different custom designed filters that could be used for sound sketching.

The designs were mostly based on recorded sound of the real coffee machine and reasonable assumptions of the frequency characteristics of the sound treatment methods.

3.2.1 Sound sketching

Different sound sketches were created for different purposes of improvement. In the first design part, 8 sound sketches were created according to the methods listed in table 3.5. And 8 possible treatments were applied on the coffee machine to get a comparison between the designed sound and the recorded sound with relevant treatment. The materials used for structural treatments are shown in fig. 3.6 to fig. 3.8. The absorbers were chosen

Table 3.4: The equipment used for sound design.

No.	Name	Description
1	TASCAM FW1884	Digital Mixing Console
2	RME	A/D - D/A Converter ADI-2
3	Behringer Miniamp AMP800	4-Channel Stereo Headphone Amplifier
4	Sennheiser HD650	Headphones
5	Adobe Audition 6	sound design software
6	Samplitude Pro X	sound design software
7	Think pad X200	computer

based on the thickness due to the limitation of space between the outer cover and the inside structure. The damping layers were chosen according to the thickness as well as the Young's modulus due to the limitation of space between the pump and the structure, and the Young's modulus of ABS plastic. The mass was chosen based on the thickness and density due to the limitation of space between the pump and the outer cover, and the weight of pump frame. The choice of each parameter will be explained further in each method.

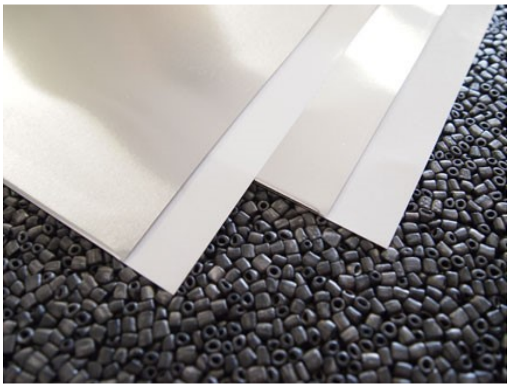
Table 3.5: The methods used during the sound sketching.

No.	Name	Description
1	Method 1	reduction of high frequency components using absorber 1
2	Method 2	reduction of high frequency components using absorber 2
3	Method 3	reduction of rattling using damping layer 1
4	Method 4	reduction of rattling using damping layer 2
5	Method 5	reduction of sound power level using extra mass 160 g
6	Method 6	reduction of sound power level using extra mass 120 g
7	Method 7	reduction of high frequency components using blocking mass and absorber 1
8	Method 8	reduction of rattling using a quiet pump

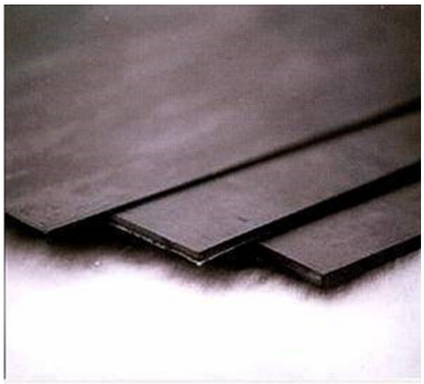
(a) absorber 1

(b) absorber 2

Figure 3.6: Absorber with thickness of 2 mm (a) and absorber with thickness of 10 mm (b).



(a) damping 1



(b) damping 2

Figure 3.7: Rubber with aluminum sheet, thickness of 1.2 mm (a) and rubber with thickness of 1.2 mm (b).

Figure 3.8: Attached extra mass.

Reduce high frequency components

The first attempt was to reduce high frequency components and lower the sound level in general. Three filters were designed for the application in Samplitude pro X to create potential target sounds that simulate a reduction at high frequencies.

The largest dimension in the structure of the coffee machine is the side chassis, which is approximately 30 cm. Based on the knowledge of sound radiation from a rigid baffled finite plate, we can derive a conclusion that above 1.1 kHz, the side chassis is almost the same dimension of the wavelength, which means the side chassis vibrates as a separate baffled plate instead as a part of a whole structure as below 1.1 kHz. Thus, the transition frequency should be not larger than 1.1 kHz to provide a better reduction on the side chassis. Meanwhile, the interesting frequency range starts at 1 kHz as discussed in section 3.1.2. Hence, the transition frequency was designed to be 1 kHz.

The cut-off frequency should be less than 15.8 kHz, which is the lower frequency of the third frequency range of interest, to guarantee a reduction at frequencies above 15.8 kHz. As 15.8 kHz is relatively a high frequency for the users of a coffee machine, which might not be perceived for many users. Consequently, 15 kHz was one of the choice for the cut-off frequency. The largest distance between the side chassis and the outer cover is around 3 cm, hence, another choice for the cut-off frequency can be 12 kHz according to the property of absorbers that could be used in this 3 cm distance.

Besides, the designed sound is required to be of lower power in general. The values of reduction were decided according to the properties of the absorbers as the biggest limitation was the space inside the coffee machine, hence, the thickness decided the type of absorber that could be used and therefore the reduction. Comsequently, the filters were designed to be with 9 dB reduction and 3.5 dB reduction at frequencies below the transition frequency respectively. At cut-off frequency, the filters were designed to be with 12.5 dB reduction and 8.5 dB reduction respectively.

The frequency responses of the filters are shown in fig. 3.9 and fig. 3.10.

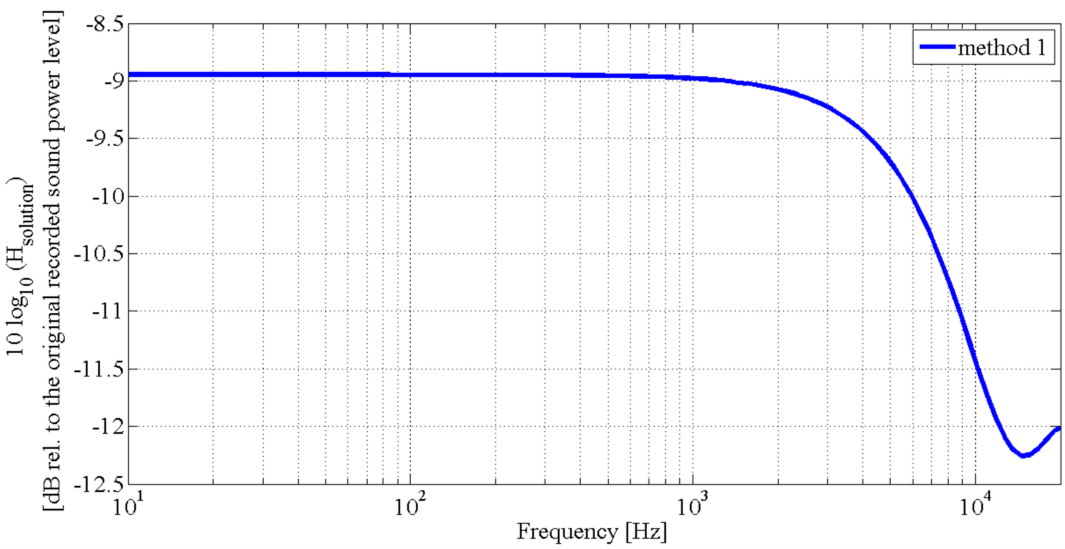


Figure 3.9: Frequency response of the filter for reduction of high frequency components using method 1.

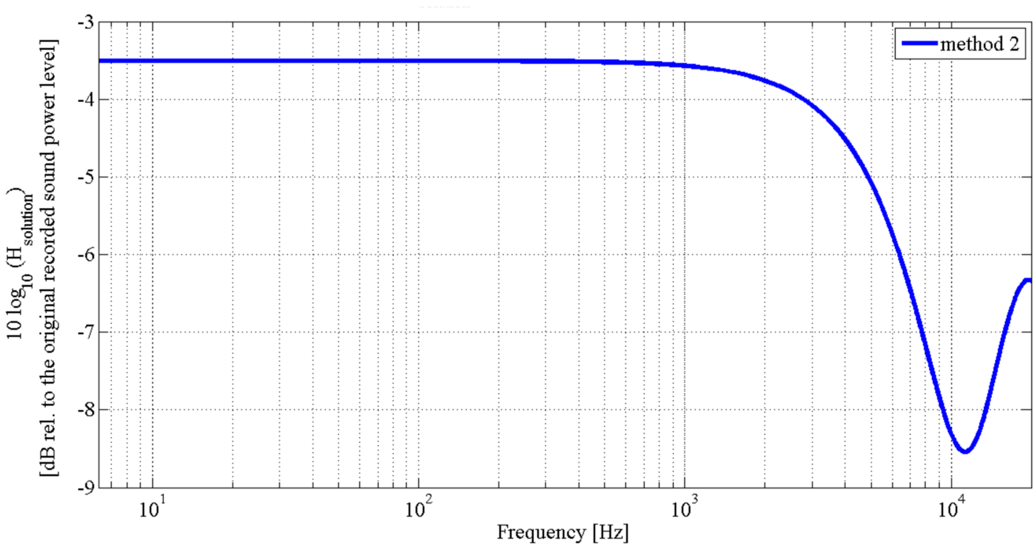


Figure 3.10: Frequency response of the filter for reduction of high frequency components using method 2.

Applying the designed filters on the recorded original sound, two potential target sounds were created. Two kinds of real absorbers were applied on the coffee machine to get a comparison between the designed sounds and the recorded sounds with absorbers.

The comparison of the A-weighted sound power level among the original recorded sound, the designed sound and the recorded sound after using method 1 and 2 are shown in fig. 3.11 and fig. 3.12. In the interesting frequency ranges, the designed filters have a higher reduction than that of the chosen absorbers, as the purpose of the designed sound is to provide a product sound with less high frequency components so that the jury group members can judge whether they prefer high frequencies, hence, it is applicable to obtain the potential target sound by using the designed filter instead of making changes on the real coffee machinethe highest.

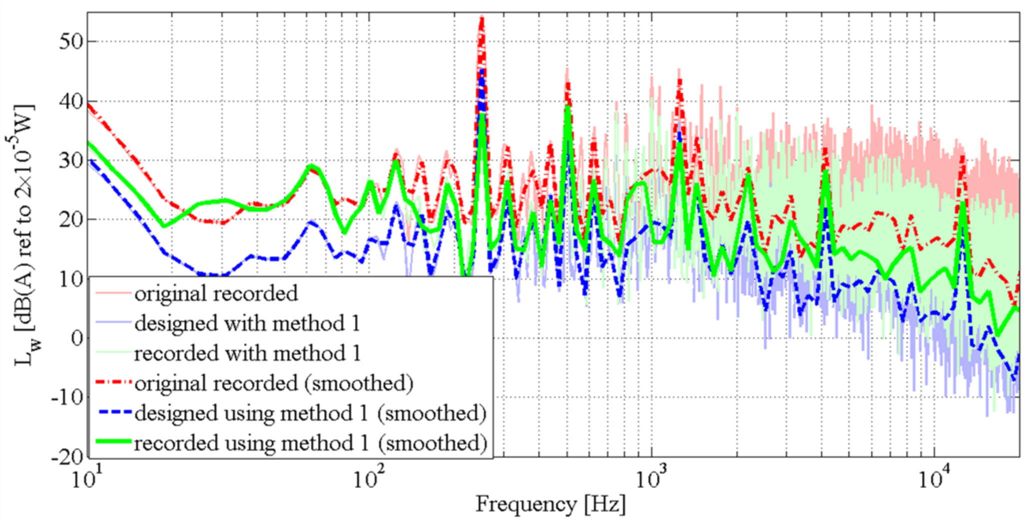


Figure 3.11: A-weighted sound power level among the original recorded sound, the designed sound and the recorded sound after using method 1, the smoothed curve is plotted in the bands whose center frequency, lower frequency, and upper frequency have the relation of $f_u/f_c = f_c/f_l = 2^{1/10}$.

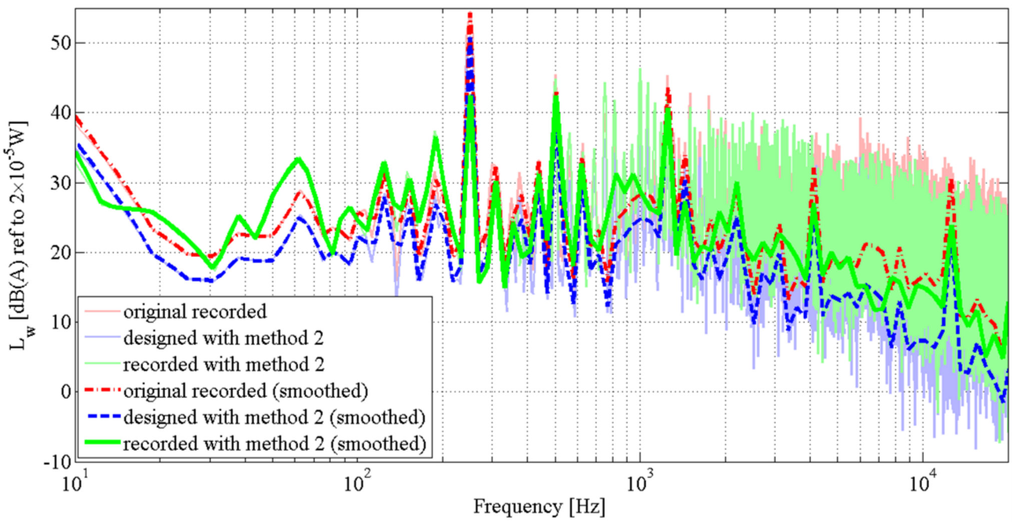


Figure 3.12: A-weighted sound power level among the original recorded sound, the designed sound and the recorded sound after using method 2, the smoothed curve is plotted in the bands whose center frequency, lower frequency, and upper frequency have the relation of $f_u/f_c = f_c/f_l = 2^{1/10}$.

For the sound radiated from the top plate, another filter which simulates the sound treatment of blocking mass on both sliders, and absorber 1 on the top plate to suppress high frequency components was designed.

As shown in fig. 3.9, absorber 1 allows more than 9 dB reduction above 3 kHz, and the most interesting frequency range starts at 1 kHz. Hence, this filter simulating blocking mass on the sliders was designed between 1 and 3 kHz.

Based on section 3.1, the harmonic frequency is 50 Hz, thus, the filter of blocking mass was designed to be notched each 50 Hz with higher reduction. The density of ABS plastic is 1100 kg/m^3, the density of zinc block is 7134 kg/m^3, the thickness for both are approximately 4mm, the length of zinc block is 2 cm, thus, the value of reduction can be calculated according to eq. (2.21).

The frequency response of the designed filter is shown in fig. 3.13:

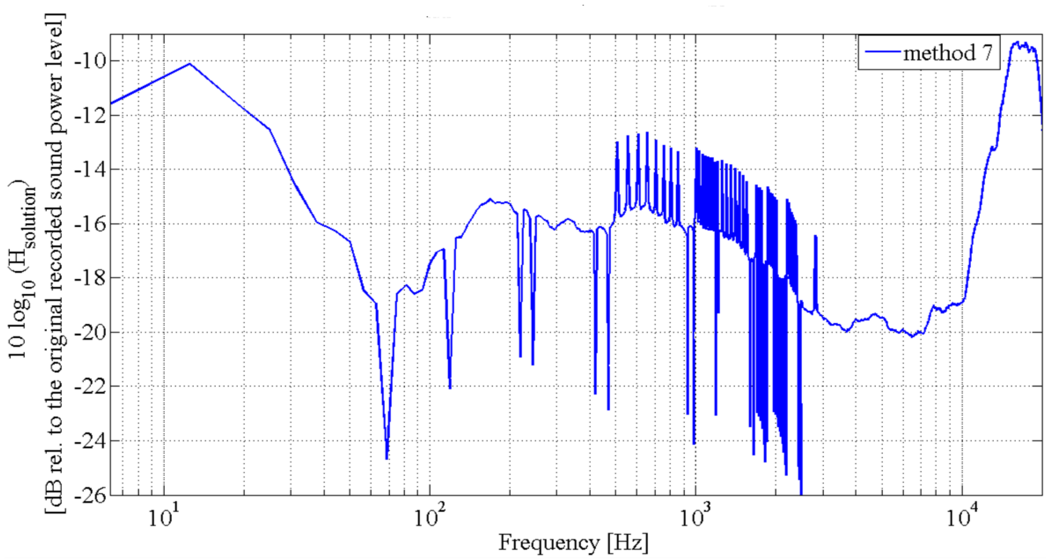


Figure 3.13: Frequency response of the filter for reduction of high frequency components using method 7.

A potential target sound was created using the designed filter. A comparison between the designed sound and the real recorded sound using method 7 was also done by attaching 20 g zinc block on either slider, and absorber 1 to the metal surfaces. The comparison of the A-weighted sound power level among the recorded original sound, the designed sound and the recorded sound after applying method 7 are illustrated in fig. 3.14. The designed sound has higher reduction than that of the recorded sound with treatment when the frequency gets higher.

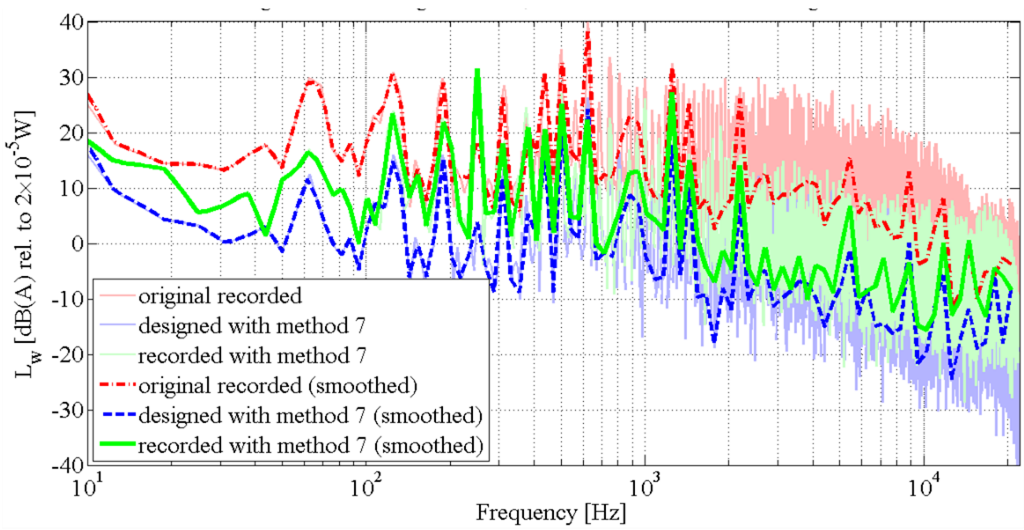


Figure 3.14: A-weighted sound power level among the original recorded sound, the designed sound and the recorded sound after using method 7, the smoothed curve is plotted in the bands whose center frequency, lower frequency, and upper frequency have the relation of $f_u/f_c = f_c/f_l = 2^{1/10}$.

Attenuation of pump rattling

The potential target sound without rattling was created using the software Adobe Audition by damping the energy at peaks to get a smoother energy distribution as shown in fig. 3.15.

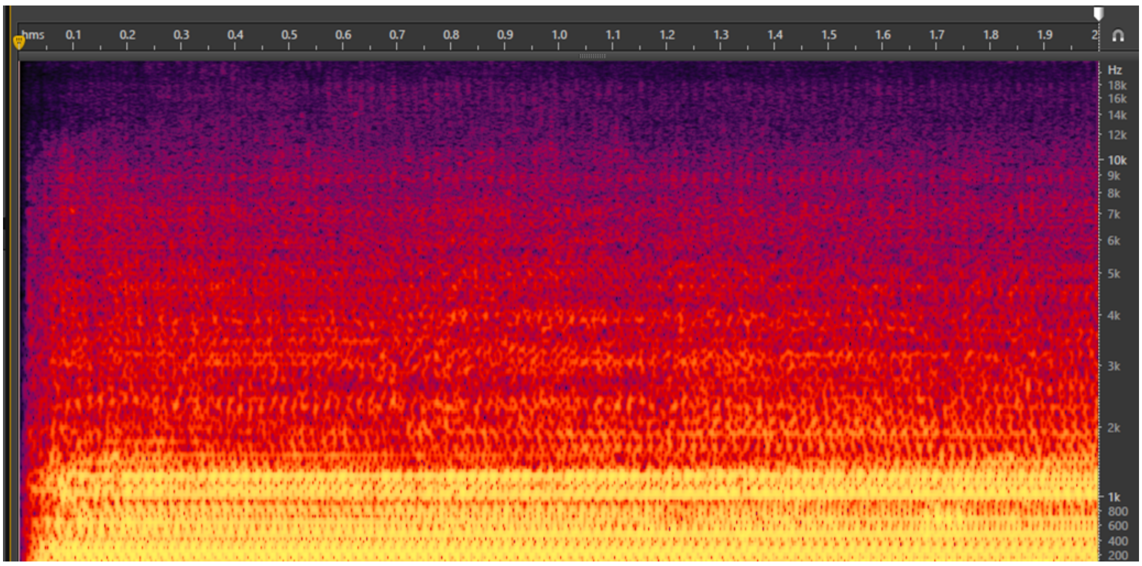


Figure 3.15: A-weighted sound power level of the sound in time-frequency domain after attenuating the pump rattling using method 8.

However, the rattling is the result of the impact excitation from the pump, which delimitates the modification and improvement on the real machine. Only a designed sound sketch was created but no treatment on the coffee machine could be introduced to achieve the potential target sound.

Damping layers achieve a general reduction in energy especially at peaks caused by pump impact.

Similar to the methodology used in 3.2.1, two filters with notches each 50 Hz were created. From fig. 3.1, one can see that the outer cover acts as a low pass filter with cut-off frequency of about 6 kHz, thus, the filters which simulate the treatment of damping layers were designed at frequencies below 6 kHz. And the interesting frequency range starts around 1 kHz, hence the filters were designed at frequencies above 1 kHz.

The thickness of the side chassis is about 1 mm, the thickness of the damping layer is 1.2 mm, the Young's modulus of ABS plastic is 3.1×10^9 N/m^2, the Young's modulus of the damping layer 1 and 2 are 0.3×10^9 N/m^2 and 0.1×10^9 N/m^2 respectively. Hence, the value of reduction can be designed using eq. (2.26). The damping layers are relatively heavy compared to the plastic components of the coffee machine, consequently, the damping layers can be taken as an extra mass attached on the transmission path, which cause less reduction at lower frequencies. When simulating a heavier damping material with lower Young's modulus, the reduction at frequencies above 1 kHz was lower, but the attached weight was heavier, therefore, the components at certain frequencies below 1 kHz were assumed to be larger than original sound. The density of damping layers are 0.93 kg/m^3 and 0.91 kg/m^3 respectively, both material are 1.2 mm thick, and the thickness of left chassis is about 1 mm, hence, values of reduction were derived after calculating the radiated power using eq. (2.50).

Two different damping layers were designed with frequency responses as shown in fig. 3.16 and fig. 3.17. Fig. 3.16 is of higher reduction in the important frequency range, fig. 3.17 is of lower reduction.

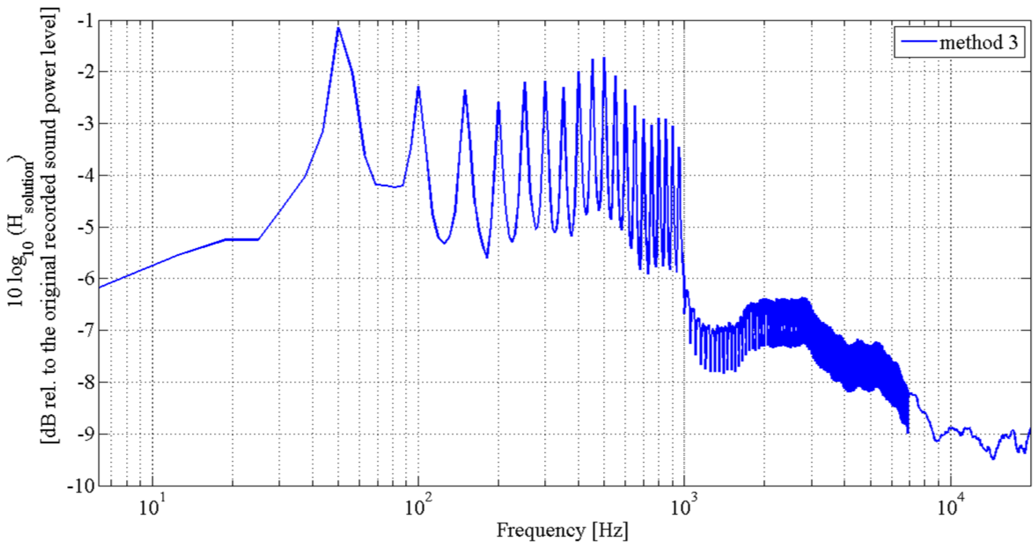


Figure 3.16: Frequency response of the filter for reduction of rattling using method 3.

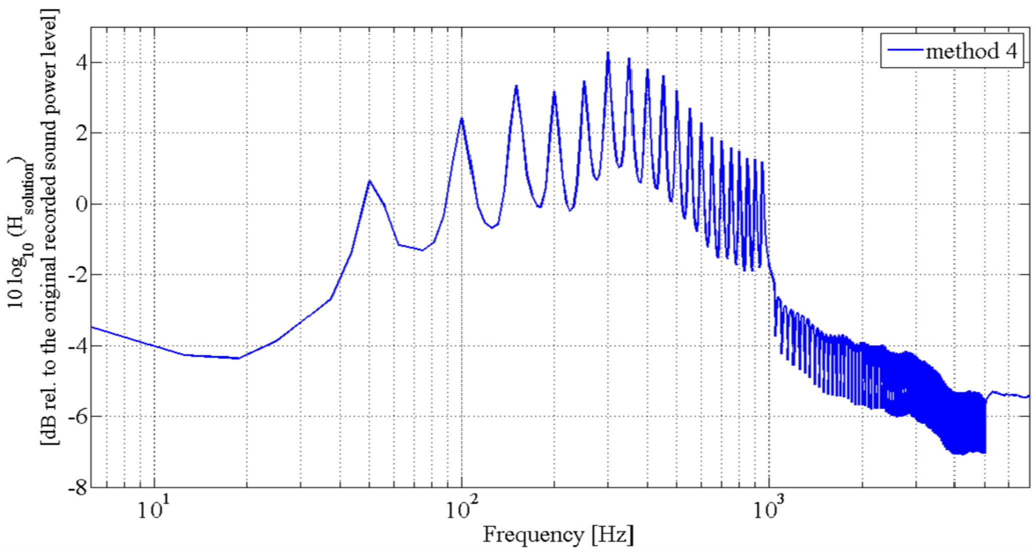


Figure 3.17: Frequency response of the filter for reduction of rattling using method 4.

Two potential target sounds were created using the designed filters based on the recorded original sound. And two kinds of real damping layers were applied on the coffee machine to get the comparison between the designed sounds and the recorded sounds using the treatment of damping layer.

In the important frequency ranges, the designed sounds have higher reduction than that of the recorded sound using damping layers, especially at peaks. At low frequencies, the designed filters have less reduction than the measured results. The comparison of A-weighted sound power level among the recorded original sound, the designed sound and the recorded sound using method 3 and 4 are shown in fig. 3.18 and fig. 3.19.

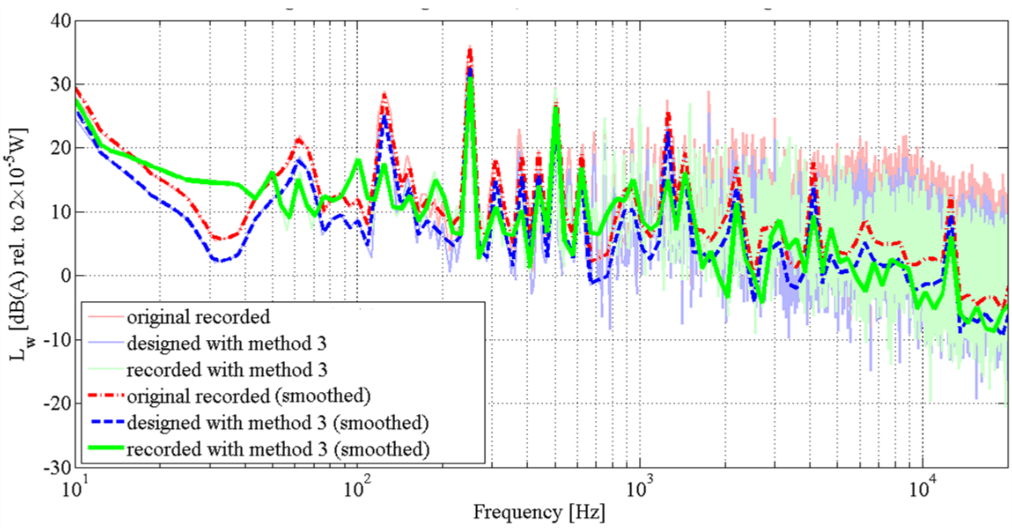


Figure 3.18: A-weighted sound power level among the original recorded sound, the designed sound and the recorded sound after using method 3, the smoothed curve is plotted in the bands whose center frequency, lower frequency, and upper frequency have the relation of $f_u/f_c = f_c/f_l = 2^{1/10}$.

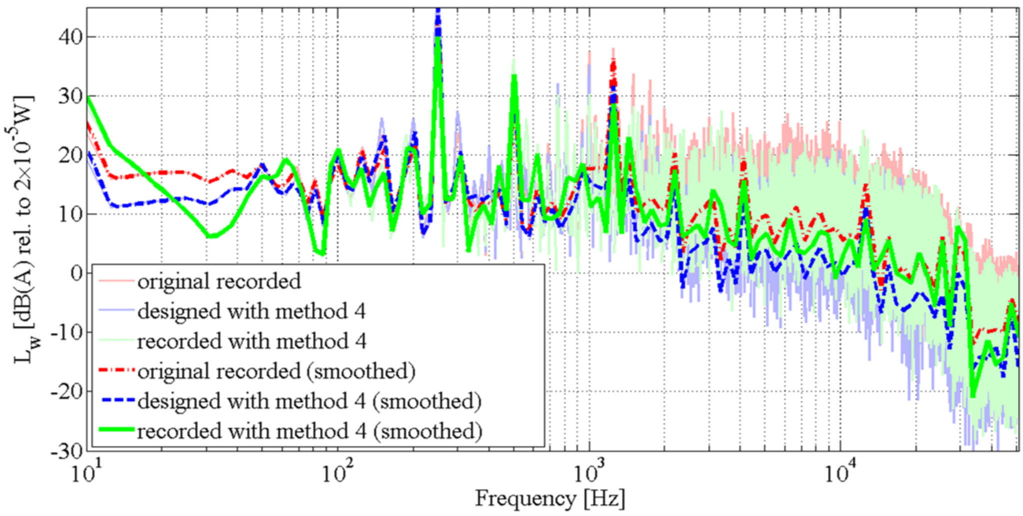


Figure 3.19: A-weighted sound power level among the original recorded sound, the designed sound and the recorded sound after using method 4, the smoothed curve is plotted in the bands whose center frequency, lower frequency, and upper frequency have the relation of $f_u/f_c = f_c/f_l = 2^{1/10}$.

Reduce sound level

The radiated sound power can be suppressed by increasing the mass per unit area at the excitation point as discussed in section 2.2.2. Filters that simulate the treatments of extra mass were designed for this purpose. The pump frame dominates in the frequency range 4 to 10 kHz, which is consequently the frequency range of the filters. As mentioned in section 3.2.1, it is only meaningful to design the effect of the attached mass up to 6 kHz. The density of the zinc block is 7134 kg/m^3, and the thickness is 4 mm, hence, the value of reduction can be derived by calculating the radiated power using eq. (2.50). Attached mass suppresses energy at certain frequencies, especially at impact peaks, but increase energy at lower frequencies.

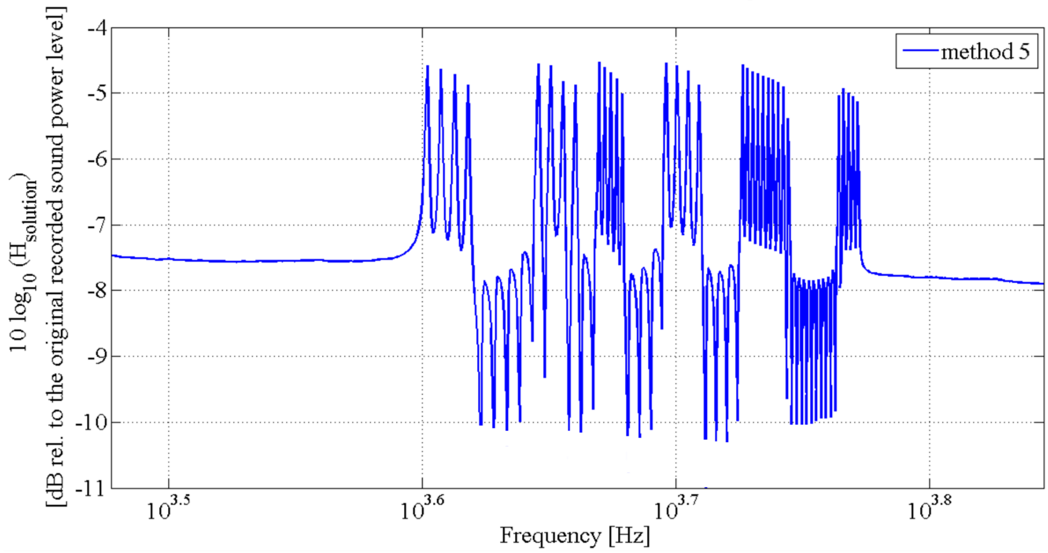


Figure 3.20: Frequency response of the filter for reduction of sound power level using method 5.

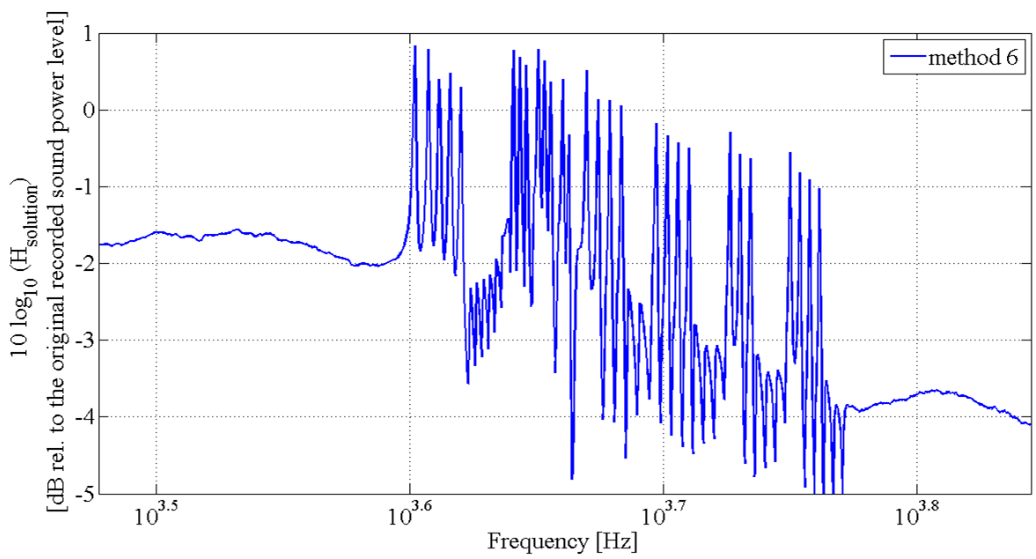


Figure 3.21: Frequency response of the filter for reduction of sound power level using method 6.

Another two potential target sounds were created using the designed filters based on the recorded original sound. Two different mass blocks were applied on the pump supporting frame to get a comparison between the potential target sounds and real recorded sounds after adding attached mass.

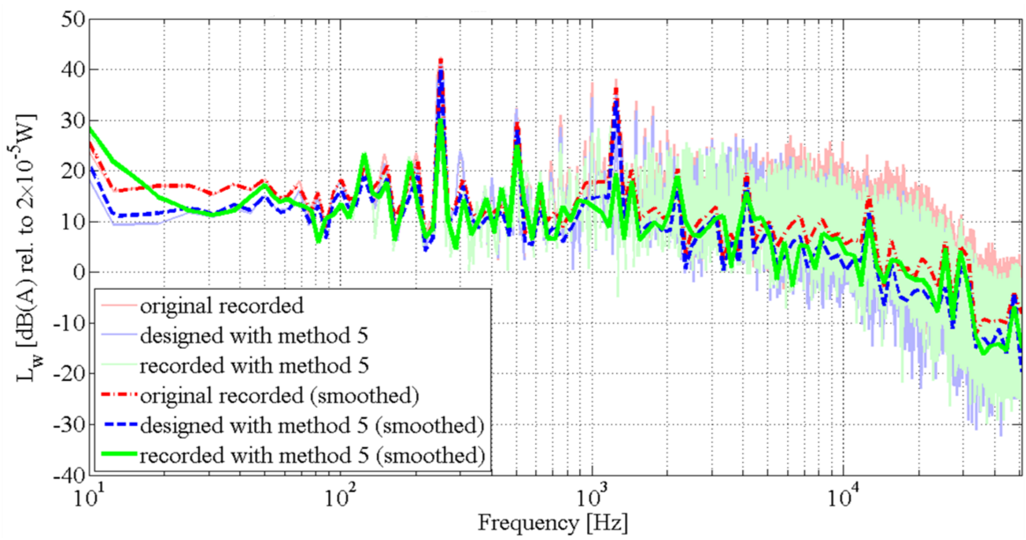


Figure 3.22: A-weighted sound power level among the original recorded sound, the designed sound and the recorded sound after using method 5, the smoothed curve is plotted in the bands whose center frequency, lower frequency, and upper frequency have the relation of $f_u/f_c = f_c/f_l = 2^{1/10}$.

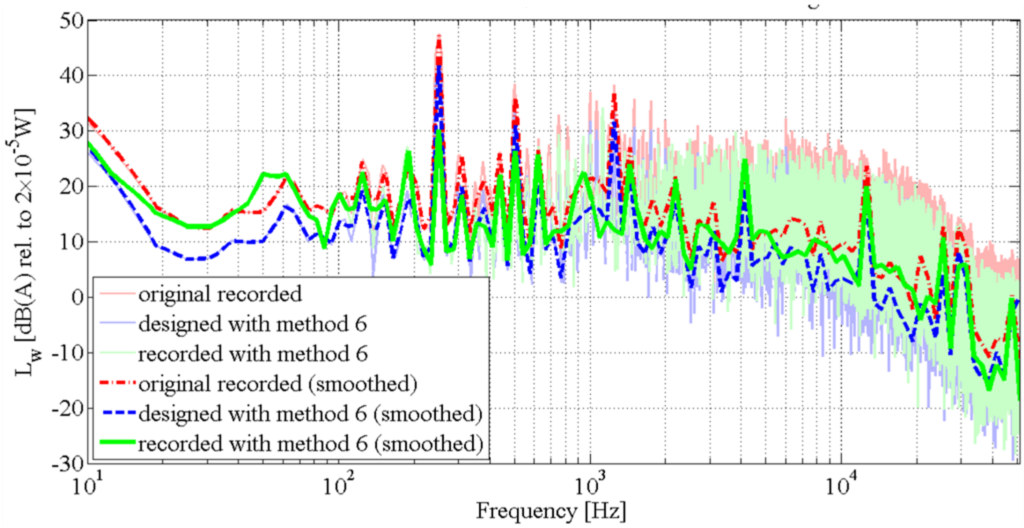


Figure 3.23: A-weighted sound power level among the original recorded sound, the designed sound and the recorded sound after using method 6, the smoothed curve is plotted in the bands whose center frequency, lower frequency, and upper frequency have the relation of $f_u/f_c = f_c/f_l = 2^{1/10}$.

The comparison of A-weighted sound power level among the recorded original sound, the designed sound and the recorded sound after using method 5 and 6 are shown in fig. 3.22 and fig. 3.23. The designed filters fit well with the measurement results. The designed sounds have higher reduction than the recorded sounds after applying attached mass at high frequencies.

3.3 First jury group evaluation

The first evaluation was performed on the 15th of September, 2014 and took around an hour. There were seven participants in the evaluation, 5 of whom were employees of Semcon, including one from A2Zound, and the remaining two were from Department of Applied Acoustics, Chalmers. The average age of the participants was 33 years old. None of the participants reported hearing impairment.

The basic information of the participants is summarized in fig. 3.24. The activity for sound listening and music listening helps to understand their sensitivity of sound; the frequency of drink coffee helps to know whether they use coffee machine frequently, which means the times they are exposed to the sound of coffee machine.

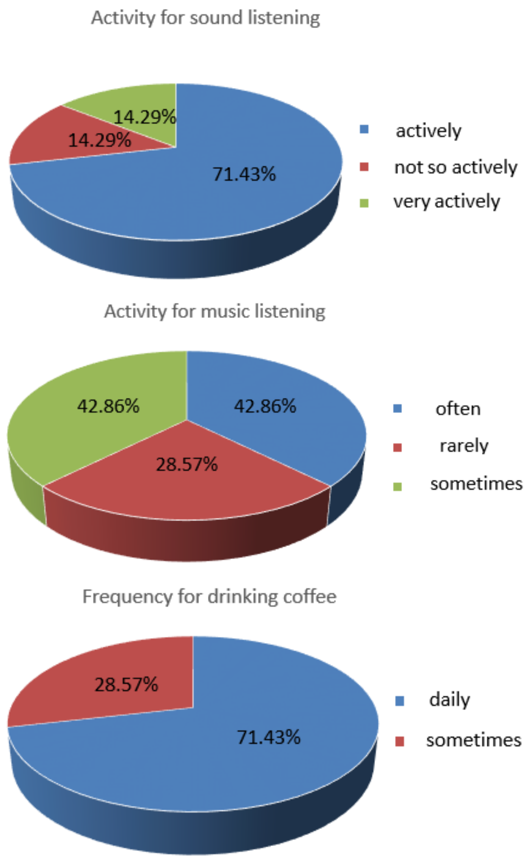


Figure 3.24: Basic information of the participants in the first jury group

Before the evaluation, the loudness level of the designed sounds presented to the participants from the loudspeakers was calibrated to match the loudness when operating machine at ear height. The set up can be found in Appendix A. The equipments used during the first jury group evaluation is listed in table. 3.6:

Table 3.6: The equipments used during jury group evaluation.

No.	Name	Description
1	Dell M4600	Computer
2	Arta	Calibration software
3	B&K calibrator	Mic calibrator
4	Superlux ECM 999	Microphone
5	KRK ROKIT 6	loudspeaker
6	Profire 610	sound card

This evaluation was divided into two parts:

Introduction

The first part consisted of a short presentation and introduction to the jury group evaluation and the questionnaire. The goal of this thesis was briefly stated and the different parts of the questionnaire were introduced with examples to give the listeners some background knowledge for answering the questionnaire.

Questionnaire and implementation

Six different sound sketches were presented (five potential target sounds and the original recorded sound of the coffee machine). Each sound sketch was 4 seconds, and was repeated as many times as necessary. Participants were not allowed to discuss these questions between each other in the first section. Two SAM (Self-Assessment Manikin) scales helped the participants represent their feelings when listening to the sound.

The first scale represents pleasantness, ranging from unpleasant over neutral, to pleasant. The second scale represents activation, from low activation, relaxed/tired to high activation, awake, active. The participants were asked to answer 3 questions per sound sketch in the first part. The first question requires the jury group members to scale how annoying they felt when hearing the sound. 1 means not annoying while 5 means very annoying. The second question requires the participants to judge whether the sound is an appropriate sound for a coffee machine. The scale is from 1, very inappropriate, to 5, very appropriate. The third question requires the participants to rate the quality of the coffee machine based on the sound.1 stands for very low quality, and 5 stands for very high quality.

Next, the six sound sketches were played again in one sequence and the participants were required to review their questionnaires and select one favorite and one least favorite sketches together with an explanation for their selection.

The last part was a group discussion based on the questionnaire and the selection done before. All participants discussed their favorites and the least preferred sound sketches.

Appendix B shows the format of the questionnaire.

3.3.1 Results of the first jury group evaluation

In this section, the results from the first jury group evaluation are presented in three subsections. The first subsection presents the results from the valence - activation scales of the designed sound sketches. The second subsection shows the result of the favorite and least favorite sound sketches and also discusses the reasons for the selection. The third subsection presents the users' expectations, for instance, what to keep and what to avoid.

Table 3.7 lists the sound sketches used in the first jury group evaluation.

Table 3.7: The sound sketches used during the first jury group evaluation.

No.	Name	Description
1	Sound 1	using method 1, 3 and 5
2	Sound 2	using method 1
3	Sound 3	using method 3
4	Sound 4	the original recorded sound
5	Sound 5	using method 8
6	Sound 6	using method 7

Emotional responses

The emotional responses were investigated by calculating the sample mean values of two scales from jury group members, one is from Calm to Alert, and the other scale is from Unpleasant to Pleasant. All the sound sketches were scaled using these two scales.

Figure 3.25 shows the Valence - Activation mapping of the potential target sounds. One can see that all designed sound sketches were rated as more pleasant compared to the original recorded sound. However the participants also stated that all six sound sketches were too loud, which lowered the general acceptance. Sound 3 and sound 5 had low activation scores due to the reduced rattling. Sound 3 reaches the highest pleasantness ranking as a result of lower pitch.

According to the questionnaires, sound 4 (the original recorded sound) is the most annoying sound and has the lowest quality ranking as shown in fig. 3.26 to fig. 3.28. Sound 3 and sound 5 are rated as the most appropriate sounds for a coffee machine of high quality for the reason that both are smoother than the other sounds. Sound 5 is even

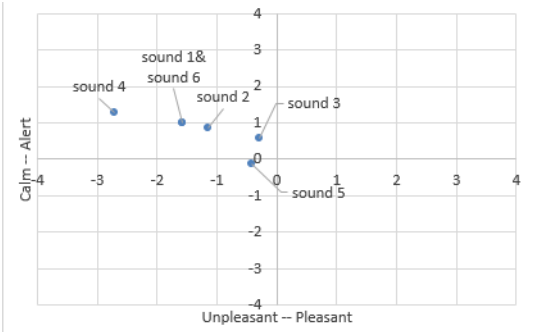


Figure 3.25: Emotional responses for potential target sounds.

smoother than sound 3 and is therefore considered to be of higher quality. Sound 1, sound 2 and sound 6 were perceived as similar according to the remarks from participants, but sound 1 was judged as the second most annoying sound as the redundant low frequency components gave the participants the impression of “broken machine” and “strong sound attack”.

Selection of the favorite and least preferred sounds

Sound 3, the sound designed from applying damping layer 1, was selected to be the favorite sound by three participants. After compared sound 3 with other sound sketches in pitch and rattling, the main reasons were figured out by the jury group members as below:

a) low pitch

b) smoother rattling components

Sound 4, the original recorded sound, was voted to be the least favorite sketch due to the following reasons obtained from the comparison with other sound sketches in frequency components, sharpness, rattling, and fluctuation:

a) the high frequency components lead to an annoying sharpness;

b) less low frequency components remind people of poor quality;

c) the clear rattling pattern;

d) the fluctuation in tone makes the machine sounds unstable.

The comments for each sound sketch and the reasons why participants liked or disliked each sound were discussed during the group discussion. Most of the comments are summarized in table 3.8:

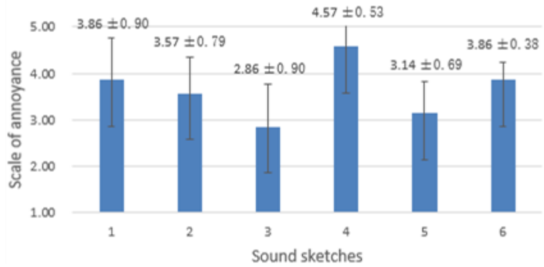


Figure 3.26: Mean value and standard deviation of scale of annoyance for potential target sounds.

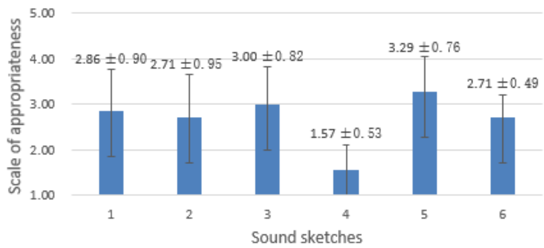


Figure 3.27: Mean value and standard deviation of scale of appropriateness for potential target sounds.

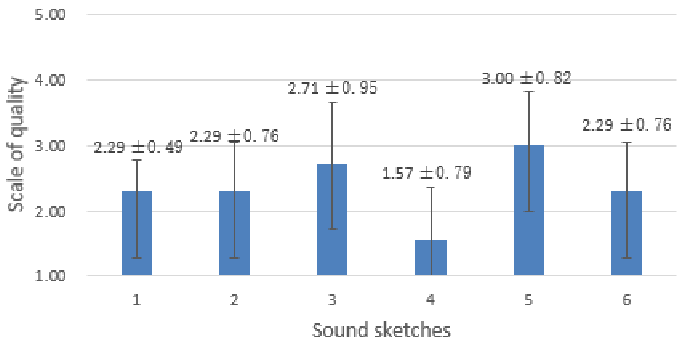


Figure 3.28: Mean value and standard deviation of scale of product quality for potential target sounds.

Table 3.8: Summary of comments for each potential target sound.

Sound no.	Likes	Dislikes
Sound 1	N/A	too strong low frequency component; rattling pattern;
Sound 2	N/A	too strong low frequency component; rattling pattern;
Sound 3	low pitch; smoother rattling;	N/A
Sound 4	sounds like a coffee machine;	high frequency component; fluctuation; rattling pattern;
Sound 5	bright and happy tone;	too smooth to be a coffee machine; fluctuation, not constant in tone;
Sound 6	N/A	high pitch; less low frequency components; rattling pattern;

Discussion on expectation and avoidance

During the final part of the jury group evaluation, what to expect and avoid in future target sounds were discussed. The comments are summarized in table 3.9.

Table 3.9: Summary of comments for future sound sketches.

Category	Expectation	Avoidance
Suggestion	smoother rattling; low pitch;	too strong low frequency component; rattling pattern; fluctuations; high pitch;

Apart from the listed comments in table 3.9, there was also a common suggestion to keep the rattling pattern to make it sounds like grinding of coffee beans during the early part while making the sound smoother or silent in the following seconds. This is hard to achieve only by mechanical modifications as the rattling pattern is produced by the pump impact as discussed in section 3.1, hence, replacing the pump is not within the scope of this thesis. However, it is a method of application which could be explored more in the future work.

3.3.2 The first improvement of sound sketches

Based on the results from the first jury group evaluation, three improved potential target sounds were created using the methods listed in table 3.10.

Table 3.10: The methods used during the first improvement of sound sketching.

No.	Name	Description
1	Method 9	reduction of high frequencies using absorber and blocking mass
2	Method 10	reduction of rattling using damping and attached mass
3	Method 11	reduction of rattling and high frequency components

Reduction of high frequency components

High frequency components were not preferred according to the first jury group evaluation. Consequently, different sound sketches with reduced high frequency components were designed to improve the sound quality. Low frequency components were expected for a coffee machine of higher quality, hence, the low frequency components were only marginally reduced compared to the high frequency components. The value of reduction was decided according to method 1 and method 7.

The frequency response of method 9 is shown in fig. 3.29 compared to that of method 1.

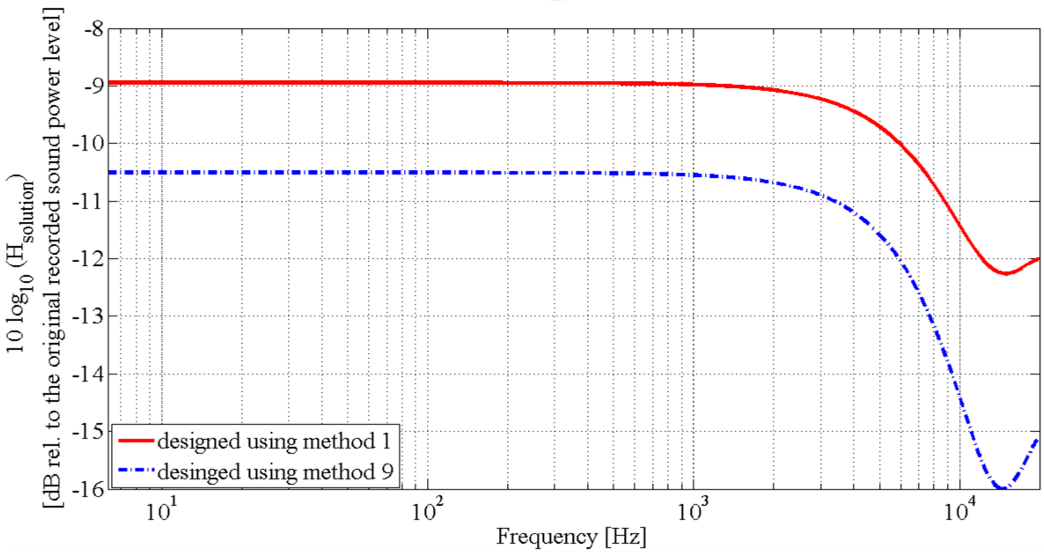


Figure 3.29: Frequency response of the filter for reduction of high frequency using method 9.

Using method 9, an improved sound sketch was created. The same absorbers and zinc block were implemented on the chassis and the top plate of the coffee machine as discussed in method 1 and method 7. The comparison among the frequency response of the designed filter of method 1, method 9 , and the measured filter using method 9 are displayed in fig. 3.32:

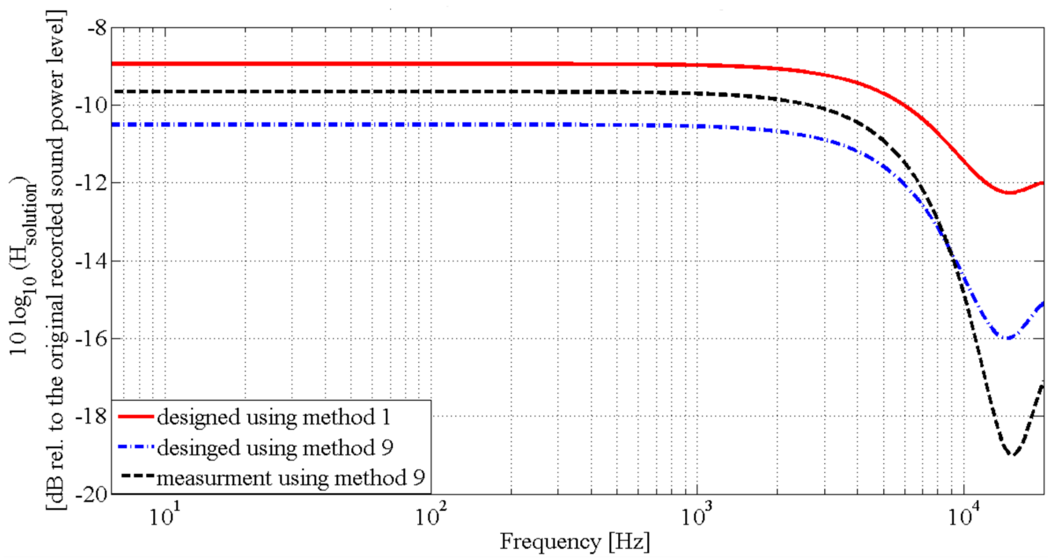


Figure 3.30: Comparison among the frequency response of the designed filters using method 1, method 9 and the measured filter using method 9.

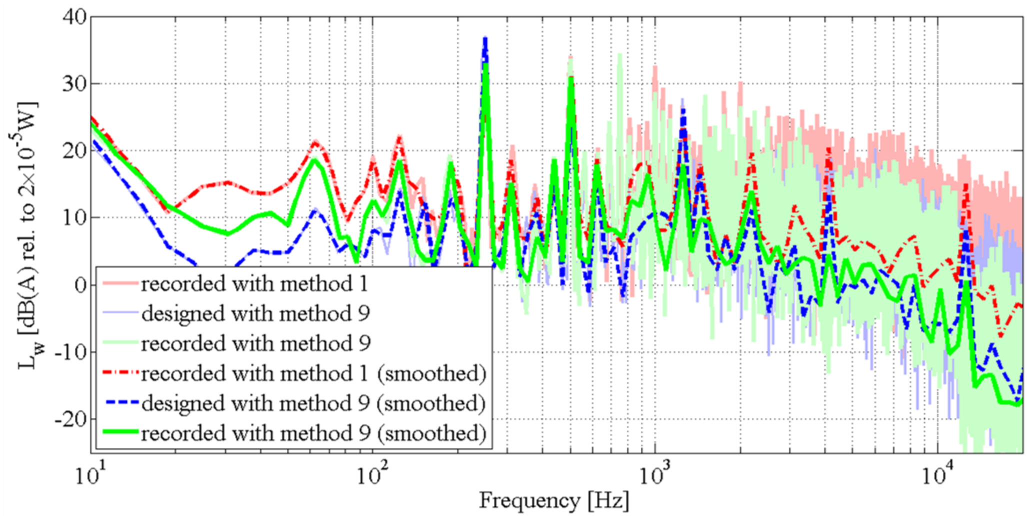


Figure 3.31: A-weighted sound power level among the recorded sound using method 1, the designed sound and the recorded sound after using method 9, the smoothed curve is plotted in the bands whose center frequency, lower frequency, and upper frequency have the relation of $f_u/f_c = f_c/f_l = 2^{1/10}$

Compared to the designed sound using method 9, the measured sound has a lower reduction at low frequencies while a higher reduction at high frequencies, but still lower than that of before improvement (method 1). In fig. 3.31, one can discover that the reduction gets higher with the increasing frequency, while at very low frequencies, the reduction of method 9 is almost the same with that of method 1.

Attenuating pump rattling

The rattling pattern was considered annoying, however, removing the rattling pattern did not lead to a favorite sound. The rattling pattern actually reminded the participants of a working coffee machine. So the potential target sound should be damped at impact peaks, but not totally smooth. And the total level should be lowered as well. For this purpose, a filter was designed using method 10 to simulate the treatment of damping material at excitation point and on the transmission paths, and also an extra mass attached at excitation point as method 5. The value of reduction was calculated in the same way of method 3.

The frequency response of method 10 is shown in fig. 3.32 compared to that of method 3.

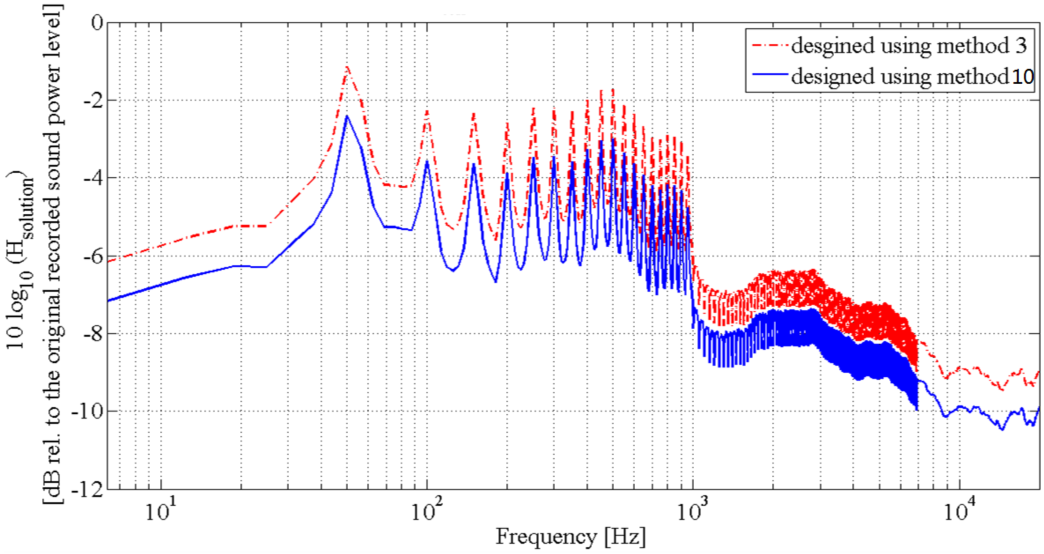


Figure 3.32: Frequency response of designed filter using method 10 to attenuate rattling.

Using method 10, an improved sound sketch was created. The same damping layers were implemented in the same places of the coffee machine as in method 3. Damping layers were also implemented on the contact surfaces between the sliders and the main body to attenuate the impact peaks. 160 g of extra mass were attached on the pump to get a further reduction.

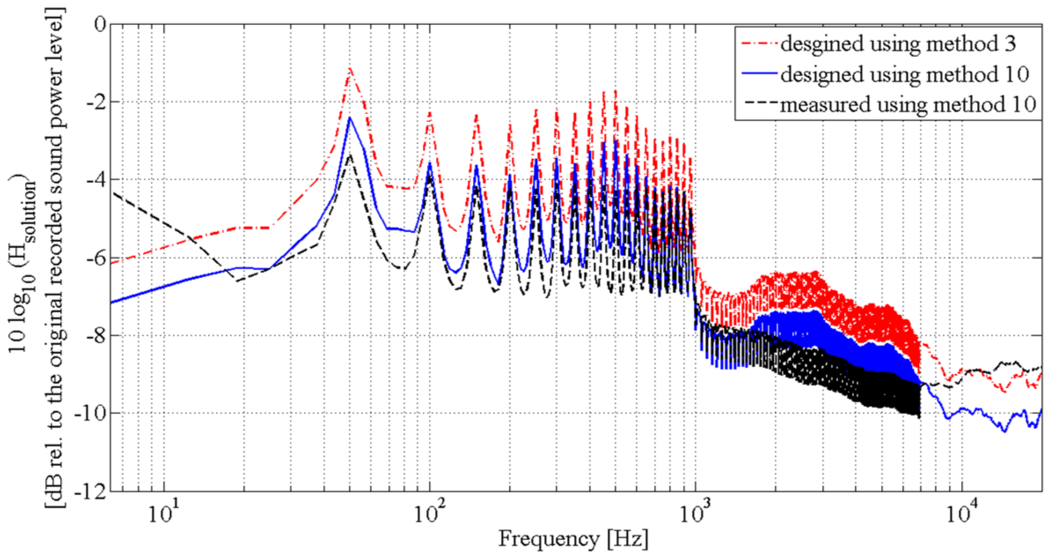


Figure 3.33: Comparison among the frequency response of the designed filters using method 3, method 10 and the measured filter using method 10.

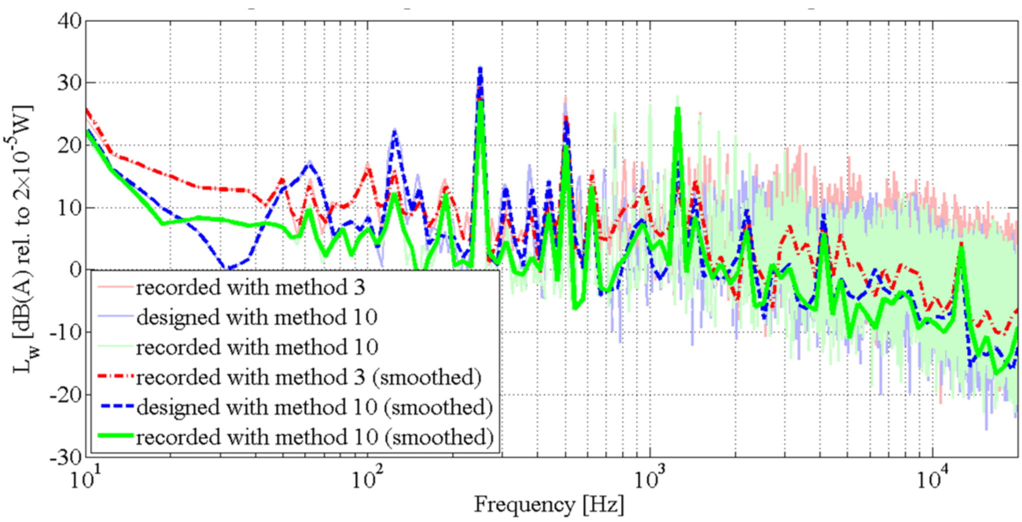


Figure 3.34: A-weighted sound power level among the recorded sound using method 3, the designed sound and the recorded sound after using method 10, the smoothed curve is plotted in the bands whose center frequency, lower frequency, and upper frequency have the relation of $f_u/f_c = f_c/f_l = 2^{1/10}$.

According to fig. 3.33 and fig. 3.34, one can see that at frequencies below 7 kHz, the designed filter using method 10 attenuates more at peaks compared to method 3, especially at frequencies between 2 kHz and 7 kHz. As the frequency increases to around 15 kHz, there is a resonance behavior becuase the wavelength approaches the dimension of the attached mass.

Reducing high frequency components and rattling

It is also meaningful to combine method 9 and method 10 to get a higher reduction of the high frequency components as well as the rattling. Hence, a potential target sound was created using method 11 as shown in fig. 3.35.

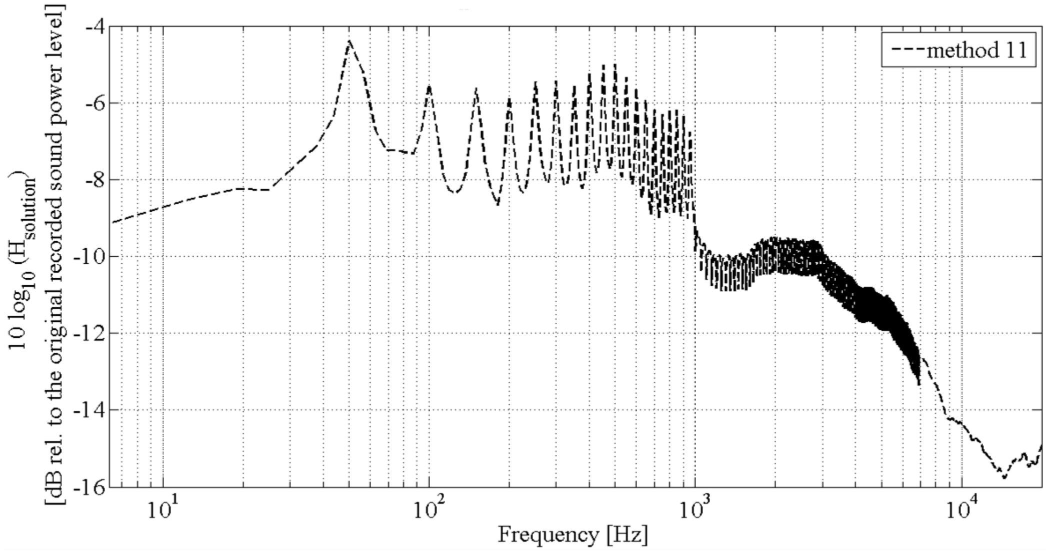


Figure 3.35: Frequency response of designed filter using method 11 for reduction of high frequency and rattling.

This can be achieved by using damping layer 1 and absorber 1 in the same positions of the coffee machine as method 9 and method 10. As the damping layers and absorbers used in this case were already heavier compared to the plastic components of the coffee machine, no attached mass was implemented to suppress the energy in order to avoid too strong low frequency components which were also unfavorable according to the jury group members.

Figure 3.36 and fig. 3.37 illustrate the frequency response of the measured filter using method 11, and the A-weighted sound power level of the designed sound sketch and the recorded sound after using method 11. An increased reduction was obtained at peak frequencies and at high frequencies as expected. But at frequencies above 6 kHz, the reduction of the recorded sound using method 11 is less than that of the designed sound.

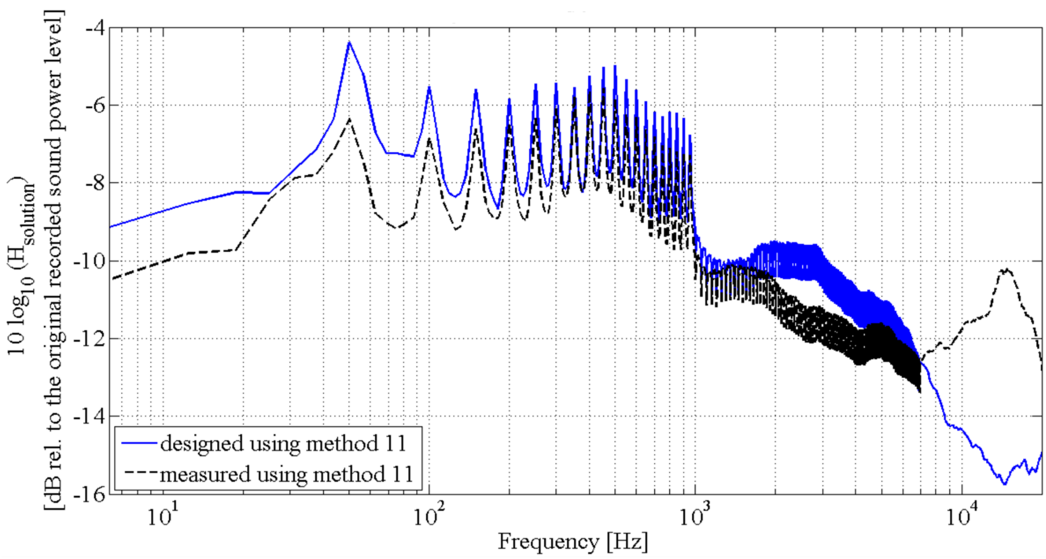


Figure 3.36: Comparison between the frequency response of the designed filter and the measured filter using method 11.

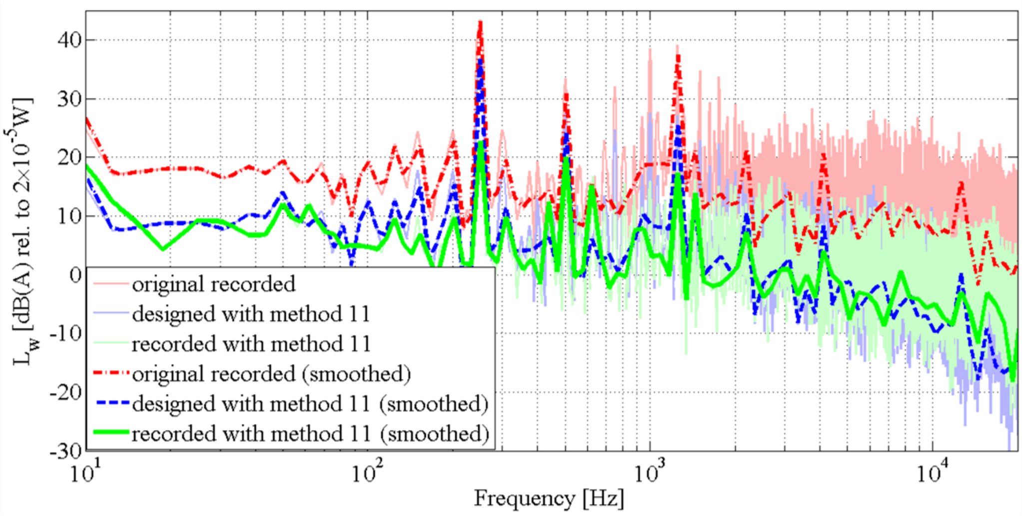


Figure 3.37: A-weighted sound power level among the original recorded sound, the designed sound and the recorded sound after using method 11, the smoothed curve is plotted in the bands whose center frequency, lower frequency, and upper frequency have the relation of $f_u/f_c = f_c/f_l = 2^{1/10}$.

Adding intentional sounds

Two sound sketches with intentional sounds were suggested by Peter Mohlin, the supervisor of this thesis work. The high frequency components and the rattling were masked in both sound sketches using suitable intentional sounds.

3.4 Second jury group evaluation

The second evaluation was performed on the 9th of October, 2014 and took around an hour. There were 6 participants in the evaluation, including one from A2Zound, and one from Department of Applied Acoustics, Chalmers. The average age of the participants was 26 years old. None of the participants reported hearing impairment.

The basic information of participants is summarized in Fig. 3.38:

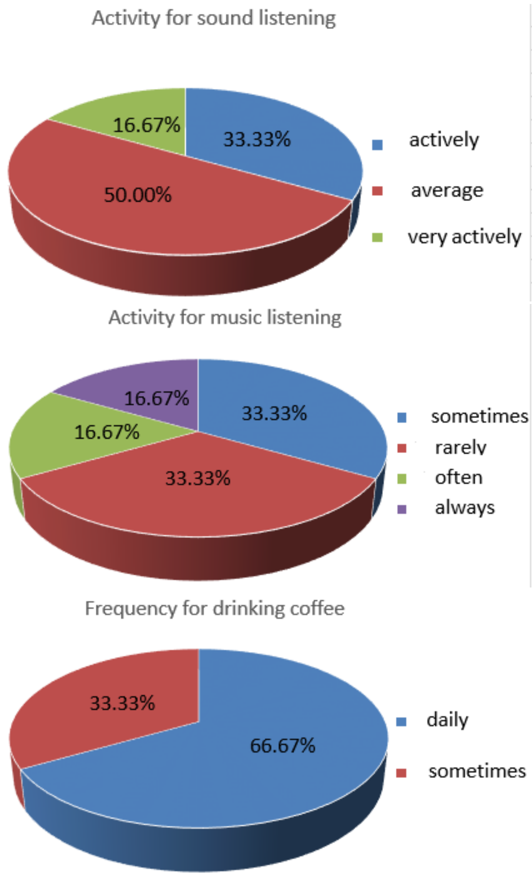


Figure 3.38: Basic information of the participants in the second jury group evaluation

Same equipments and questionnaires were used in the second jury group evaluation. The evaluation was held in the same room with same arrangement of furniture and decoration. The positions of loudspeakers, the coffee machine, and the participants as well as the setup of equipments used in the second jury group evaluation were kept the same as the first evaluation.

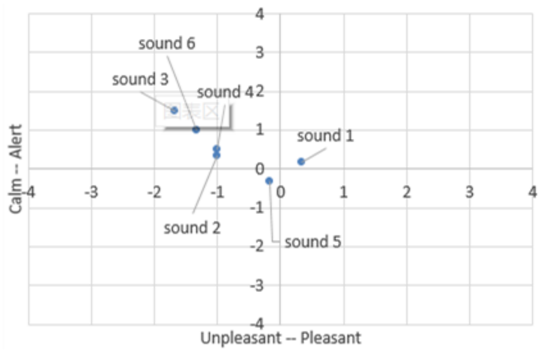


Figure 3.39: Emotional responses for potential target sounds.

3.4.1 Results of the second jury group evaluation

In this section, the results from the second jury group evaluation are presented in the same structure of the first evaluation.

Table 3.11 lists the sound sketches used in the first jury group evaluation.

Table 3.11: The sound sketches used during the second jury group evaluation.

No.	Name	Description
1	Sound 1	the sound with intentional sound 1
2	Sound 2	the sound with intentional sound 2
3	Sound 3	the original recorded sound
4	Sound 4	using method 9
5	Sound 5	using method 11
6	Sound 6	using method 10

Emotional responses

The emotional responses were investigated by calculating the sample mean values of two scales from jury group members, one is from Calm to Alert, and the other scale is from Unpleasant to Pleasant. All the sound sketches were scaled using these two scales.

Figure 3.39 shows the emotional responses mapping of the potential target sounds. One can see that all designed sound sketches were rated as more pleasant compared to the original recorded sound. Sound 6 had the lowest activation score due to the reduced rattling and lower pitch. Sound 1 reaches the highest pleasantness ranking as a result of the added intentional sound.

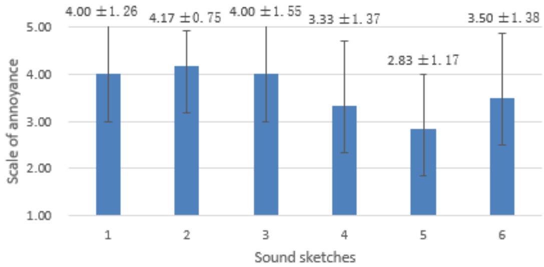


Figure 3.40: Mean value and standard deviation of scale of annoyance for potential target sounds.

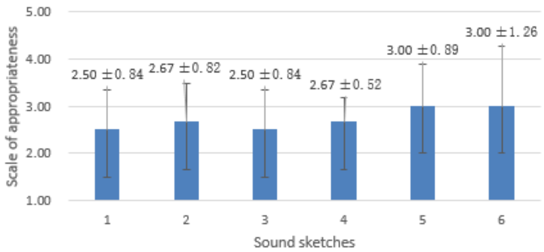


Figure 3.41: Mean value and standard deviation of scale of appropriateness for potential target sounds.

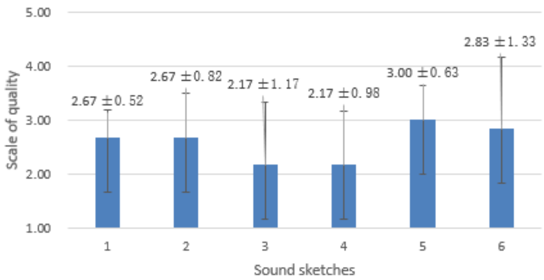


Figure 3.42: Mean value and standard deviation of scale of product quality for potential target sounds.

The comments from group members for each scale are deviated more compared to the first jury group evaluation as shown in figure **??**. According to the questionnaires, sound 3 (the original recorded sound) is of the lowest quality ranking and the lowest appropriacy as shown in fig. 3.40 to fig. 3.42. Sound 5 and sound 6 are rated as the most appropriate sounds for a coffee machine of high quality for the reason that both are smoother than the other sounds. Sound 5 has lower pitch than sound 6 and is therefore considered to be of higher quality. Sound 1, sound 2 and sound 3 were rated similarly in annoyance, but sound 1 was judged as the most annoying sound because the intentional sounds gave the participants the impression of "complicated" and "sad".

Selection of the favorite and least preferred sounds

Sound 2, the sound designed with intentional sound 2nd, was selected to be the favorite sound by three participants even if none of them considered it a appropriate sound for a coffee machine. The main reasons were:

a) attractive intentional sounds;

b) smoother rattling components;

c) low pitch.

Sound 3, the original recoded sound, was voted to be the least favorite sketch due to the following reasons:

a) the high frequency components lead to an annoying sharpness;

b) less low frequency components remind people of poor quality;

c) the clear rattling pattern;

d) the fluctuation in tone makes the machine sounds unstable.

The comments for each sound sketch and the reasons why participants liked or disliked each sound were discussed during the group discussion. Most of the comments are summarized in table 3.12:

Table 3.12: Summary of comments for each potential target sound.

Sound no.	Likes	Dislikes
Sound 1	intentional sound;	sounds like a computer
Sound 2	intentional sounds; smoother rattling; low pitch; fade-in and fade-out of intentional sound;	decreasing tone; combined intentional sound; fluctuation in intentional sound
Sound 3	sounds like a coffee machine;	high frequency component; fluctuation; rattling pattern;
Sound 4	sounds like a coffee machine;	too strong low frequency components; rattling pattern;
Sound 5	low pitch; smoother rattling;	N/A
Sound 6	smoother rattling;	N/A

Discussion on expectation and avoidance

During the final part of the jury group evaluation, what to expect and avoid in final target sounds were discussed. The comments are summarized in table 3.13.

Table 3.13: Summary of comments for future sound sketches.

Category	Expectation	Avoidance
Suggestion	smoother rattling; low pitch; proper intentional sound	rattling pattern; fluctuations; high pitch;

Apart from the listed comments in table 3.13, there were many suggestions about the added intentional sounds, for instance, to make them sound as a flowing water to remind users of the water pumped up from the tank during the early part; to lower the tone and the loudness at the beginning and the end, but increase them in the middle. It is difficult to compose the most appropriate intentional sound to obtain low annoyance and high quality due to the limitation of time, therefore, the second favorite sound, sound 5, which has the lowest annoyance score and the highest ranking of quality and appropriacy, was selected to be the final target sound.

3.4.2 The second improvement of sound sketches

According to the results from the second jury group evaluation, the designed sound using the method 11 was selected to be the final target sound. This sound sketch had already been improved during the first evaluation and no further improvement were made.

3.5 Mechanical modification

In this section, mechanical modifications made on the coffee machine to achieve the final target sound are introduced, including the modifications on the generator, the transmission paths and the radiating surfaces. After applying modifications, the sound of the coffee machine was recorded and compared with the final target sound. The physical feasibility was analyzed to provide possibility of further modifications either on the coffee machine or on the designed sound to get a better match.

Modifying the generator

Four prices of extra mass (each mass equals to 10 g) were attached on the pump to increase the mass per area at the excitation point. On the surface of the attached mass, a piece of absorber 1 was attached to reduce the radiated sound from the mass surfaces. The treatments on the generator is shown in fig. 3.43.

Modifying the transmission paths

Three pieces of damping layer 1 were attached on the contact surfaces between the pump and the main body, and between the sliders on either side and the main body to attenuate the impact peaks. Two mass blocks of zinc (each weights 10 g) were attached to the side chassis as blocking mass, some pieces of absorber 1 were attached on the surfaces of the zinc block to reduce the radiated energy from the metal surfaces. The implementation of treatments are shown in fig. 3.44.

Modifying the radiating surfaces

Absorbers were applied on the top plate and the metal handle to reduce the radiated sound. Two pieces of zinc (each mass weights 10 g) were attached on the sliders as blocking mass.

3.5.1 Sound comparison and physical feasibility analysis

The sound of the coffee machine with treatments mentioned above was recorded and compared to the final target sound as shown in fig. 3.46.

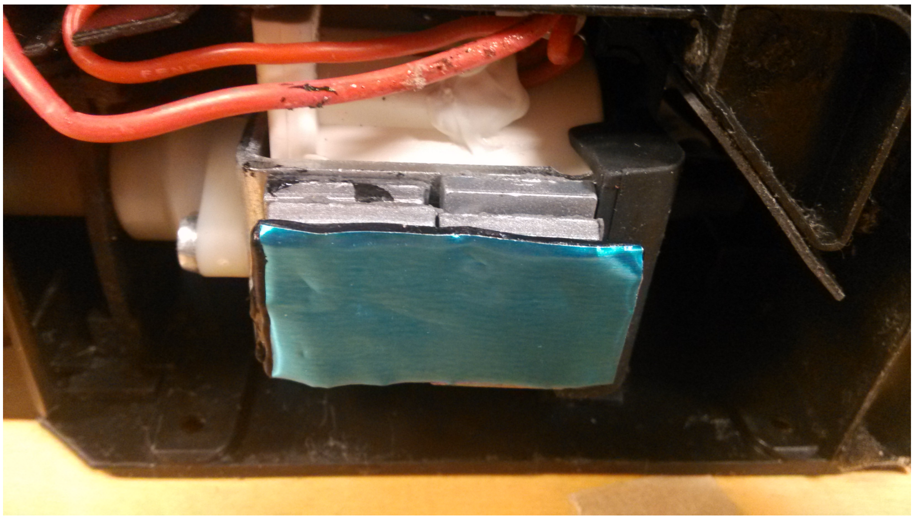
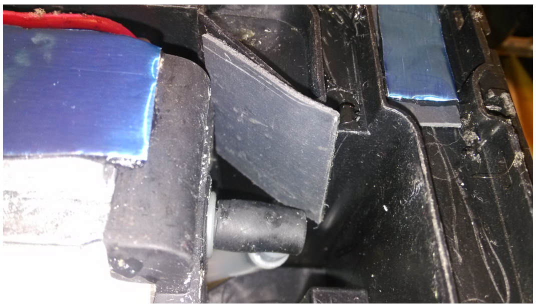
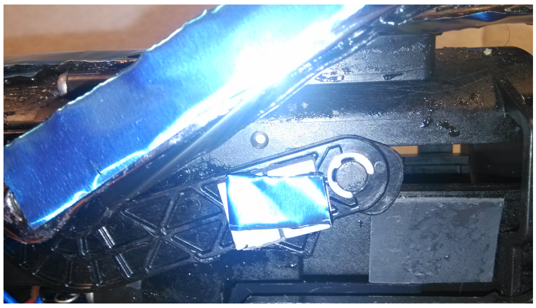


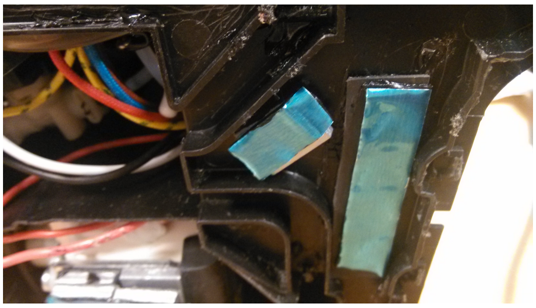
Figure 3.43: Treatments on the generator.



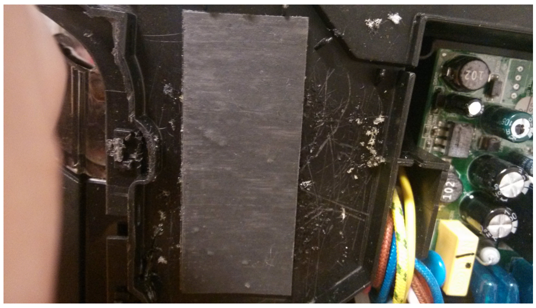
(a) damping layer at excitation point.



(b) damping layer between the slider and the main body.

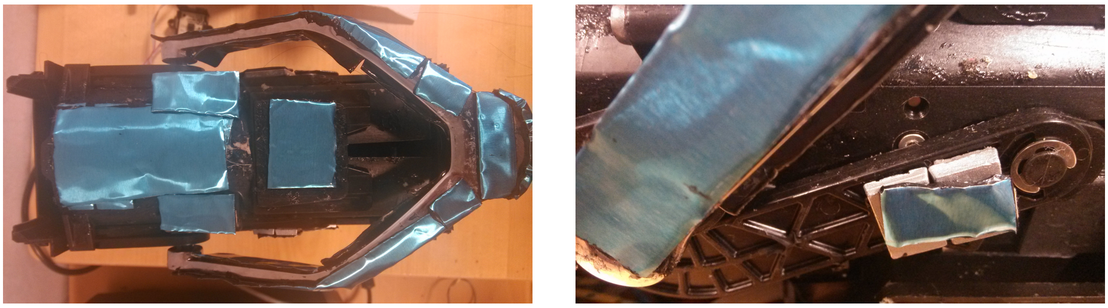


(c) blocking mass on side chassis.



(d) damping layer on side chassis

Figure 3.44: Treatments of the damping layers on the contact surfaces (a) and (b), the blocking mass on the side chassis (c), and the damping layers on the side chassis (d).



(a) absorber on the top plate.

(b) blocking mass on sliders and absorber on metal handle.

Figure 3.45: Treatments of the absorber on the top plate and the metal handle (a), and the blocking mass on the slider (b).

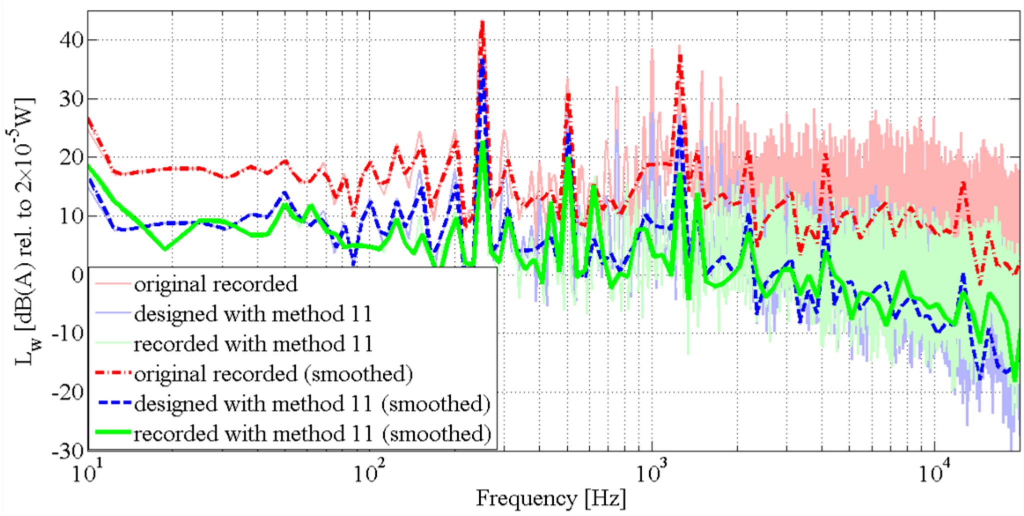


Figure 3.46: A-weighted sound power level of the original recorded sound ,the designed and recorded sound using method 11, the smoothed curve is plotted in the bands whose center frequency, lower frequency, and upper frequency have the relation of $f_u/f_c = f_c/f_l = 2^{1/10}$.

The recorded sound has a lower reduction at frequencies above 7 kHz than desired, hence, further modifications are required. Based on the studies in section 3.1, sound reduction above 7 kHz can be achieved by modifying the pump frame and the top plate. And according to the studies in section 2.2, the method of damping layer, sandwich structure or attached mass can be applied to suppress the energy above 7 kHz efficiently. Due to the limitation of space, adding damping material or extra mass on the pump frame or the top plate are not allowed. Hence, the existent material and structure were taken into consideration. As shown in fig. 3.47, the original damping material on the pump frame is very thin and soft (low Young's modulus). As the loss factor is proportional to the product of thickness and Young's modulus, therefore, the original damping was

(a) the existent damping. (b) replaced by damping 1.

Figure 3.47: The damping on the pump frame.

replaced with damping layer 1 which is of an increased thickness and higher Young's modulus to introduce a higher reduction.

Table 3.14: The methods used after physical feasibility analysis.

No.	Name	Description
1	Method 12	more reduction at frequencies above 7 kHz by replacing the existent damping of pump.

The comparison between the A-weighted sound power level of the recorded sound using method 12 and the the designed sound using method 11 is shown in fig. 3.48.

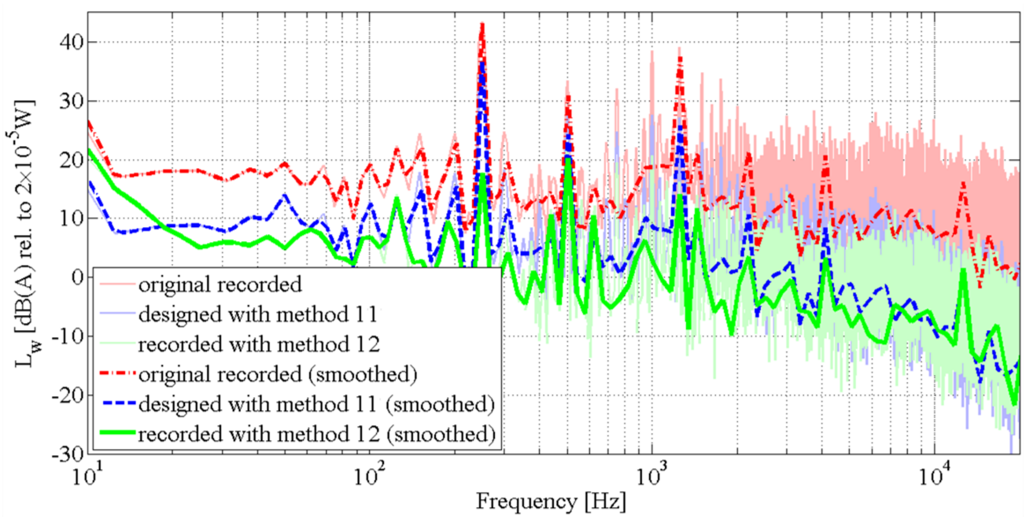


Figure 3.48: A-weighted sound power level of the original recorded sound, the designed sound using method 11, and the recorded sound using method 12, the smoothed curve is plotted in the bands whose center frequency, lower frequency, and upper frequency have the relation of $f_u/f_c = f_c/f_l = 2^{1/10}$

From fig. 3.48 and fig. 3.37, one can see a higher reduction above 7 kHz compared to the recorded sound using method 11, but the reduction is still lower than expectation when compared to the designed sound using method 11. However, there is no enough room for further treatment to achieve the desired reduction.

4

Discussions and conclusions

Sound sketching and sound evaluation is an iterative process. During sound sketching, the filters can be designed based on the sound characteristics and also vibro-acoustics. For example, the cut-off frequency of a filter that simulates an absorber can be decided approximately according to the dimension of product components. The value of reduction can be calculated based on the thickness, density and Young's modulus of the treatment materials and the product material.

From the results of jury group evaluations, a sound with low pitch, less rattling is taken to be the favorite product sound for a coffee machine. Meanwhile, proper intentional sounds which can remind the users of the product are preferred by users. However, a sound with too strong low frequency components is less preferred. During jury group evaluation, it is important to remind the group members of the product. In the second jury group evaluation, the participants were attracted by the added intentional sounds and selected the sound to be the favorite sketch even if they did not think it was an appropriate sound for a coffee machine. According to the remarks from the participants, it is important to make the intentional sound relevant to the product itself. For instance, sound 2 in the second jury group evaluation (the designed sound with intentional sound 2) was preferred by half of the jury group members, but it was also scaled to be an inappropriate sound due to the fact that members could not image a working coffee machine from that sound.

The X_ω mentioned in eq. (1.1) requires the information of the generator, for instance, the impact excitation. The frequency characteristics or the magnitude of excitation should be investigated during the vibro-acoustic study. Using this information, one can design the filter of $H_{solution}$ for the generator to smooth the product sound (impact rattling). The machinery structure should be studied to obtain the highest correlated machine components with the interesting sound characteristics and the frequency response of different transmission paths relative to the excitation point. This information leads to $H_{structure}$. Based on $H_{structure}$, one can design the filter of $H_{solution}$ for the transmission paths and the radiating surfaces. $H_{radiation}$ can be controlled by fixing the room acoustic parameters, and the position of the product and the microphones, such as positions. With X_ω, $H_{structure}$, $H_{solution}$ and $H_{radiation}$, the target product sound Y_ω is obtained.

After sound sketching, the product can be modified according to the expectations. Analysis in physical feasibility is necessary to decide whether more improvements are allowed on the product under the limitations of space or weight.

In general, it is possible to explore a methodology for consequential product sound design based on the recorded original sound and the knowledge of technical acoustics. The recorded original product sound allows the designers for sound analysis and diagnosis to figure out the interesting characteristics. Different methods of vibro-acoustic treatments can be simulated as simplified filters that can be used to create target sounds. The sound sketches can be evaluated by jury group evaluations and improved according to the results from the evaluations to obtain a final target sound. After that, different treatments can be applied on the product to obtain the desired improvement. An iterative process containing sound comparison between the recorded sound and the final target sound, and physical feasibility analyses are required. The machinery structure is further modified to achieve the final target sound. In some cases, the final target sound should be modified due to the limitation of physics.

5

Future work

Future work can be done for more accurate filters, for instance, the value of reduction can be calculated under bending wave incident. And there are some general physical parameters that are relevant for the sound characteristics, such as the speed of rotation, the number of fans. A model based on these general parameters will help the designers more during sound sketching to identify the limitations of improvements.

From the sound designer's perspective, there are certain things which should be noticed during the sound sketching and evaluation. The most important one is that the sound can remind the users of the product. Otherwise, the sound might be rated high in quality and pleasant, but low in appropriateness. The participants should be reminded of the product to avoid a selection for an interesting sound which is not appropriate for the product. From the results of the jury group evaluations, one can find that people of different age react to the same sound differently. Aged users prefer the product sound to be of less high frequencies and less fluctuation, while younger users prefer the sound to be of rich low frequency components and an increasing tone. Hence, the product sound should be designed for the target consumers who can be specific, for instance, by age or gender.

Moreover, from the results of jury group evaluations, intentional sound is preferred by users, hence, there is of value to design appropriate intentional sounds for the product. The intentional sounds can be controlled separately, hence, the users can select the sound they prefer.

Bibliography

[1] R. S. Thomas, P. J. Stanecki, H. P. Creutz, A development process to improve vehicle sound quality, Tech. rep., SAE Technical Paper (1991).

[2] K. Tsuge, K. Kanamaru, T. Kido, N. Masuda, A study of noise in vehicle passenger compartment during acceleration, Tech. rep., SAE Technical Paper (1985).

[3] K. Fujita, T. Nishiyama, Y. Hayakawa, Research on sound quality evaluation methods for exhaust noise, JSAE Review 9 (2).

[4] K. Genuit, R. Sottek, Objective and subjective analysis of the sound of door slamming, in: INTER-NOISE and NOISE-CON Congress and Conference Proceedings, Vol. 1995, Institute of Noise Control Engineering, 1995, pp. 921–926.

[5] G. Irato, G. Ruspa, A. Tarzia, Noise quality of transient sounds: An application to the noise emitted by push-push buttons on the control panel of a car, in: INTER-NOISE and NOISE-CON Congress and Conference Proceedings, Vol. 1995, Institute of Noise Control Engineering, 1995, pp. 943–946.

[6] N. C. Otto, G. H. Wakefield, A subjective evaluation and analysis of automotive starter sounds, Noise Control Engineering Journal 41 (3) (1993) 377–382.

[7] M. Bodden, H. Iglseder, Active sound design: Vacuum cleaner, in: Revista de Acustica (special issue Forum Acusticum, Sevilla, Spain), Vol. 33, 2002.

[8] S. K. Goumas, M. E. Zervakis, G. S. Stavrakakis, Classification of washing machines vibration signals using discrete wavelet analysis for feature extraction, Instrumentation and Measurement, IEEE Transactions on 51 (3) (2002) 497–508.

[9] J. Y. Jeon, J. You, H. Y. Chang, Sound radiation and sound quality characteristics of refrigerator noise in real living environments, Applied Acoustics 68 (10) (2007) 1118–1134.

[10] R. C. Berfield, Noise reducing means for vacuum cleaner, uS Patent 4,435,877 (Mar. 13 1984).

[11] R. Lyon, Designing for product sound quality, CRC Press, 2000.

[12] H. Van der Auweraer, K. Wyckaert, W. Hendricx, From sound quality to the engineering of solutions for nvh problems: case studies, Acta Acustica united with Acustica 83 (5) (1997) 796–804.

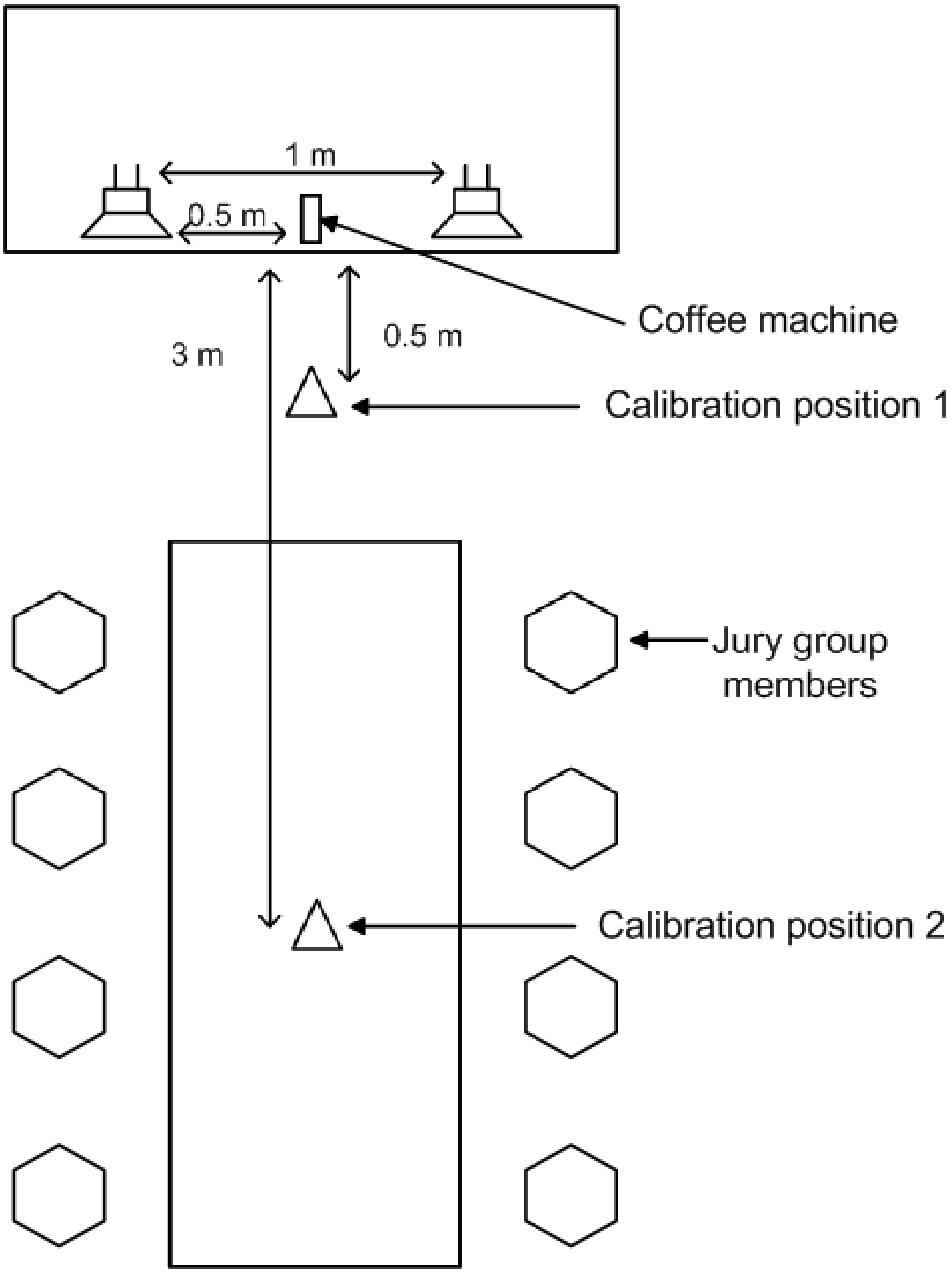
[13] L. Cremer, M. Heckl, B. A. Petersson, Structure-borne sound: structural vibrations and sound radiation at audio frequencies, Springer, 2005.

A

Appendix A

Setup of jury group evaluation

SETUP OF JURY GROUP EVALUATION



B

Appendix B

Questionnaire for first jury group evaluation

Welcome to a jury group evaluation of coffee machine sounds!

Participant background:

Age: _____ years

Gender: (*select one*) ☐ Male ☐ Female

Job/Occupation: _______________

Hearing impairment: (*select one*) ☐ Yes ☐ No

Highest education level completed:

☐ Less than High School

☐ High School/GED

☐ College

☐ Bachelor's Degree (BA, BS)

☐ Master's Degree

☐ Doctoral Degree (PhD)

How actively do you listen to sounds (apart from music)?

1	2	3	4	5
Not actively at all	Not so actively	Average	Actively	Very actively

How often do you listen to the sounds from a coffee machine?

1	2	3	4	5
Never	Rarely	Sometimes	Often	Always

Do you drink coffee?

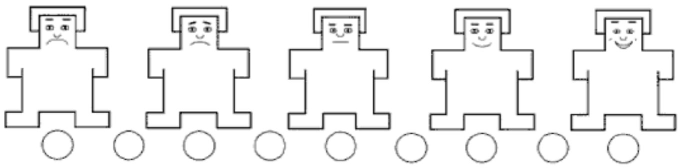
1	2	3
No	Yes, sometimes	Yes, daily

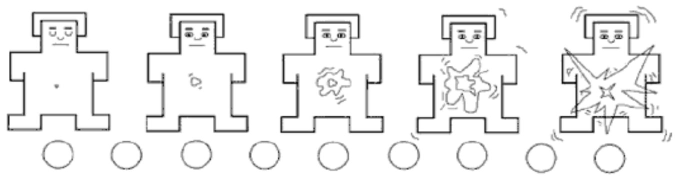
Part 1. Scales

Six different sound sketches will be presented. For each sound, there are five questions required to be answered. Each sound sketch lasts for 4 seconds, and is repeated as many times as you find necessary. You are not allowed to discuss these questions between each other in this section.

The two SAM (Self-Assessment Manikin) scales below represent your feelings when you have listened to the sounds. The top scale represents pleasantness, ranging from unpleasant (left) over neutral, to pleasant (right). The scale below represents activation, from low activation, relaxed/tired (left) to high activation, awake, active (right).

Sound 1





1. **Please mark how you felt when hearing the sound using both scales.**

2. Do you think the sound is **annoying**?

1	2	3	4	5
No, not annoying				Yes, very annoying

3. Do you think the sound is **an appropriate sound for a coffee machine?**

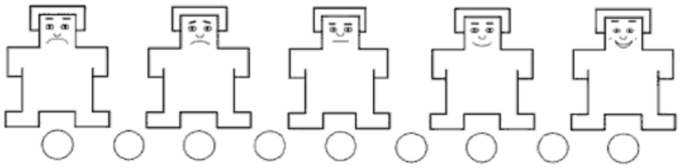
1	2	3	4	5
Very inappropriate	Inappropriate	Average	Appropriate	Very appropriate

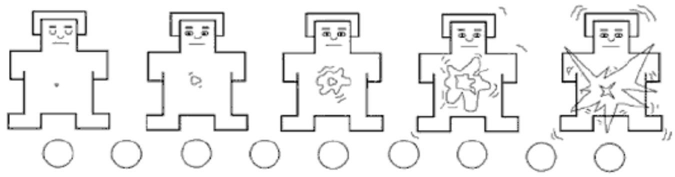
4. **Based on the sound, how would you rate the quality of the coffee machine?**

1	2	3	4	5
Very low quality	Low quality	Average quality	High quality	Very high quality

Comments: (brief explanation on *why* you like or dislike this sound for a coffee machine)

Sound 2





1. **Please mark how you felt when hearing the sound using both scales.**

2. Do you think the sound is **annoying**?

1	2	3	4	5
No, not annoying				Yes, very annoying

3. Do you think the sound is **an appropriate sound for a coffee machine?**

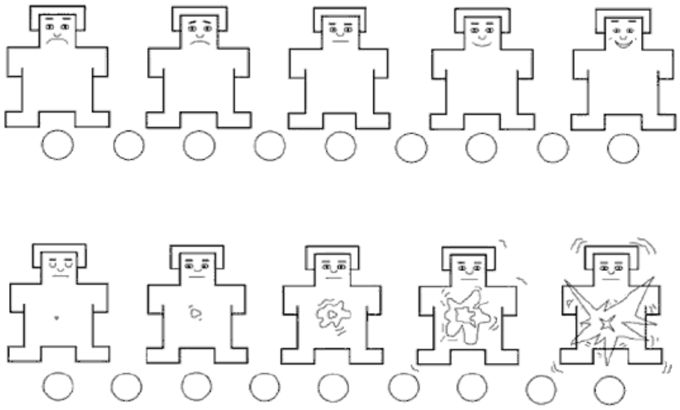
1	2	3	4	5
Very inappropriate	Inappropriate	Average	Appropriate	Very appropriate

4. **Based on the sound, how would you rate the quality of the coffee machine?**

1	2	3	4	5
Very low quality	Low quality	Average quality	High quality	Very high quality

Comments: (brief explanation on *why* you like or dislike this sound for a coffee machine)

Sound 3



1. **Please mark how you felt when hearing the sound using both scales.**

2. Do you think the sound is **annoying**?

1	2	3	4	5
No, not annoying				Yes, very annoying

3. Do you think the sound is **an appropriate sound for a coffee machine?**

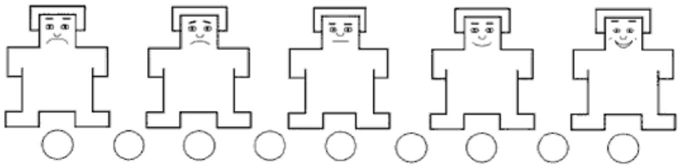
1	2	3	4	5
Very inappropriate	Inappropriate	Average	Appropriate	Very appropriate

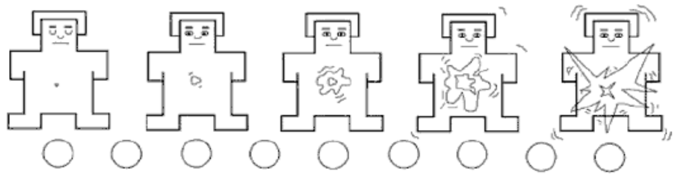
4. **Based on the sound, how would you rate the quality of the coffee machine?**

1	2	3	4	5
Very low quality	Low quality	Average quality	High quality	Very high quality

Comments: (brief explanation on *why* you like or dislike this sound for a coffee machine)

Sound 4





1. **Please mark how you felt when hearing the sound using both scales.**

2. Do you think the sound is **annoying**?

1	2	3	4	5
No, not annoying				Yes, very annoying

3. Do you think the sound is **an appropriate sound for a coffee machine?**

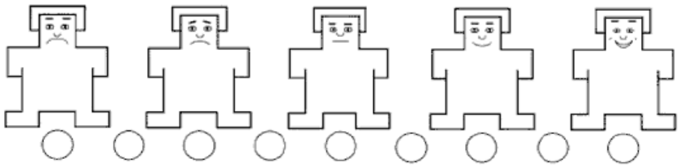
1	2	3	4	5
Very inappropriate	Inappropriate	Average	Appropriate	Very appropriate

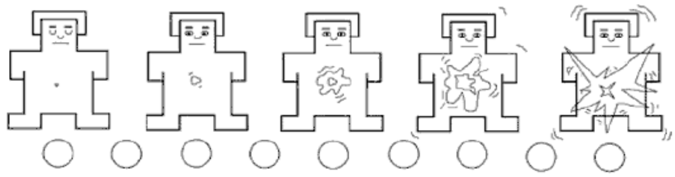
4. **Based on the sound, how would you rate the quality of the coffee machine?**

1	2	3	4	5
Very low quality	Low quality	Average quality	High quality	Very high quality

Comments: (brief explanation on *why* you like or dislike this sound for a coffee machine)

Sound 5





1. **Please mark how you felt when hearing the sound using both scales.**

2. Do you think the sound is **annoying**?

1	2	3	4	5
No, not annoying				Yes, very annoying

3. Do you think the sound is **an appropriate sound for a coffee machine?**

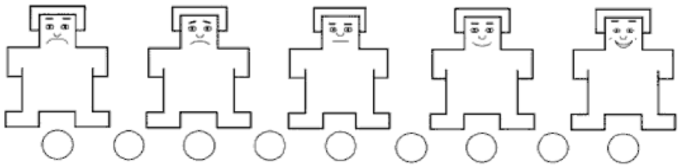
1	2	3	4	5
Very inappropriate	Inappropriate	Average	Appropriate	Very appropriate

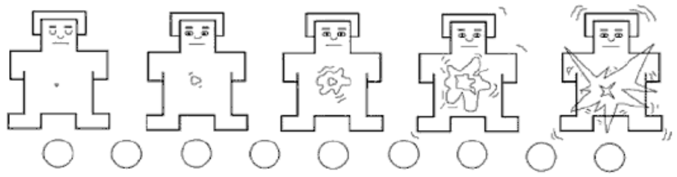
4. **Based on the sound, how would you rate the quality of the coffee machine?**

1	2	3	4	5
Very low quality	Low quality	Average quality	High quality	Very high quality

Comments: (brief explanation on *why* you like or dislike this sound for a coffee machine)

Sound 6





1. **Please mark how you felt when hearing the sound using both scales.**

2. Do you think the sound is **annoying**?

1	2	3	4	5
No, not annoying				Yes, very annoying

3. Do you think the sound is **an appropriate sound for a coffee machine?**

1	2	3	4	5
Very inappropriate	Inappropriate	Average	Appropriate	Very appropriate

4. **Based on the sound, how would you rate the quality of the coffee machine?**

1	2	3	4	5
Very low quality	Low quality	Average quality	High quality	Very high quality

Comments: (brief explanation on *why* you like or dislike this sound for a coffee machine)

Part 2. Selection

The six sound sketches will be presented again in one sequence and the participants will be required to review their questionnaire and select one favorite and one least favorite sketches. Explain your selection and focus on why you dislike/like the sound:

Part3. Discussion

Group discussion is based on the questionnaire and the selection done before. Focus on why you dislike/like the sound.

Thank you for your participation!

C

Appendix C

Results from jury group evaluations

QUESTION NO.	1		2	3	4
		sound 1			
participant	happiness	activity	annoying	appropriate	quality
1	1	1	5	2	2
2	-2	2	5	3	3
3	-2	0	4	1	2
4	2	-2	5	3	3
5	1	1	3	3	3
6	2	-1	2	3	3
intentional 1	0.3333333	0.166667	4	2.5	2.666667
		sound 2			
participant no.	happiness	activity	annoying	appropriate	quality
1	0	1	5	2	2
2	0	-1	3	4	4
3	-2	0	4	3	3
4	-2	0	5	2	2
5	-1	1	4	3	3
6	-1	1	4	2	2
intentional 2	-1	0.333333	4.166667	2.666666667	2.666667
		sound 3			
participant no.	happiness	activity	annoying	appropriate	quality
1	0	2	4	3	3
2	-4	4	5	2	2
3	-4	2	5	2	1
4	4	-4	1	4	4
5	-4	3	5	2	1
6	-2	2	4	2	2
original	-1.666667	1.5	4	2.5	2.166667
		sound 4			
participant no.	happiness	activity	annoying	appropriate	quality
1	1	2	3	3	3
2	-4	3	4	2	2
3	-2	0	4	3	1
4	2	-4	1	3	3
5	-3	2	5	2	1
6	0	0	3	3	3
high frequency	-1	0.5	3.333333	2.666666667	2.166667
		sound 5			
participant no.	happiness	activity	annoying	appropriate	quality
1	-1	3	3	3	3
2	1	-2	2	4	4
3	0	-2	3	4	3
4	0	0	1	3	3
5	0	0	4	2	2
6	-1	-1	4	2	3
rattling and hf	-0.166667	-0.33333	2.833333	3	3
		sound 6			
participant no.	happiness	activity	annoying	appropriate	quality
1	-1	2	3	4	4
2	-4	3	4	2	2
3	-2	2	4	3	2
4	4	-4	1	5	5

5	-3	2	5	2	2
6	-2	1	4	2	2
rattling	-1.333333	1	3.5	3	2.833333

QUESTION NO.	1		2	3	4
		sound 1			
participant	happiness	activity	annoying	appropriate	quality
1	1	1	5	2	2
2	-2	2	5	3	3
3	-2	0	4	1	2
4	2	-2	5	3	3
5	1	1	3	3	3
6	2	-1	2	3	3
intentional 1	0.3333333	0.166667	4	2.5	2.666667
		sound 2			
participant no.	happiness	activity	annoying	appropriate	quality
1	0	1	5	2	2
2	0	-1	3	4	4
3	-2	0	4	3	3
4	-2	0	5	2	2
5	-1	1	4	3	3
6	-1	1	4	2	2
intentional 2	-1	0.333333	4.166667	2.666666667	2.666667
		sound 3			
participant no.	happiness	activity	annoying	appropriate	quality
1	0	2	4	3	3
2	-4	4	5	2	2
3	-4	2	5	2	1
4	4	-4	1	4	4
5	-4	3	5	2	1
6	-2	2	4	2	2
original	-1.666667	1.5	4	2.5	2.166667
		sound 4			
participant no.	happiness	activity	annoying	appropriate	quality
1	1	2	3	3	3
2	-4	3	4	2	2
3	-2	0	4	3	1
4	2	-4	1	3	3
5	-3	2	5	2	1
6	0	0	3	3	3
high frequency	-1	0.5	3.333333	2.666666667	2.166667
		sound 5			
participant no.	happiness	activity	annoying	appropriate	quality
1	-1	3	3	3	3
2	1	-2	2	4	4
3	0	-2	3	4	3
4	0	0	1	3	3
5	0	0	4	2	2
6	-1	-1	4	2	3
rattling and hf	-0.166667	-0.33333	2.833333	3	3
		sound 6			
participant no.	happiness	activity	annoying	appropriate	quality
1	-1	2	3	4	4
2	-4	3	4	2	2
3	-2	2	4	3	2
4	4	-4	1	5	5
5	-3	2	5	2	2

6	-2	1	4	2	2
rattling	-1.333333	1	3.5	3	2.833333