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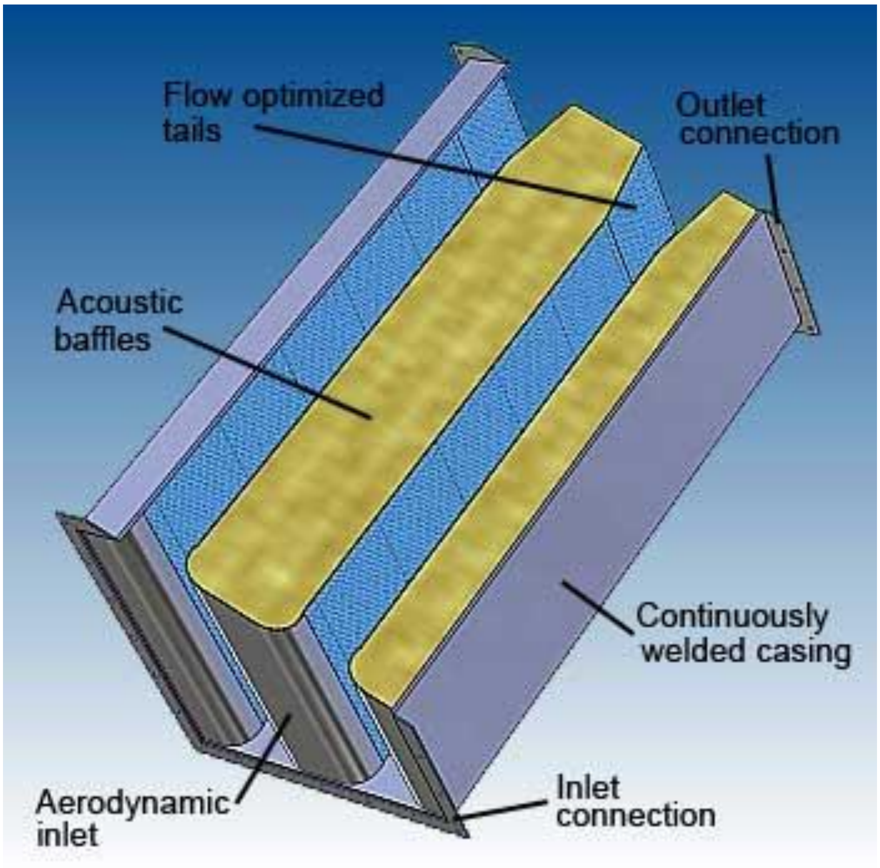

Calculation methods for predicting attenuation of parallel baffle type silencers

For use in power plant ducts and exhausts

Master’s Thesis in the Master’s programme in Sound and Vibration

DARIO BOGDANOVIC

Department of Civil and Environmental Engineering
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CHALMERS UNIVERSITY OF TECHNOLOGY

Göteborg, Sweden 2014
Master´s Thesis 2014:145

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Cover:
Typical appearance of a parallel baffle silencer, explained in detail in chapter 2.

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ABSTRACT

Power plant can be immensely loud facilities and are often in need of substantial sound reducing measures. Parallel baffle silencers are often used in ducts and exhausts, which can contain hot and eroding gases, requiring exotic materials to withstand such conditions. Attenuation for these silencers is therefore important to know beforehand. This thesis investigates available literature to find methods for predicting the attenuation of baffle silencers. The methods found are evaluated for accuracy and usability and are presented so that they can be used by other engineers. Two older methods were found that could be readily used and presented satisfactory results, with one of them being recommended over the other. Additional methods have been obtained but require numerical solutions to yield results. The recommended method has been converted into a MATLAB script for simplified calculations.

AKNOWLEDGEMENTS

I would like to extend my deepest gratitude to my supervisors, Martin Almgren from ÅF and Jens Forsén, associate professor at Technical Acoustics, for helping and guiding me throughout the duration of the thesis work. It is through their guidance that this thesis was completed. I would also like to extend my thanks to Ingrid-Marie Lindqvist for her support and help in testing various parts of the work done in the thesis.

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1 INTRODUCTION

1.1 Background

The power plant industry is a large business with large amounts of money being invested in various facilities. Most of these plants have some sort of gas turbines or other very noisy equipment inside them, resulting in high sound levels around the perimeter of the plant. Some of these plants might even be located close to living quarters and other forms of public places that people occupy. The closer the plants are to people the stricter the demands for reasonable sound levels. Depending on the size and the type of machines used in the power plant, sound levels can range between 120 dBA and 155 dBA which puts a large emphasis on reducing the sound to acceptable levels (Ray, 2006). Not only can this cause issues such as sleep disturbance and noise annoyance at housings in close proximity to power plants, such high sound levels are also harmful to employees and staff, further increasing the need for measures to reduce the sound. Since the plants often deal with very hot gases, sometimes containing chemicals that might have eroding properties, exotic and expensive materials are needed when creating the sound absorbers that are meant to last. As such, the properties and the size of the absorbers need to be known well in advance before they are implemented in the final product. Given the various dimensions of the ducts in need of absorption, the generated sound levels and other factors that may affect the outgoing sound, such as pressure drops and flow induced noise, the absorbers have to be accurately calculated so as to be cost efficient. Sizing a silencer correctly also allows for the right amount of materials being used, leading itself to be more environmentally benign as well. While methods to reduce sound in power plant air intakes and exhausts have existed for a lot of years, mostly through baffle silencers, there has not always been a definite theory behind the prediction of the silencers. Some companies do not involve theory at all, instead they are using pre-existing values of transmission loss that has been developed by tests performed in labs. These tests are performed by the manufacturers of the silencers themselves, resulting in a lot of manufacturer data on silencer absorption (Galaitsis & Vér,1992). One problem that arises from this is that the method of choosing the correct silencer is not keeping pace with the advance in technology and new scientific discoveries (Anonymous, 1999). This leads to that each manufacturer are using internal methods of calculating absorption or that they do not have any calculation methods at all, using old measurements that might be outdated or simply not accurate enough. By today’s standards an inaccurate method is simply not cost efficient enough and a fast production cycle is becoming more and more important. Even though this is the case some manufacturers might not be willing to change their methods that have been used for several years. Their method for prediction can be based on guaranteed insertion loss and experience of customer complaints. It is therefore important that new research and new discoveries are brought to light so that they can be implemented to produce better approximation models.

1.2 Purposes of the thesis

The purpose of this thesis will be to carry out a study on the sizing of parallel baffle silencers in order to determine the recommended approach for approximating their attenuation, presenting methodology that can easily be utilized by other engineers. Theories on predicting parallel baffle attenuation are rather by now several decades old and will have to be gathered for this study. This leads to another important aspect of this project, to find more recent and updated research and methods regarding parallel baffles. The age of the older theories makes it relevant to find newer and preferably more accurate methods of predicting attenuation. Of particular interest will be methods which are relatively simple to use and that produce easily interpreted results which can be used in engineering. While the methods themselves need not be overly simple their presentation in this study will be such that any engineer from any discipline will be able to utilize them. Methods and theories involving complex calculations will be included in the study but may not be recommended for use and may not be included in any evaluations.

A secondary purpose of this thesis is to use the recommended approach to predicting baffle attenuation and create computation software which performs the same calculations. This software will be made as a supplement to the methods and theories in this project and should not be used exclusively by itself. The purpose of this software is to primarily cut down the time spent making attenuation calculations but also to evaluate the various parameters used for the calculations.

1.3 Tasks and approach

The primary task of this thesis is to find, present and evaluate methods for predicting parallel baffle attenuation. Emphasis will be put in attempting to find more recent research and theories. Optimistically providing more accurate, or at least more updated, methods than the older, more established ones. All methods will have to be verified in order to ascertain their accuracy, preferably through comparison with measured attenuation values. Amongst the obtained methods the ones that are the most accurate and easiest to use will be highlighted and recommended for use. Additionally computational software will be made to complement the written methods.

To accomplish this, a literature study will be carried out to find any relevant information regarding baffle silencers. The study will include older established methods while also attempting to find more recent ones. Finding experimental measurements and manufacturer data, which will be used for comparisons, will also be a part of the study. The obtained methods will be tried out and compared to the experimental data gathered from the literature study. Upon finding the ones most suitable they will be highlighted in the thesis and written into a Matlab script. This script will function as a supplement to the methods found in the literature study, making multiple calculations easier. The script will be used to make a parameter study by making several computations with varying inputs, allowing conclusions to be made regarding the importance of the various parameters.

1.4 Delimitation

Air intakes, ventilation shafts and exhaust ducts in power plants can all have different shapes and different types of silencers installed in them. To keep the thesis to the point only one type of shape will be considered as well as one type of silencer, the parallel baffle silencer in a rectangular duct. Other types of silencers will be mentioned in the thesis, especially the lined duct silencer, but will not be a focus point; rather it will be used for discussing various theories and silencer implementations. Another limitation of this thesis is that only existing theories and methods will be used to make attenuation predictions. No attempts will be made to create new prediction models or theories as the effort required to do so would correspond to an entire master's thesis. Additionally acquired methods that require extensive numerical solutions will not be solved in the thesis. The limitations for the Matlab script are that no new information is used in it that is not present in the thesis.

2 BASIC ACOUSTICS INVOLVED WITH BAFFLE SILENCERS

While the majority of the methods here are relatively easy to understand, only some basic understanding of acoustic terms and glossary is needed. As such this chapter will explain some of the terms and expressions used for the calculation methods.

2.1 Wavelength and speed of sound

The wavelength of an incoming sound depends on the frequency of the sound as well as the speed of the sound in the medium it is propagating in. The wavelength is very important when damping and attenuation of sound is considered as the effectiveness of a silencer can be directly related to its size relative the size of wavelength of the incoming sound. The wavelength is expressed by the following equation:

$$\lambda=\frac{c}{f} \tag{2.1}$$

Where:	c=Speed of sound	[m/s]
	f=Frequency of the sound	[Hz]
	λ=Wavelength	[m]

The speed of sound itself is dependent on a few factors such as temperature and medium density. While the speed of sound is 343 m/s in air, different gases will result in different velocities. The speed of sound in an ideal gas is:

$$c = \sqrt{\gamma RT} \tag{2.2}$$

Where:	γ=Adiabatic constant	[-]
	R=Molar gas constant	$[\frac{J}{mol\ K}]$
	T=Temperature in Kelvin	[K]

2.2 Sound pressure and sound power

Sound pressure levels and sound power levels are two quantities normally used in acoustics. The sound pressure level is used to measure the pressure caused by the sound source, representing that which is actually heard. The sound power represents the acoustic power emanating from the source. In order to easily understand these parameters one need only to look at a regular light bulb. If one moves away from the bulb it will appear to be less bright than up close, despite the fact that the electrical power going into the bulb remaining the same. Sound power level is generally preferred since it does not change and has the distinct advantage over sound pressure levels which require detailed information about the measurements. Sound power cannot be heard however; in the same manner that one cannot observe the watt in a light bulb. The sound power is used to easier monitor insertion and transmission losses. The sound pressure level is expressed as:

$$L_p = 10\log_{10}\left(\frac{p}{p_0}\right)^2 \ re\ 20 * 10^{-6}\ dB \qquad (2.3)$$

Where: p =pressure of incoming sound wave in [Pa]
p_0=reference pressure corresponding to 20*10^{-6} [Pa[

While sound power is expressed as:

$$L_W = 10\log_{10}\left(\frac{P}{P_0}\right) \ re\ 10^{-12}\ dB \qquad (2.4)$$

Where: P=Sound power of source in [W]
P_0=Reference sound power corresponding to 10^{-6} [W]

2.3 Mufflers

Mufflers are a term commonly found in literature regarding silencers, along with absorbers it is simply a synonym. There are two types of mufflers that are used to attenuate sound, reactive and dissipative mufflers. Reactive mufflers reduce sound through resonances and anti-resonances determined by their geometrical shape. One or several expanded chambers along a duct can make up a reactive muffler where every chamber works to negate sound at a specific frequency. While a reactive muffler is very efficient for attenuation at specific frequencies it is not as efficient when it comes to reduction of sound with a wider frequency band. Sound at some frequencies may be reinforced in a reactive muffler if caution is not taken. Dissipative mufflers, on the other hand, are quite effective for wide frequency bands. Dissipative mufflers attenuate sound mainly through the presence of sound absorbing materials with a certain flow resistivity. These mufflers are the most useful for noise control where fan noise, flow induced noise and engine noise are of concern (Galaitsis & Vér, 1992).

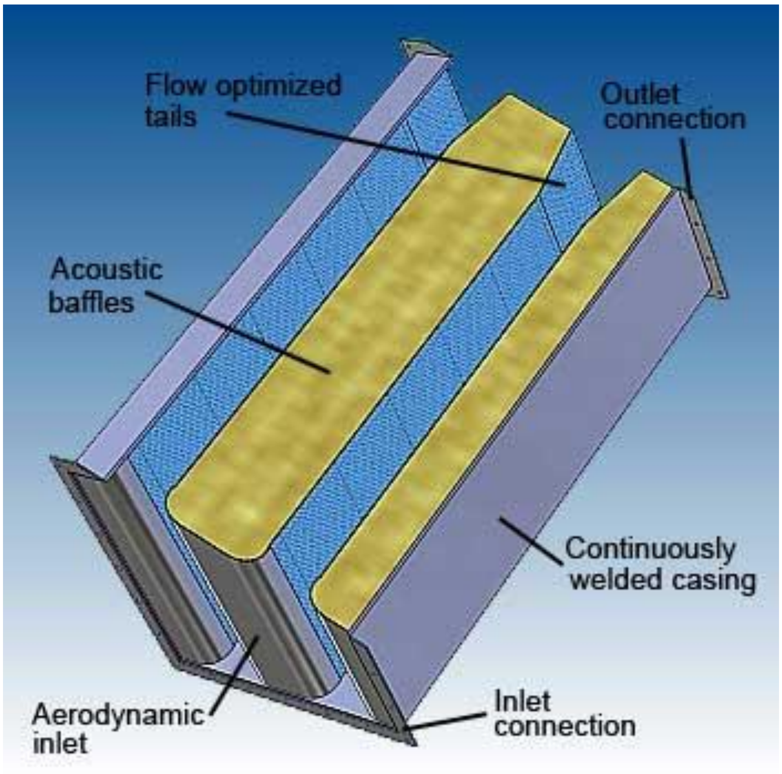

Figure 2.1: Basic appearance of a parallel baffle silencer (dB Noise reduction inc.)

One type of product that is similar to baffle absorbers is a lined duct absorber. Similar to the baffle absorbers it is fitted inside a duct and attenuates the sound in a similar manner to a baffle absorber.

Figure 2.2: Typical lined duct absorber

The attenuation mechanism inside a lined duct consists of the sound pressure from a sound wave in a duct causing a pumping action of the air going in and out of the porous material, converting acoustic energy into heat. Lined ducts function very similarly to parallel baffle absorbers and Embleton makes the assumption that one can consider a baffle absorber as a series of parallel line absorbers in a duct though one cannot simply replace baffle theory with lined duct theory (Embleton, 1971). More research has been devoted to lined ducts than to parallel baffle absorbers which make them a good entryway to finding more information about baffle silencers.

2.4 Insertion loss

Insertion loss is a term used to describe the difference in sound pressure level before and after incorporating a change to a given system, it is used to gauge the impact of sound reducing measures. In the image below the sound pressure level is changed when the duct is expanded with its impact being the insertion loss.

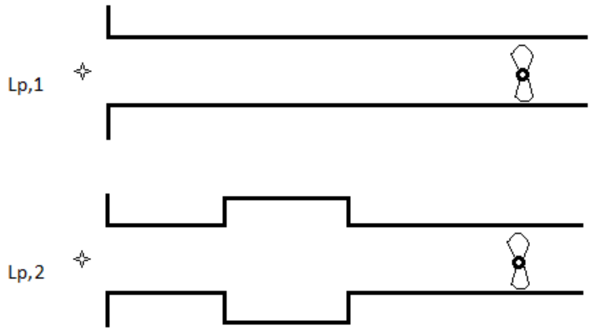

Figure 2.3: Insertion loss caused by changing the dimensions of the duct

$$IL = L_{p,1} - L_{p,2} \qquad (2.5)$$

2.5 Transmission loss

Transmission loss is the relation between the input power and the transmitted power from an absorber or element in a duct system. It is defined in the equation below. The transmission cannot be directly measured since sound power itself cannot be measured.

$$TL = 10\log_{10}\frac{W_i}{W_t} \tag{2.6}$$

Transmission losses and insertion losses are both used when describing sound reducing measures though the transmission loss is more common and is related more to the sound power which is generally better to use. Transmission losses from several sources can sometimes be superposed together in order to receive the total transmission loss which is often the thing that one is looking for.

2.6 Flow resistance and flow resistivity

Flow resistivity is a measure of the material's resistance to being penetrated by a sound wave (Ray, 2010). Low values of flow resistivity mean that a sound wave will easily penetrate a material which in turn will increase attenuation through the material. Very low values of flow resistivity are good for attenuating high frequencies while very low values are good for attenuating lower frequencies, meaning that there are trade-offs when choosing baffle material. Flow resistivity is sensitive to changes in gas viscosity, making it easily affected by changes to operating temperatures. Although it is not a real acoustic quantity it is used extensively in the source material for this report as a quantity that determines the properties of the damping material in the silencers.

Flow resistance is not the same parameter as flow resistivity. Resistivity is linked to the properties of the material while flow resistance is a specific parameter for a certain configuration of baffles with a specific flow resistivity.

2.7 Pressure drop

A pressure drop is defined as a difference in pressure between two points in a flowing fluid. The drop in pressure is usually caused by friction originating from the gas in the duct coming in contact with the duct walls. The roughness of these walls determine the amount of friction between the gas and the wall thus also determining the amount of friction, however introducing elements into the duct such a silencer has much higher overall effect on the pressure drop. Pressure drops create problems in that the reduced pressure creates a need for higher flow speeds. If the pressure drop is high and the flow speed not high enough the fluid in the duct will not be properly carried to its destination, whether it is to a chamber or the outside. Higher flow speeds require more power to fans or other flow inducing equipment which may not be economically feasible. The pressure drops should be sufficiently low so that the losses in pressure do not influence the emanation of the fluids. It should be noted that there is always some pressure drop in any given duct and that there are no measures that can be taken to completely remove the pressure drop.

3 LITERATURE STUDY

Any muffler or absorber to be used in a duct system should fulfill certain criteria in order to be considered properly designed (Embleton, 1971). These criteria can typically be condensed down to five separate goals that have to be met in one or another way. The criteria are:

1. Acoustic: The silencer must be able to reduce incoming sound by the desired amount. Expressed as the noise reduction with respect to frequency. The environment in which the silencer will eventually operate must be known since the acoustical performance is greatly affected by temperature, gas composition, alternating flow velocities as well as large steady flow velocities.
2. Aerodynamic: The silencer must not create a needlessly large pressure drop in the duct while operating
3. Geometrical: The silencer must fit in the space that is available
4. Mechanical: The durability of the silencer must be sufficient so that it can handle high temperatures as well as exotic gases which may erode less durable materials. Additionally the absorber should not require an excessive amount of maintenance in order to be fully efficient for long periods of time
5. Economical: An absorber should not be needlessly expensive since economic margins are a high priority for many manufacturers. Due to this a silencer should not be bigger than it has to be. Additionally the silencer should not absorb too much so that it is not cost inefficient

These criteria are often conflicting with each other, it is therefore important to find a method in which all of the criteria can be fulfilled to a satisfactory degree. The geometrical and the mechanical criteria are oftentimes something that has to be met in advance before any other criteria can be fulfilled. The operating conditions and the geometry of the duct have to be taken into consideration first and foremost before the size and thickness of the baffles can be predicted. These two criteria are omitted from the literature study since they are generally specified by a client and there are no calculations done surrounding them, with the exception for making sure that a silencer fits inside the duct and that the absorptive material can handle the operating conditions. Fulfillment of the acoustic criterion is the focus of this literature study, the prediction methods and theories regarding parallel baffle silencers have been searched for with this criterion in mind. The aerodynamic and economic criteria are somewhat connected and will not be explored fully. Should the aerodynamic criterion not be fulfilled then the economic one will typically not be fulfilled either. The same does not necessarily apply if the opposite is true. The literature found will be presented in order of relevance.

3.1 Embleton

Mufflers; in Noise and Vibration control

This book by Beraneck features several chapters written by other authors, one amongst which is Embleton with a chapter devoted almost exclusively to dissipative absorbers and ducts. Parallel baffle silencers are explained as being as special type of lined duct with each passage between the baffles being a lined duct of its own, essentially meaning that a parallel baffle silencer can be viewed as a series of lined ducts. Transmission losses in lined ducts come from the sound being absorbed in the porous lining as it travels down the duct. When the cross sectional area of the lining becomes as large as the open area through which sound travels, waves reaching the end of the lining will start to reflect back which will increase losses in addition to the losses from the lining itself. At this point the muffler acts like a reactive silencer as well. For sound traveling along the longitudinal axis of the duct the attenuation can be calculated. An assumption is made that only plane waves are traveling inside the duct. Even though true plane waves cannot exist inside ducts due to losses at the boundaries this makes calculations easier. Sound traveling through ducts with lining can generally be assumed to only consist of lower modes since these will “survive” the longest; higher modes quickly dissipate in the lining. These simplifications result in the prediction method giving conservative results for the total attenuation. Embleton presents a graph that can be used to estimate the total attenuation of a parallel baffle absorber with only a few parameters given. The parameters needed are the baffle width, the spacing between the baffles, their length and the wavelength of sound.

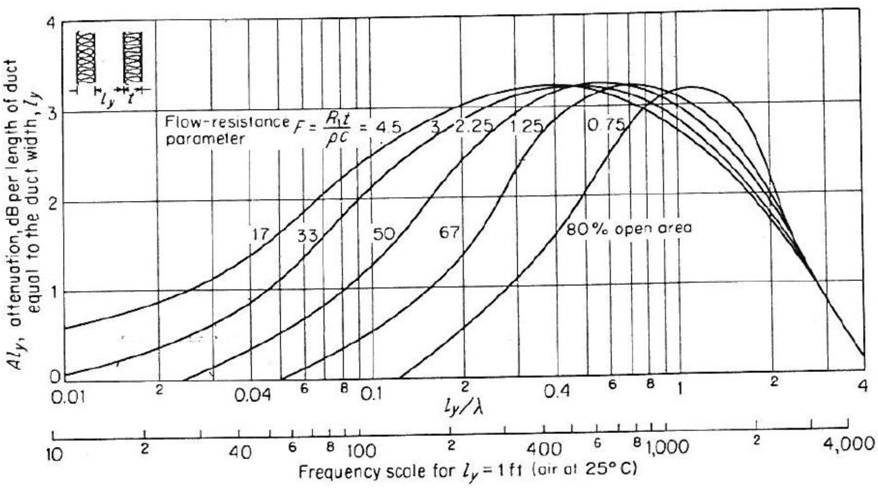

Figure 3.1: Graph used to predict attenuation values

Using this graph gives values for the attenuation per length of duct equal to duct width, which can then be converted to total attenuation for the entire length of the baffle silencer. The use and interpretation of this graph is further explained in chapter 4. It should be noted that use of this graph assumes that there is no flow in the duct and that the frequency scale shall be used under the assumption of air at 25 °C though small variations make little difference. The graph can be used for any gas with known speed of sound and temperature. The attenuation curves seen in the graph are apparently not sensitive to changes in flow resistivity as it makes little to no difference for values ranging from twice to one half of normal values according to Embleton. The theory behind these curves is referenced to Cremer 1953 and curves by Ingard. The Ingard reference refers in turn to Lukasik and Nolle 1955, which is described below. As is seen in the graph the attenuation starts to rapidly drop at higher frequencies. Embleton explains this as being caused by a “beaming effect” as high frequency sound simply passes in between the baffles, avoiding the absorptive material. Embleton proposes using silencers with a staggered baffle arrangement, eliminating straight paths through the baffles as seen in Figure 3.2. The accompanying attenuation curves show that this kind of configuration would be very effective as they maintain good attenuation also at higher frequencies. Unfortunately there are no instructions as to how the attenuation for these kind of silencers can be predicted as Embleton makes no further mention about them.

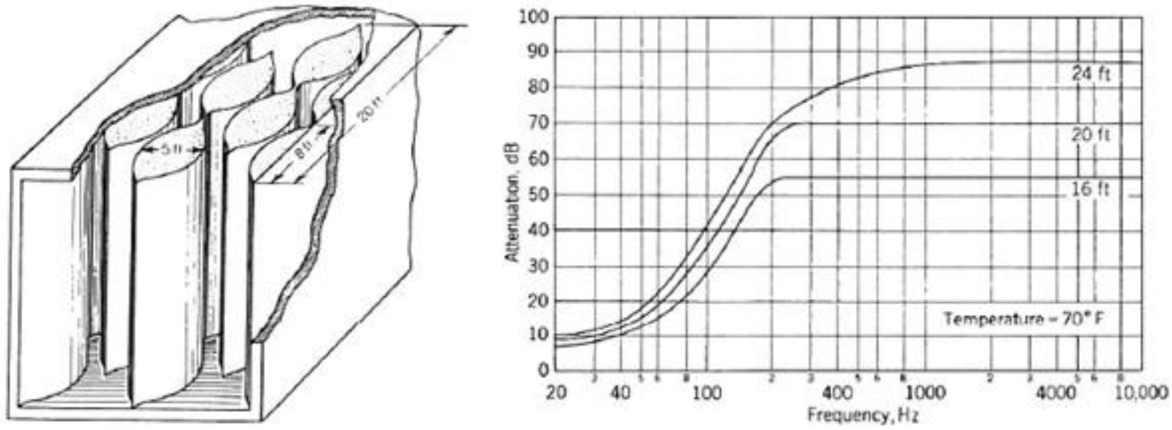

Figure 3.2: Proposed configuration and typical attenuation behavior

Also seen in Figure .1 is that attenuation is low for very low frequencies. The prediction method used for baffle absorbers loses accuracy for very low frequencies. When $l_y/\ \lambda<0{,}1$ the prediction method is no longer usable as the results become unreliable. For these frequencies Embleton proposes using a different method, using an equation by Sabine who empirically found that attenuation in ducts for low frequencies can be expressed as:

$$A' = 12.6(\bar{\alpha}_{Sab})^{1.4} * \frac{P}{S} \tag{3.1}$$

Where:

A'= Attenuation	[dB/ft]
$\bar{\alpha}_{Sab}$= Sound absorption for the duct lining	[-]
P= Acoustically lined parameter of duct	[in.]
S= Cross-sectional open area of duct	[in.^{2}]

This equation is limited to low frequencies as it becomes increasingly inaccurate as $l_y/\ \lambda$ becomes larger, its use must be restricted to $l_y/\ \lambda<0{,}1$. Transformed to SI-units the equation becomes:

$$A' = 12.6(\bar{\alpha}_{Sab})^{1.4} * \frac{P}{39{,}37 * S} \tag{3.2}$$

Where:

A'= Attenuation	[dB/m]
$\bar{\alpha}_{Sab}$= Sound absorption for the duct lining	[-]
P= Acoustically lined parameter of duct	[m]
S= Cross-sectional open area of duct	[m^2]

Operating conditions for ducts used as exhausts include a certain gas flow. This affects the total attenuation in one of two ways, depending on the direction of the flow. When gas flows in the same direction as the sound waves the resulting attenuation will be lowered due to the sound having less time to be absorbed by the baffles. When the direction of the flow is in the opposite direction the total attenuation is instead increased. Embleton presents an equation for how the effect of flow can be estimated given the speed of the flowing gas.

$$Attenuation = A\left[\frac{1+\gamma|M|}{1+M}\right] \tag{3.3}$$

Where:

A= Attenuation without flow	[dB]
M= Mach number	[-]
γ= Experimental factor	[-]

The Mach number is the relation between the speed of the flow and the speed of sound. The experimental factor is not explained and features no further explanation other than being dependant on “many factors”. The attenuation equation, and its use, is further examined in chapter 4.

3.2 Galaitsis & Vér

Passive silencers; in Noise and Vibration control engineering: Principles and applications

In Beranek’s *Noise and vibration control engineering: Principles and applications* from 1992 there is a chapter written by Anthony Galaitsis and István Vér that covers dissipative silencers and also has parts exclusively focused on parallel baffle absorbers. Galaitsis and Vér cover baffle absorbers rather extensively in the same manner that Embleton did though they have a more comprehensive collection of information. One new addition to this version is the entrance and exit losses of the parallel baffles. These losses, while small, still contribute to the overall attenuation of the baffle configuration and could be considered as a safety margin of sorts. The entrance losses depend on the cross dimensions of the duct and the wavelength of the incident sound wave. If the cross dimensions are much larger than the wavelength the sound wave will typically consist of higher order modes. The conversion from a semi diffuse sound field to a plane wave in the narrow passages typically results in attenuation of 3-6dB. It is stated that experienced engineers with a lot of knowledge in the field can approximate the losses from 0 dB for low frequencies up to 8 dB for higher frequencies. This is not recommended for inexperienced engineers and a figure describing the behavior of entrance losses depending on wavelength and baffle spacing is provided.

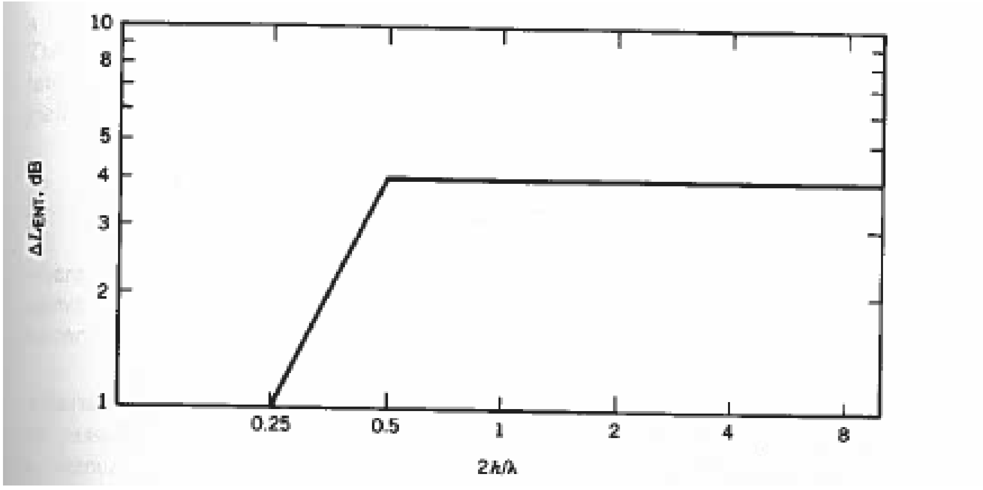

Figure 3.3: Curve describing entrance losses

The exit losses are caused by the end reflections that occur at the end of the baffle configuration and are usually very small. They can either be ignored or considered to be a part of the safety margin mentioned above. It should be noted that the importance of the entrance and exit losses becomes increasingly less important as the length of the baffles increases and can, once again, be viewed as a part of the safety margin.

Galaitsis and Vér have a different approach to describing the reduction caused by the parallel baffles than the one used by Embleton. Here the total reduction is described using the cross sectional area of passage, the total lined perimeter, silencer length and a parameter that is dependent on geometry, acoustic characteristics of the absorbing material, frequency, temperature as well as flow speed. It is stated that this parameter is usually referred to as attenuation per channel width, meaning that it is probably similar to the one that Embleton uses. Despite presenting this equation Galaitsis and Vér do not use it in their attenuation predictions, forgoing it in favor of a more graphical solution. The method that they propose is dependent on using normalized graphs for various flow resistances and percentages of open area. These graphs are similar to the one used by Embleton though each one is specific for a certain percentage of open area and features several curves, each representing a values for the normalized flow resistivity. The presented graphs can be seen in appendix B and their usage for predicting silencer attenuation is featured in chapter 4. Apparent in all the graphs is that despite the various curves all having different flow resistivity values the reduction values only differ slightly. This is something that Galaitsis and Vér comment on themselves, stating that a flow resistivity twice or half the normal value makes no significant difference on the total attenuation and that larger variations only result in minor changes. This is fortunate as lack of knowledge and control over material properties is currently the weakest link in the prediction process.

An important aspect of baffle sizing is the thickness of the baffles themselves. Despite having the same percentage of open area two baffle configurations can have different attenuation values depending on the thickness of the baffles. This can be shown by comparing baffles with the same length but different thickness. Figures 3.4 and 3.5 illustrate this. In the first graph all the baffle thicknesses have a fibrous material with the same value for flow resistivity. This implies that the total flow resistance increases with baffle thickness. The baffles from the second are instead chosen so that their total normalized flow resistance is the same regardless of their thickness. The total flow resistance for the baffles in the second graph is also lower than for the baffles in the first graph.

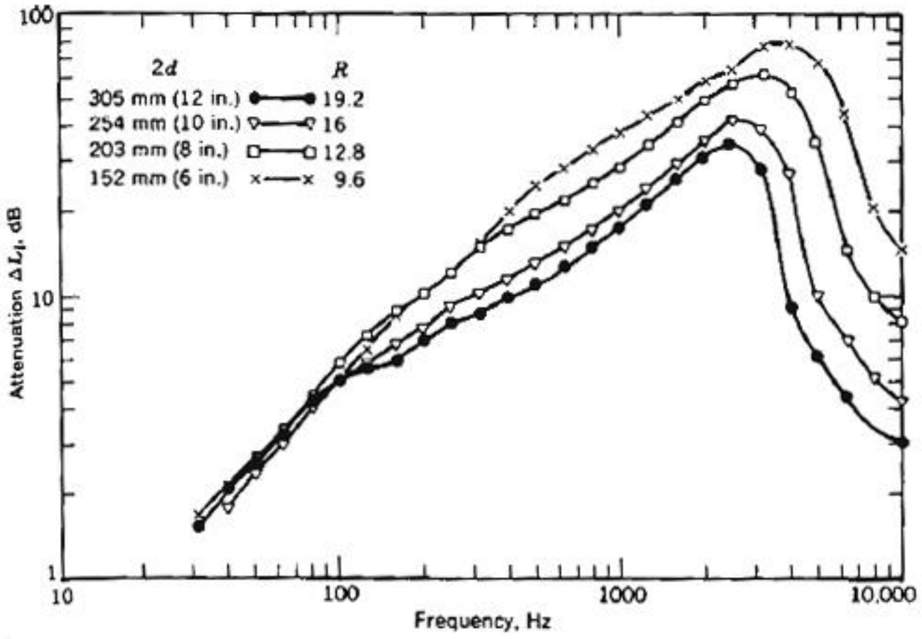

Figure 3.4: Total attenuation for several baffles with varying baffle width but same percentage of open area

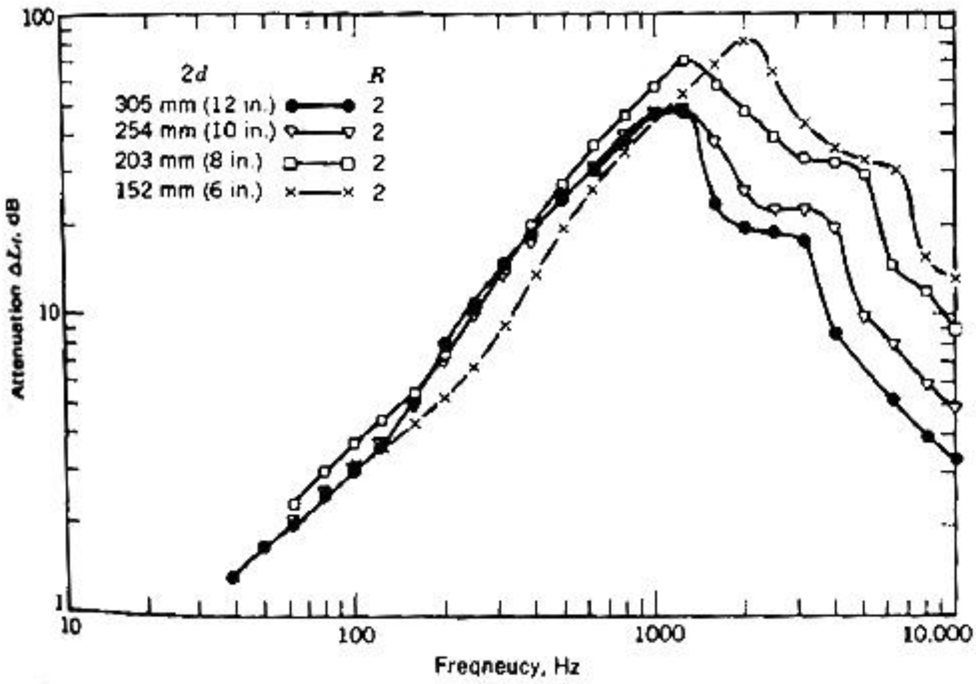

Figure 3.5: Attenuation for several baffles with the same flow resistance parameter

What these graphs show is that a higher quantity of thinner baffles result in greater attenuation for the mid frequencies than a few thick ones, despite having the same percentage of open area. The lower frequencies are mostly unaffected by the thickness of the baffles. Thinner baffles are more expensive than thick ones which could lead to a large increase in cost if a lot of baffles are required. As the second graph shows the thickness is actually not an important factor for the mid frequency attenuation as long as the baffles are filled with material that has sufficiently low flow resistivity. Galaitsis and Vér make a point however that those materials fulfilling the requirements for flow resistivity while also being capable of withstanding the usually high operating temperatures may not be readily available. In general baffles which are designed for very high temperatures are seldom thicker than 200 mm.

Temperature affects more than just the choice of material. When the temperature rises it changes the density and the viscosity of gas. This leads to changes in the speed of sound as well as affecting the flow resistivity. Galaitsis and Vér present equations for how this can be calculated as well as presenting a graph that illustrates the basic principle of temperature affecting the attenuation of a baffle silencer.

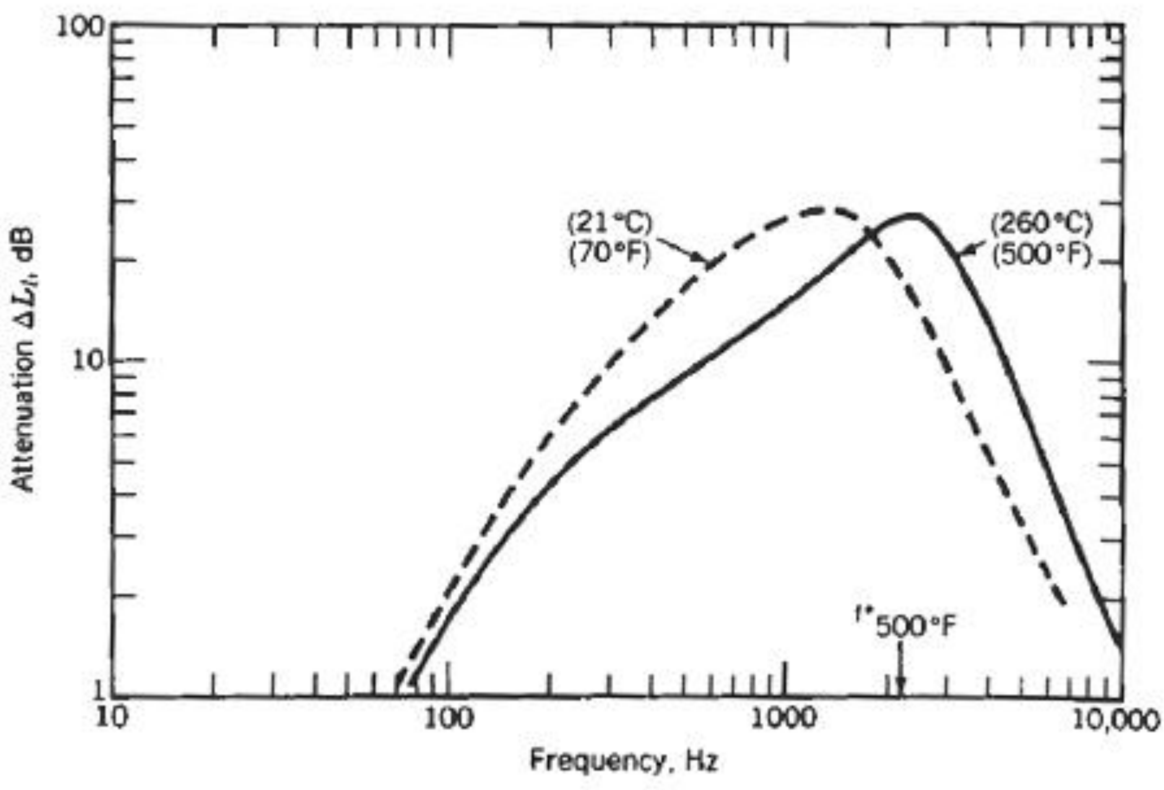

Figure 3.6: Influence of temperature on attenuation

The change in temperature does not affect the peak attenuation at all, merely shifting the entire curve instead. The small distortion of the curve is due to the change in flow resistivity. These equations are described in more detail in chapter 4.

The effect of flow is another aspect of muffler design that is brought up by Galaitsis and Vér. Flow affects sound by changing the propagation speed of sound in the duct and also changes the flow resistivity somewhat. Galaitsis and Vér present methods that can be used to estimate the effect of flow on attenuation. This estimation is based on using attenuation curves for no flow and shifting them in the appropriate direction. The amount of information is somewhat lacking however, as can be seen in chapter 4 where they are evaluated further.

3.3 Almgren

Attenuation in sound absorber Parallel baffle

In a report for the former Svenska Fläktfabriken, now Fläkt Woods, Almgren investigates various methods for evaluating the attenuation of parallel baffle silencers. Almgren covers the theory of calculating reduction for parallel baffle absorbers, using Embleton's chapter from Beranek's *Noise and vibration Control* while also including some additional equations. The final pages of the report are dedicated to compare the theoretical reduction values with measurements which makes are a valuable reference. Additionally equations on how to calculate the pressure drop in a duct as well as how to adjust the total attenuation of a baffle absorber affected by a known flow are also included.

Investigation of Calculation manual for Baffle silencers

In this report Almgren makes several comparisons between theory and measured results for a, then, commercially available parallel baffle silencer. Comparing several different theories for calculating attenuation, including Embleton and Cremer, Almgren concludes that the method specified by Embleton is the most accurate one. Presented in the report are several charts where different baffle configurations are compared to their measured counterparts, with varying parameters on baffle spacing, length and width. A few of the charts as well as his comparisons between measurements and various theories are available in the appendix section of this report.

While there are a few more reports done by Almgren for Svenska Fläktfabriken they do not cover parallel baffle absorbers themselves and are not directly relevant.

3.4 Mechel

Formulas of Acoustics

A very miniscule part of the book is dedicated to parallel baffle silencers and the information that is given is rather complicated, focusing on eigen-equations and complex wave numbers. Though an equation describing the transmission loss through a parallel baffle absorber is shortly described the equation requires the use of additional calculations from earlier parts of the book. Whether this equation provides good results or not is not known since it required to much effort to be used for any meaningful conclusions to be drawn.

A later edition of the book also features a short segment about baffle absorbers though the same problem exists here as well. The equations pose the same problem as the ones from the previous book, being intricate and requiring extensive effort to fully utilize. Thus their usefulness is not explored here.

Theory of baffle type silencers

In this text Mechel provides equations for use in estimating transmission loss through parallel baffle silencers. The equations are developed using the wave-equation as well as certain boundary conditions. Mechel presents these equations but does not provide any numerical solutions to them. The equations themselves seem to not be dependent on any kind of flow, temperature or flow resistivity, focusing on the sound field as it propagates through the baffles. Additionally these equations seem to only provide a value for the total transmission loss for the entire silencer with no clear method of seeing the attenuation for certain frequencies. These equations by Mechel seem to provide no method of predicting the attenuation for parallel baffle silencers, focusing instead on how sound behaves around them.

Numerical results to theories of baffle type silencers

This text provides numerical solutions to the previously presented equations. Comparison between measured insertion loss and total transmission loss shows good agreement. The computed insertion loss is divided into propagation loss, front side reflection loss and back side reflection, which illustrates their importance, seen in Figure 3.7. Despite featuring numerical solutions the equations are still not suitable for predicting silencer attenuation. Another important aspect of these equations is that there is no specification regarding the scope of the baffle silencers in the theory. There is no mention whether or not these equations can be used for baffle silencers in power plant exhausts with high temperatures and occasionally large volume flows.

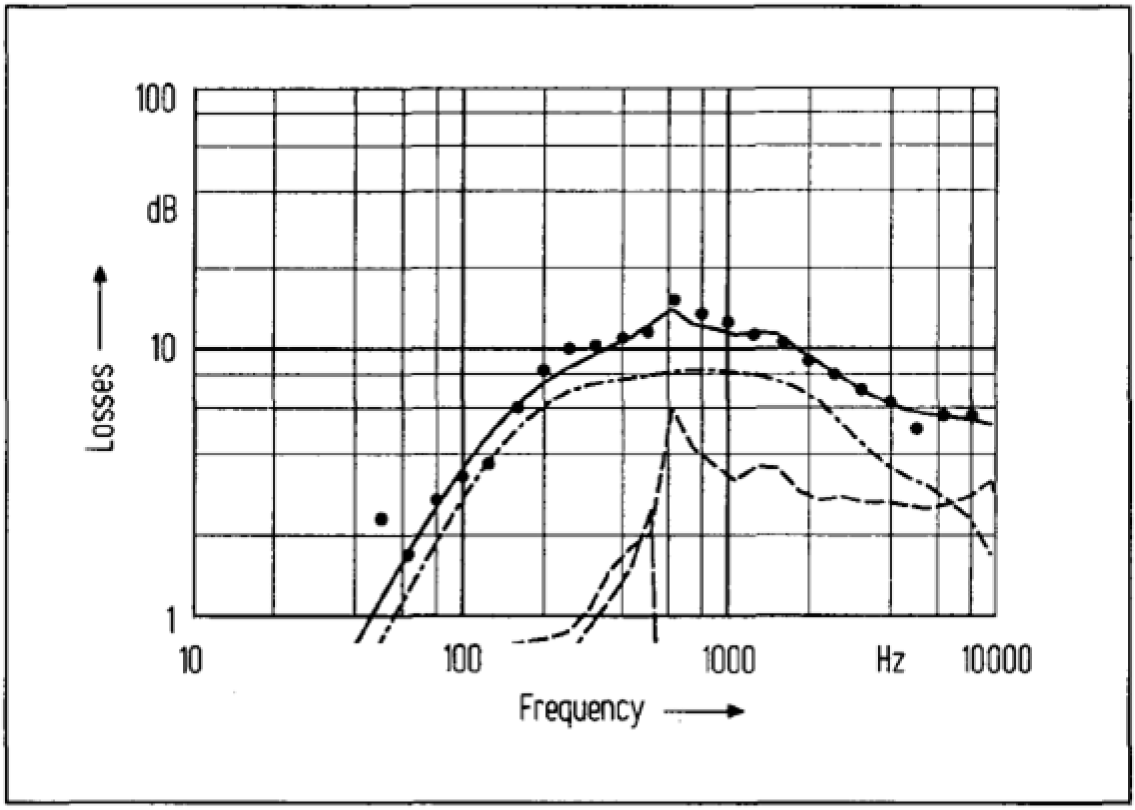

Fig. 7. Losses of a short-baffle-type silencer; points: experiment; curves: computed, continuous: total transmission loss, dash-dotted: propagation loss, dashed: front side reflection loss, long dash: back side reflection loss. $T = 0.5$ m; $H = 0.24$ m; $D = 0.36$ m; $\Xi = 11$ kPa s/m^2; $Z_{ser}/Z_0 = 1.0$; $m'' = 0.2$ kg/m^2.

Figure 3.7: Various losses for a baffle silencer as shown by Mechel. Includes experimental and computed values as well as front and back side losses

3.5 Munjal

Acousics of ducts and mufflers with application to exhausts and ventilation system design

Munjal briefly covers baffle silencers this book. The chapter features very little information on how to calculate attenuation of parallel baffles and instead mostly focuses on the wave equation. When discussing parallel baffles he references multiple times to attenuation values calculated by Vér, which are the same ones that Galaitsis and Vér later used in Beraneks book. One unique feature of this chapter is a reference to a prediction model for flow induced noise made by Vér. Munjal presents an empirical formula for calculating the sound power level caused by introducing a parallel baffle in a duct. The flow induced noise must be taken into consideration when calculating the insertion loss of the parallel baffle. Munjal makes references to empirical schemes on flow-induced noise made by Vér in the issue of Inter noise 1972. This book has not been available in the databases and the libraries used for this thesis meaning that there will be a lack of comparison material for flow induced noise.

Analysis and design of pod silencers

This short report, while mainly focusing on circular pod silencers, briefly touches on the subject of parallel baffle silencers. The insights given in the introduction chapter mostly serves as a basis for the pod silencers.

3.6 Ray

Absorptive silencer design

Elden Ray gathers simplified calculations and a lot of the existing theories surrounding baffle silencers in this short report, using examples to showcase the calculation process. The calculation process is the same one used by Galaitsis and Vér, using the same graphs in the attenuation calculations. One addition that Ray adds is an equation for adjusting the attenuation rate for a given flow. The equation is taken from Bies and Hansen and has the criteria $-0,3<M<0,3$. More information about this equation and its implementation can be found in chapter 4. Ray writes about the influence of self induced noise, stating that it is simply reduced from the total reduction. There are no calculations regarding the self induced noise however and Ray instead refers to spreadsheets in published catalogs. Additionally the subject of pressure drops is not covered at all in this text.

3.7 Ingard

Notes on Duct Acoustics N4; Notes on Sound Absorption technology

This book is a part of a bigger series by Uno Ingard made to be a theoretical groundwork for a computer program that is accompanied with the books. This program has many uses, amongst which are attenuation prediction, transmission loss and insertion loss for lined ducts. These programs are made to use DOS and have some issues with more modern operating systems. The content of this particular book, N4, is of interest due to its lined ducts chapter. It should be noted that Ingard himself does not bring up parallel baffle silencers at all, focusing entirely on lined ducts. As mentioned earlier parallel baffles can be considered a special case of lined ducts and as such the information here is still relevant. Ingard makes a distinction between two types of lining which behave differently, locally reacting and non-locally reacting linings. Parallel baffle absorbers can be considered as non-locally reacting linings. Ingard mentions that locally reacting ducts need to be separated by rigid partitions that are closely spaced between each other. Due to being practically difficult to implement lined ducts are rarely considered to be locally reacting in reality. The book contains no prediction methods or any other theories that can be used simply straightforward to predict attenuation by one self, instead referring to the associated software where the equations have been implemented. The book instead features theory and observations regarding lined ducts. One such observation is how the open area affects the attenuation spectra. In four different graphs seen in Figure 3.8

Ingard shows the attenuation spectra for four different percentages of open area as well as different flow resistivity for a non-locally reacting liner.

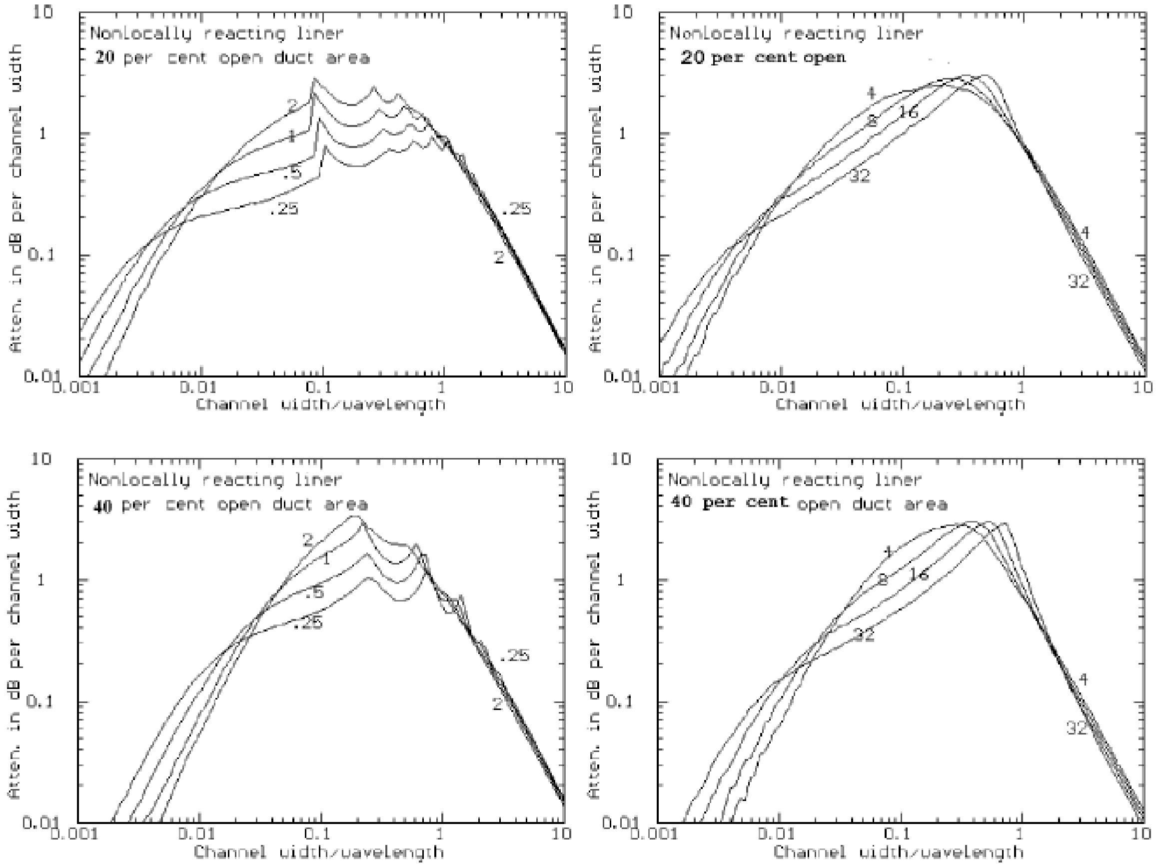

Figure 3.8: Influence of open area on attenuation curves for non-locally reacting liners

The graphs show how the spectra of attenuation become smaller for higher percentages of open area; additionally the peak attenuation becomes higher with more open areas. There are no graphs for percentages above 40% for non-locally reacting liner though for locally reacting ones there are graphs up to 70%. The curves in these graphs behave in very much the same way albeit with some differences in detail.

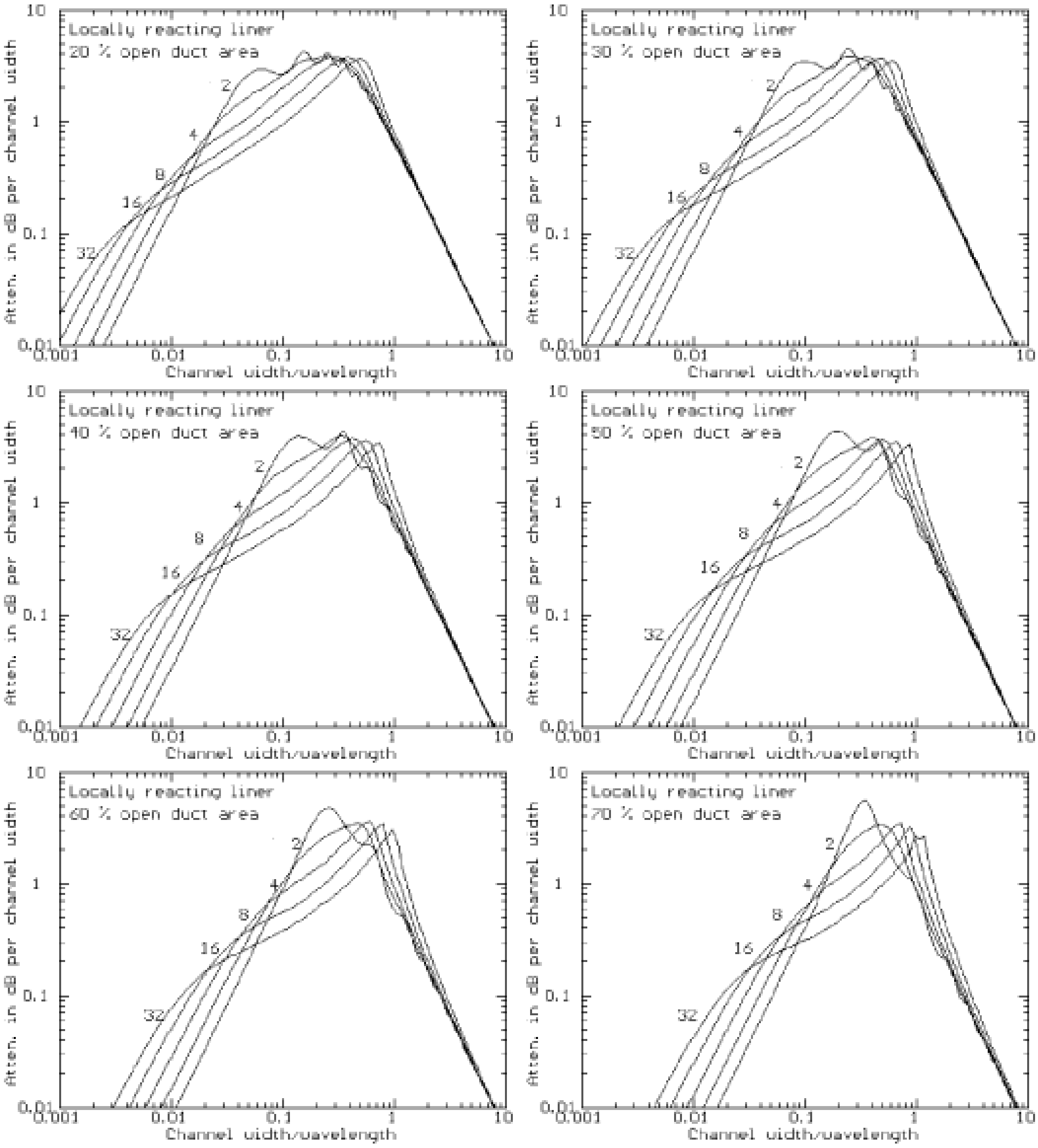

Figure 3.91: Influence of open area on locally reacting liners

These graphs are closely tied to another subject, pressure drop. Ingard states that the open area should not be reduced needlessly just to obtain broader attenuation spectra since this would lead to too high pressure drops. These are not permissible since whatever fan powers the flow in the duct would have to compensate for the energy lost through the pressure drop. One application exists for having minimal open area percentage, the porous plug. A porous plug is simply a duct with absorptive material covering the entire inner area of the duct. This can be used for noise with very high frequencies though it is generally not recommended. Partially due to the pressure drop typically being unreasonably high where porous plugs are used and partially because the noise that the plugs are effective against ranges in frequencies above that of human threshold of hearing. There are no specified prediction methods available for this part of the book either as Ingard once again refers to the associated software. The insights given by this book do little to expand on the theories already discovered though it does present interesting observations that are good to have in mind. The equations presented in the text offer a good basis upon which newer theories could be developed however these would need to be solved for any such result.

3.8 Ochmann & Donner

Investigation of silencers with asymmetrical lining. I: Theory

This report looks to create a theoretical basis for calculating the transmission losses in a lined duct. The most interesting parts of this report are the theories for lined ducts with different flow resistivity on the different linings. As is mentioned in the report these theories could be applied to parallel baffle silencers as well, making this a very valuable source. The theories presented are based on rather complex differential equations that would take quite some time to implement and calculate. Ochmann and Donner state that the solutions to these theories will be presented in the second part of this paper. Unfortunately there exists no second part with no explanation or reason why, making it difficult to use any of these theories without solving the equations one self.

3.9 Cremer

Theorie derLuftshall-dämpfung im Rechteckkanal mit schluckender wand und das sich dabei ergebende höchste Dämpfungsmass

Quite a few of the authors in this literature study have used this work by Cremer as reference, using his theories to calculate attenuation in ducts. The entire text is in German however and was not translated for the purpose of this report, meaning that its content will not be featured in later chapters. Despite being well referenced in other literary works on baffle silencers it is safe to assume that more recent literature has covered everything that Cremer did in this article and that the more recent authors have had the possibility to expand upon his theories. Thus Cremer’s article would only serve as to identify the origin of the theories used rather than expand upon current knowledge in the field, especially considering the age of this literary work.

3.10 Bell

Industrial Noise Control: Fundamentals and Applications

Parallel baffle absorber are covered quite extensively in this book though there are no given methods of calculating reduction values nor are there any references to other literature. The majority of the chapter on baffle absorbers is devoted to analyzing manufacturer data and explaining the usage of absorbers in ducts. The chapter also covers the topics of the influence of flow on reduction values, flow generated self-noise and pressure drop. These chapters also lack any form of theoretical background and are based solely on manufacturer measurement data. Tables are featured with the influence of flow velocity on attenuation however there are no details about the sizing of the baffles, only their length. This makes it impossible to directly compare the data in the book to the theories developed by other authors.

3.11 Kirby

Simplified techniques for predicting the transmission loss of a circular dissipative silencer

In this report for the Journal of Sound and Vibration Kirby presents equations for predicting the transmission loss of a circular silencer. His work focuses primarily on circular silencers and does not mention parallel baffles. Moreover he almost exclusively uses eigenequations together with rather intricate calculations, making it difficult to use his work. While he does perform experiments in order to validate his equations and manages to get good agreement with theory the equations are not easy to understand or use, hence they will not be featured in further chapters.

3.12 Lukasik & Nolle

Handbook of acoustic noise control

Written as a part of a bigger series, this report was commissioned by the U.S. Air force in 1955 to look at how noise could be reduced in various facilities used by air force personnel. One chapter is devoted to lined ducts which makes this text interesting for the purpose of this thesis. The authors mainly work with the wave equation with certain boundary conditions in order to calculate the attenuation. While there is no mention of parallel baffles lined ducts can be used as an approximation as is already established. The authors, Lukasik and Nolle, provide charts which can be used to approximate the attenuation in a lined duct given size of the duct and the lining as well as the flow resistivity. These charts are identical to the ones used by Embleton in *Noise and Vibration control* which is of little surprise as the authors have used a report from Beranek as basis for their calculations. This provides little new information as a lot more has happened in the field of duct acoustics since 1955. Given the fact that more recent work has had access to better calculations through computers this text can be considered obsolete.

3.13 Su

Acoustics of parallel baffles muffler with Micro perforated panels

Master thesis where Su performs measurements on parallel baffle silencers covered with micro-perforated panels. These baffles do not consist of any type of absorptive material but are hollow with only the perforated panels making up the baffles. This would not be a problem for making comparisons as these panels themselves would inherently have a value for flow resistance attached to them. The measurements in the thesis are not suitable for comparison with prediction models due to all results being expressed as transmission loss with respect to frequency. The measurements are performed by having the baffles placed between an anechoic chamber and a reverberation room, measuring the intensity and sound pressure in both rooms before and after the sound source is turned on. There is however no information about the level of sound pressure or the intensity, making the measurements unusable for comparison.

3.14 Verein Deutscher Ingenieure

VDI 3733: Geräusche bei Roheleitungen Noise at pipes

This document is a part of several from VDI, a German association of engineers that makes guidelines for various disciplines within engineering. In this particular report guidelines for predicting noise in pipes are presented. While the majority of the document focuses on pipes specifically and not ducts, as well as on other concepts not directly related to parallel baffle silencers there are a few mentions on how flow induced sound is dependent on flow speed and frequency. The image below shows how this behaves:

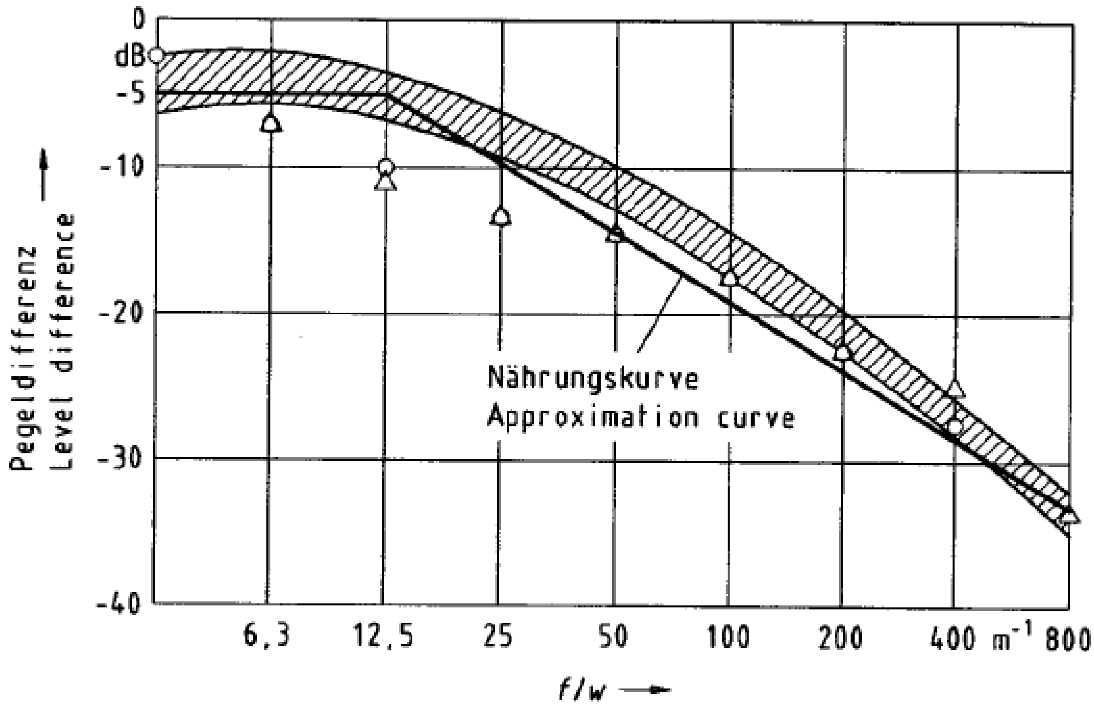

Figure 3.10: Change of flow induced sound with respect to frequency

While this graph is used specifically for pipes and not ducts it is still useful in that it shows a relation between frequency, flow speed and sound level. Direct application of this graph is impossible but it can still be used to get an approximate idea of the influence of flow.

3.15 Conclusion

The most valuable literary sources from the literature study proved to be the two editions of the same book by Beranek, *Noise and Vibration control* from 1971 and *Noise and Vibration control engineering: Principles and Applications* from 1992. The chapters regarding parallel baffle silencers and the calculation methods for predicting their attenuation presented in these two books adequately presents the needed information to make attenuation predictions. Hence these two sources are best suited for later calculations regarding the predictions for parallel baffle silencers. The various texts by Almgren provided helpful data regarding comparisons between predicted attenuation and measured reduction values while authors such as Ray and

Munjal provide more credibility to the methods proposed by Galaitsis and Vér. Many of the other sources, the ones that did not provide any useable prediction models, are present to show the extent of the literature study. Additionally they showcase the general lack of newer material regarding predictions for parallel baffle silencers. Newer publications, from the 2000's and later, rarely present any new methods or prediction models. Instead older works are referenced, mostly Galaitsis and Vér's chapter.

4 CALCULATION METHODS AND THEORETICAL REDUCTION VALUES

The various calculation methods collected from the literature study are presented in this chapter and the theoretical reduction values produced from them are presented for various parallel baffle configurations. Many of the calculation methods that have been found are based either on the method by Embleton or the one by Galaitsis and Vér, therefore only these two will be presented. The examples presented in their respective sources assume no flow through the duct and normal room temperature, resulting in a normal speed of sound. This is not representative of the real operating conditions for parallel absorbers in power plants, as such additional equations for the influence of both temperature and flow will be included. As such, later examples will feature the influence of both temperature and flow.

Worth noting is that while other theories have been found, primarily from Munjal and Ochmann, they are overly complicated and have proven to be too much of a mathematical challenge to overcome and have thus been omitted from this chapter.

4.1 Embleton

Embleton's method of calculating the reduction is based on visually interpreting the graph in Figure so that the attenuation per duct width can be obtained.

The parameters needed to obtain the attenuation per duct width are the width of the baffles and the spacing between each baffle. These are used to calculate the percentage open area as well as the l_y/λ parameter, which is the ratio between the spacing and the wavelength and is used as the horizontal axis in the figure. It is obvious that with changes to sound speed the wavelength will also change, shifting the attenuation curves in the appropriate direction. As mentioned earlier the attenuation in a baffle absorber is not particularly sensitive to changes in flow resistance, it is thus generally ignored in Embleton's method.

Having obtained the open area percentage and the frequency axis it is simply a manner of following the correct curve of open area to obtain the attenuation per ducts width, A_{ly}, for each frequency. This is parameter is then multiplied with a length factor, L_{ly}, which depends on the duct spacing and the length of the duct. One important aspect that should be noted is that the baffle width used in the theories is in fact only half the width of the entire baffle. The reason for this is that the other half "belongs to the other baffle" as seen in Figure 4.***2***.

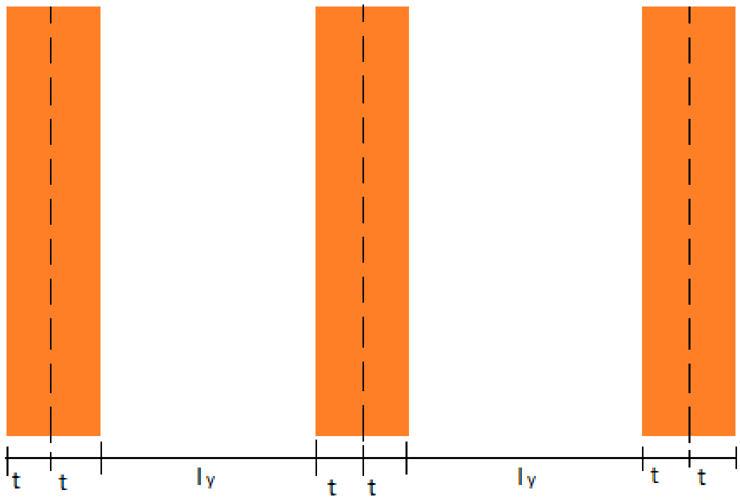

Figure 4.2: Parameters for baffle width and thickness

An example will further illustrate how this is done.

Example

A parallel baffle arrangement consists of baffles 100mm thick with a spacing of 100mm with a length of 1m. Assume room temperature and a low air flow i.e. c=340m/s.

The attenuation for this particular duct is calculated using these following steps:

1. Calculate the open area

$$l_y = 100mm = 0{,}1m$$

$$t = \frac{100}{2} = 50mm = 0{,}05$$

$$OA = \frac{l_y}{l_y+2t} = \frac{100}{100+2*50} = 50\%$$

The curve for 50% will be used

2. Calculate the ratio l_y/λ for the frequencies of interest

Frequency	100	200	500	1000	2000	4000
l_y/λ	0.029	0.058	0.147	0.294	0.588	1.176

3. Visually determine A_{ly} from the graph for the frequencies of interest

Frequency	100	200	500	1000	2000	4000
l_y/λ	0.029	0.058	0.147	0.294	0.588	1.176
A_{ly}	0.2	0.6	1.8	2.9	3.2	3

4. Multiply A_{ly} with the length factor L_{ly} to get A

L=1m

$$L_{l_y} = \frac{L}{l_y} = \frac{1}{0{,}1} = 10$$

Table 4.1: Final reduction values using Embleton's method

Frequency	100	200	500	1000	2000	4000
l_y/λ	0.029	0.058	0.147	0.294	0.588	1.176
A_{ly}	0.2	0.6	1.8	2.9	3.2	3
A	2	6	18	29	32	30

The last row shows the absolute attenuation for each frequency over the entire baffle silencer. This method is very quick and gives approximate values for the attenuation at the frequencies of interest. The downside of this is that the reduction values are not pinpoint accurate as they depend on how well one interprets the graph. Additionally the flow resistance is not taken into account here as the graph can be used for nominal values differing from one half to twice of that given, again making small variations in the end reduction values. Overall this method should be considered to give conservative values and should not be used to obtain exact values for attenuation.

4.2 Galaitsis &Vér

The method employed by Galaitsis and Vér is similar to the one used by Embleton as they all use preexisting graphs in order to calculate the attenuation of a baffle absorber in a duct. The method employed here is somewhat more cumbersome as it s based on the use of transparent paper with logarithmic scales, used to trace attenuation curves based on the parameters of the baffle absorber. The transparent paper is placed over the attenuation graphs provided in the literature and traced, giving the attenuation curve for the current configuration. Several steps are taken in order to accurately shift the attenuation curves. One advantage to this method is that it takes the flow resistance into account. The downside being that the attenuation graphs from the book are needed and that there are only graphs for a few specific open area percentages with no way of approximating curves for other percentages. Using the same example as before the steps taken to calculate the attenuation are done in the following manner:

<u>Example</u>

A parallel baffle arrangement consists of baffles 100mm thick with a spacing of 100mm with a length of 1m. The flow resistance of the baffles is R1t=5ρc. Assume room temperature and a low air flow i.e. c=343m/s.

1. <u>Calculate *f'*</u>

   $$f' = \frac{c}{l_y} = \frac{340}{0{,}1} = 3400\ Hz$$

2. <u>Determine</u> $\frac{L}{l_y/_2}$

   $$\frac{L}{l_y/_2} = \frac{1}{0{,}05} = 20$$

3. Determine open area and choose appropriate graph

   $OA = \frac{l_y}{l_y+2t} = \frac{100}{100+2*50} = 50\%$

   For OA=50% the graph from appendix B2 should be taken

4. Determine *R* and choose appropriate in the graph

   $R = \frac{R_1 * t}{\rho c} => R = 5$

   Corresponds to solid curve

5. Mark the frequency *f'*=3400 on the transparent paper and align it with *η*=1 in the graph

6. Shift the transparent paper vertically until L_h=1 on the graph corresponds to $\frac{L}{l_y/_2} = 20$ on the transparent overlay

7. Copy the solid line from the graph onto the transparent paper, the resulting line is the attenuation for the entire baffle configuration

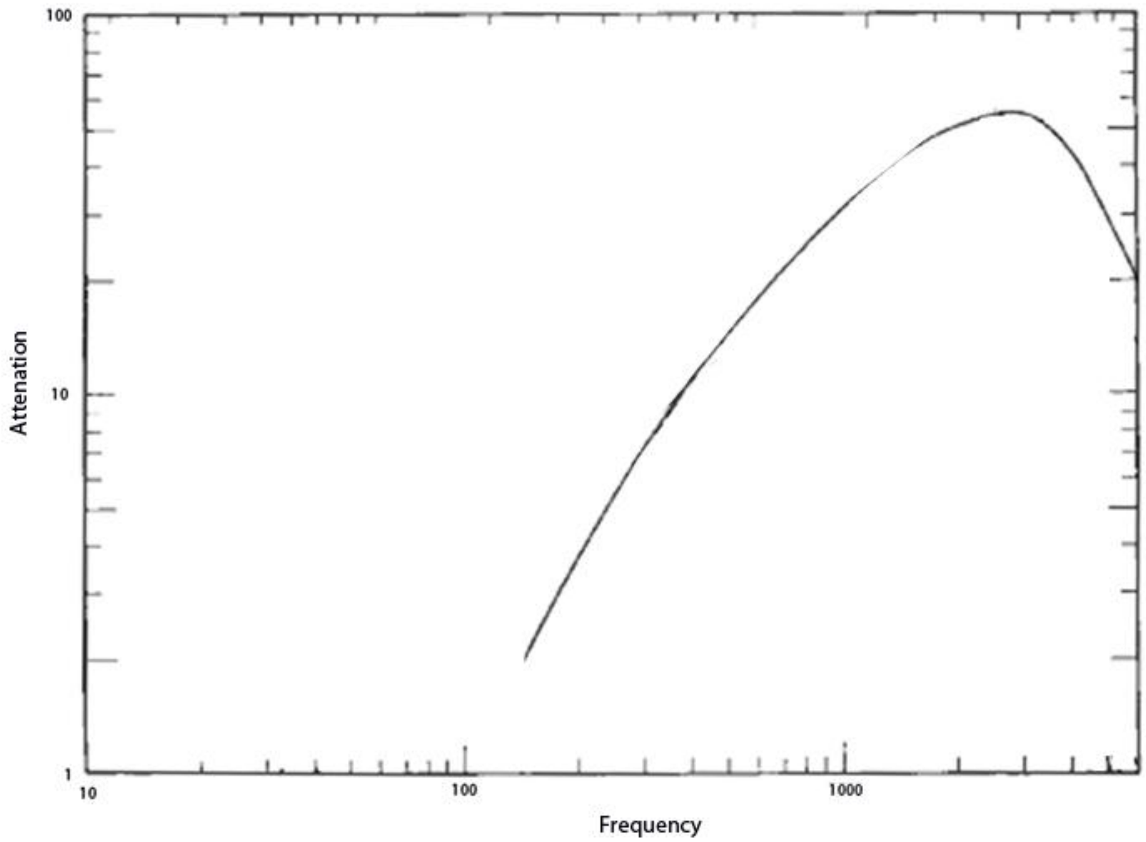

Figure 3.2: Attenuation curve according to Galaitsis and Vér's method

Table 4.2: Final reduction values using Galaitsis and Vér's method

Frequency	100	200	500	1000	2000	4000
Attenuation	-	3.5	18	32	52	48

This method takes more time and is unwieldy to use, partially due to transparent graph paper not being used to the same extent today as when the book was written and partially due to the difficulty in understanding the example presented by Galaitsis and Vér. This method is severely limited for several reasons. Since the attenuation curve is shifted both horizontally and vertically there are cases where the curve will just end in the middle of the graph as can be seen in the image above. Though the rest of the curve can be extrapolated this remains as a limit of the method. This method is also primarily a graphical interpretation of the attenuation for a baffle silencer, making room for human errors along the way. The reduction values produced from this method vary from the ones produced by Embleton's though there is no way of knowing which ones are correct until a proper comparison is made. This will be done in the next chapter. Despite all the shortcomings this method could still potentially be useful as it takes the flow resistance into account and can also be used in reverse to obtain baffle sizing from reduction values, though this assumes that the reduction values are close enough to one of graphs from which the attenuation is obtained. The usefulness of the inclusion of the flow resistivity depends on whether it is extremely high or extremely low. Galaitsis and Vér mention themselves that ignoring the flow resistivity would produce an error of barely a few dB though care should be taken when the resistivity starts to vary too much. As mentioned earlier, having half or double the value does little to nothing to change the outcome while larger differences can start causing variations in the results.

4.3 Effect of temperature

A different gas temperature will result in the speed of sound in the gas having different properties. While the speed of sound is approximately 343 m/s in air at 20° C it will increase or decrease depending on the actual temperature. An equation describing this behavior is given by Galaitsis and Vér as:

$$c(T) = c_0 * \sqrt{\left(\frac{273+T}{293}\right)} \qquad (4.1)$$

Where c_0 is the speed of sound in air at room temperature and T is the temperature in degrees Celsius. There is an error in the book, stating that the value 530 should be used and not 293. This made no sense partially because 293K is normal room temperature in Kelvin and partially because calculations with the value 530 produced nonsensical results.

When temperatures start reaching values around 100C and more the speed of sound becomes drastically higher. A higher speed of sound will result in longer wavelengths. This will influence the way that the sound is absorbed in parallel baffles due to their absorption being related to the wavelength of the incoming sound. The changes that this brings to calculating the total attenuation of a baffle absorber is minuscule, requiring only an additional calculation before proceeding as usual. The example below will illustrate how a change in temperature affects the attenuation using Embleton's method.

Example

A parallel baffle absorber, with the same size and spacing as the previous example, operates at temperatures of 200°C and no flow.

The increased temperature will result in the speed of sound being 432 m/s.

$$c(200) = c_0 * \sqrt{\left(\frac{273+200}{293}\right)} \approx 436\ m/s$$

From here the calculations are done in the same manner as previously, calculating *ly/λ* with the new speed of sound and reading the graph for A_{ly}. The values for this particular temperature would be:

Table 4.3: Temperature affected reduction values using Embleton's method

Frequency	100	200	500	1000	2000	4000
l_y/λ	0.029	0.046	0.115	0.229	0.458	0.917
A_{ly}	0.13	0.5	1.4	2.6	3.2	3.05
A	1.3	5	14	26	32	30.5

While it may seem that an increase in temperature will result in an overall reduction of the attenuation, there is actually more to it than that. The attenuation is smaller for the lower frequencies but is slightly larger for the higher ones, indicating that the attenuation curve has been shifted up in frequency. Comparing the results for room temperature and for 200C graphically one can see this pattern more clearly.

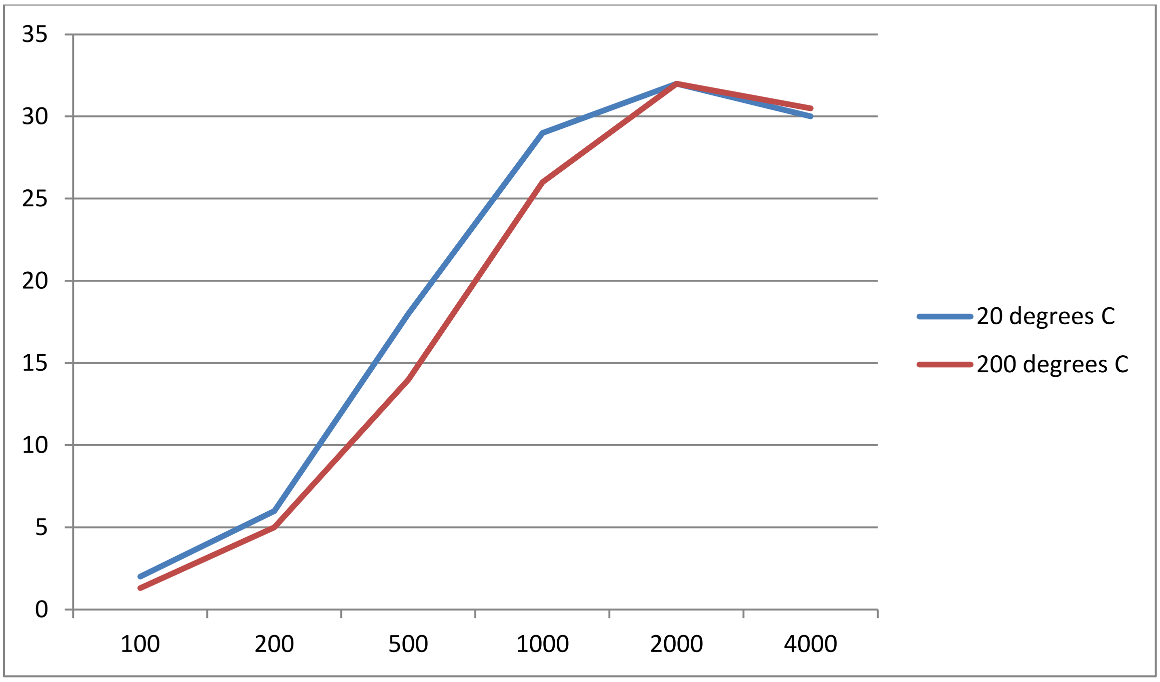

Figure 4.3: Difference in attenuation with two different temperatures

The change in sound speed is related to another alteration dependent on temperature, the alteration of the density of the medium in which the sound wave is propagating. Variations in temperature will have a distinct influence on air density in a given duct. The behavior of the variations can be explained by the equation:

$$\rho(t) = \rho_0 * \left(\frac{293}{273+t}\right) \tag{4.2}$$

Where *t* is the temperature in degrees Celsius and $\rho_0$is the density at 20° C, being $\rho_0 = 1{,}20$. The change in flow resistance with respect to flow is similar to the density and is expressed as:

$$R(t) = R_0 * \left(\frac{273+t}{293}\right)^{1{,}2} \tag{4.3}$$

Where *t* is once again temperature in C and R_0 is the flow resistivity at 20° C. This relation is useful when calculating baffle attenuation while using the method by Galaitsis and Vér. The below example will clarify further.

<u>Example</u>
A parallel baffle absorber, with the same size and spacing as the previous example, operates at temperatures of 200C and no flow.

Similarly to the previous example the calculations start with the speed of sound:

$$c(200) = c_0 * \sqrt{\left(\frac{273+200}{293}\right)} = 436\ m/s$$

Looking at the procedure used by Galaitsis and Vér this will affect the first step in the process, the determination of f'. At the current temperature and speed of sound f' will be:

$$f(T) = \frac{c(T)}{l_y} = \frac{430}{0{,}1} = 4360\ Hz$$

The rest of the calculations are done normally up to the calculation of the flow resistivity where the temperature once again is used. The new flow resistivity is then:

$$R(T) = R_0 * \left(\frac{273+T}{293}\right)^{1{,}2} = 5 * \left(\frac{273+200}{293}\right)^{1{,}2} = 8{,}88 \approx 9$$

This changes the curves used from the graph in appendix B2 from the solid one, R=5, to the one closest to R=9, the dash dotted one representing R=10, changing the reduction curve to the following shape.

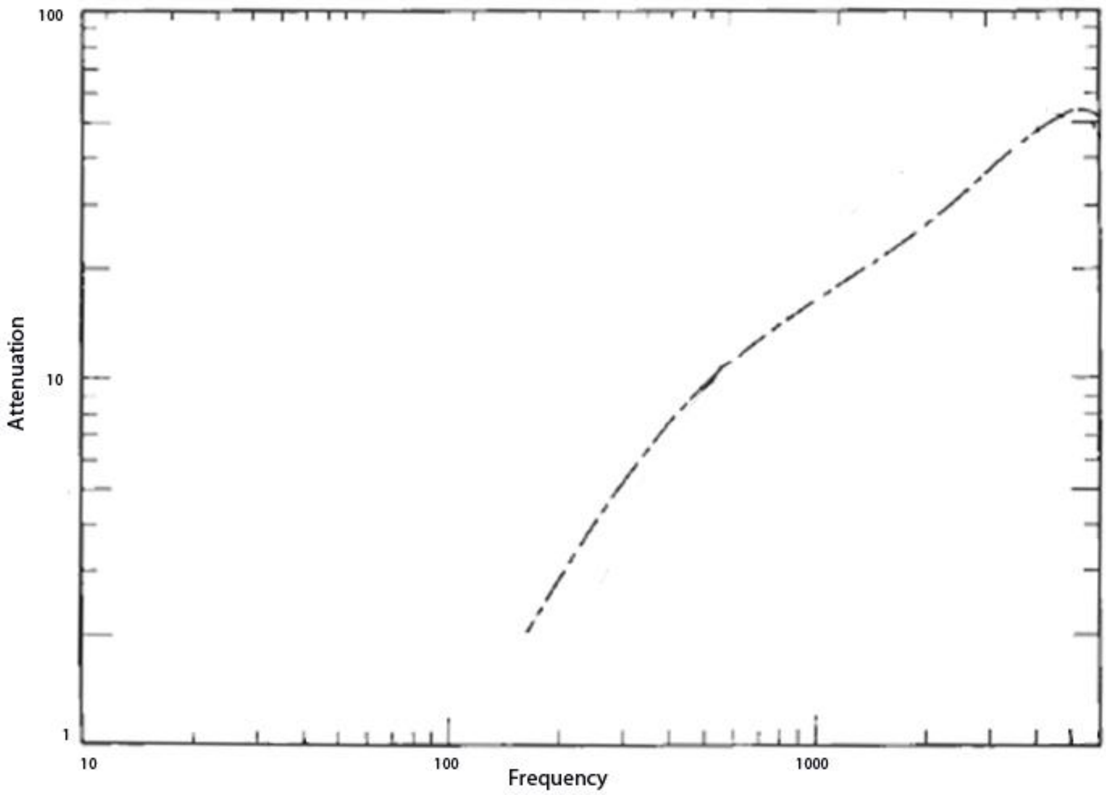

Figure 4.4: Attenuation curve influence by temperature

Table 4.4: Temperature affected reduction values using Glaitsis and Vér's method

Frequency	100	200	500	1000	2000	4000
Attenuation	-	2	11	18	27	50

The most obvious change is that the attenuations values are once again shifted to higher frequencies; additionally the shape of the curve is slightly different. Just like in the previous example the shift is due to the increase in sound speed, the change in shape is due to the increase in flow resistivity.

There is very little literature on the subject of attenuation for baffle silencers at different temperatures, the only real source of information being Galaitsis and Vér. There is also no real calculation method which is used specifically for the attenuation as every method encountered, despite the majority of them being based on the same theories, state that the attenuation at different temperatures should be extrapolated from the attenuation at room temperatures. This can be assumed to be a valid assumption as several authors on the subject of duct acoustics recommend this approach.

4.4 Effect of flow

The flow has a very distinct effect on the overall attenuation of a baffle absorber. Depending on the direction of the flow the attenuation can either increase or decrease making it important to know beforehand how much of a flow an absorber will be exposed to. Unfortunately the information regarding calculations of flow is somewhat ambiguous in some cases and simply lacking in some other. Galaitsis and Vér briefly cover the influence of flow in a duct, stating it affects sound speed as well as the effective flow resistance of the material in the baffles (Galaitsis & Vèr, 1992). Exactly how much it affects the resistivity or how this can be approximated is not explained. The effect of flow on attenuation is show through two graphs:

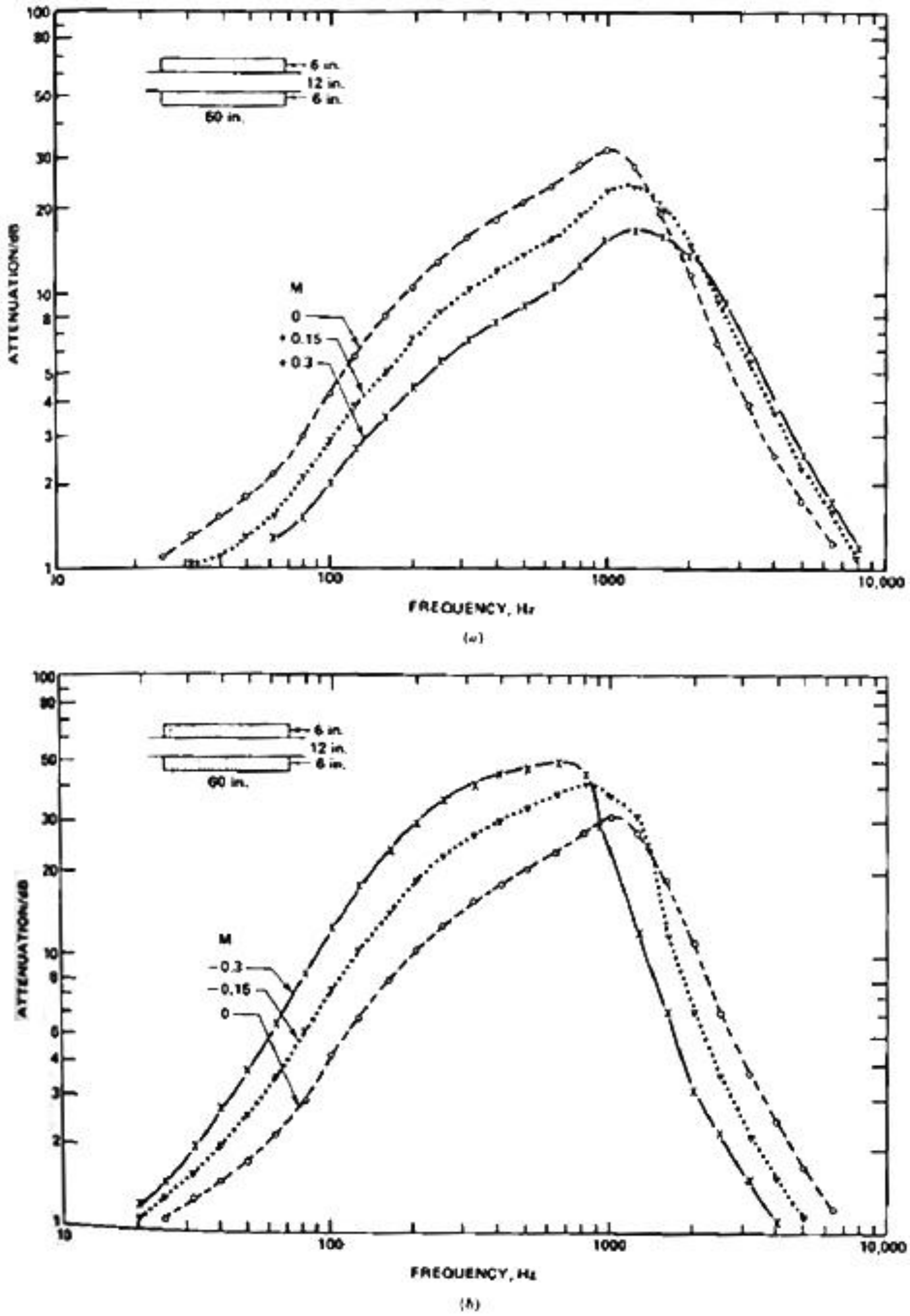

Figure 4.5: Changes in attenuation caused by various flow speeds

Clearly seen in the two graphs is that the attenuation is higher at lower frequencies when the flow is in the opposite direction of the sound propagation and vice versa. Although it is stated that the graphs are calculated and shifted from the standard case of no flow and room temperature there are no details as to how this is done. The only explanation given is another graph stating the rules for shifting the standard curve.

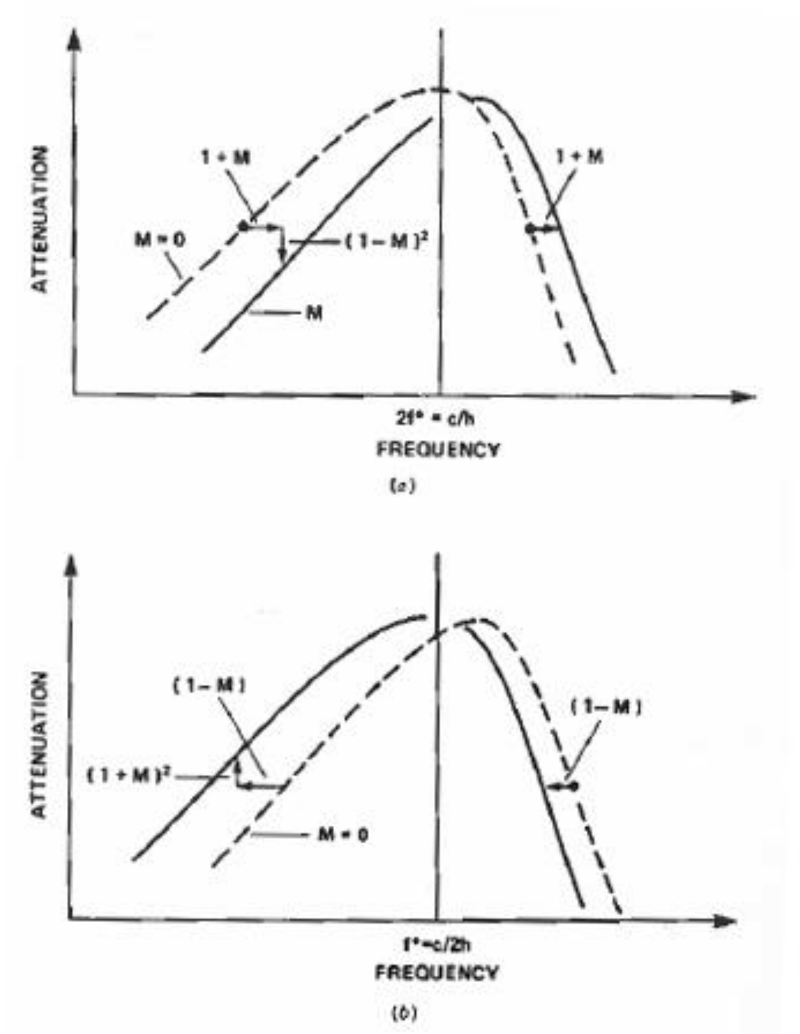

Figure4.6: Proposed method of shifting attenuation curve with respect to flow

Aside from this graph there are no further explanations by Galaitsis and Vér that state how this shifting should be performed. Attempts have been made to draw some conclusions on how this can be done and different methods have been tried though no results have come close the images above. The two authors do state that the shifting is based on experience with parallel baffle absorbers which could mean that there are no simple methods of shifting the attenuation and that Figure is just to illustrate the principle behind the shifting. With no further instructions on how to incorporate the shifting into the calculations the entirety of Galaitsis and Vér's method is unusable, despite evidence from the two authors that state that shifting a curve produces results that are in good agreement with computed values. This can be seen in appendix B4. However, in a ventilation system with a parallel baffle absorber, the flow velocity should be kept below 20 *m/s* in order to limit the pressure drop. This speed corresponds to a Mach number of 0,06, which is rather low, and the effect of such a low Mach number can probably be neglected.

Another method of applying the influence of flow on attenuation is presented by Embleton (1971). Assuming that there is steady flow through the duct the attenuation with respect to flow will be:

$$Attenuation = \frac{A_0(1+\gamma|M|)}{1+M} \tag{4.4}$$

Where:

A_0= Attenuation without flow
M= Mach number of the flow
γ= Experimental factor

The Mach number is dependent on the velocity of the flow, being positive downstream and negative in the opposite direction of the flow. It is simply expressed as v/c where v is the velocity of the flow and c is the speed of sound. If the velocity of flow is equal to the speed of sound then M will equal to 1 or -1 depending on the direction of the flow. It is shown that for small values of M equation (4.4) can instead be written as

$$Attenuation\ downstream = A_0[1 + (\gamma - 1)M]$$

$$Attenuation\ upstream = A_0[1 + (\gamma + 1)M]$$

One aspect about these equations that is plainly obvious is that there is no value attached to the variable γ. As stated earlier this is an experimental factor that is dependent on "many factors" though these are never clarified by Embleton, nor is there any further information of how this factor can be obtained. This is however elaborated upon somewhat by Almgren, giving some information on how this can be used (Almgren, 1978). While the experimental factor is not explained in full detail, stating that it is dependent on frequency and some boundary conditions amongst other factors, its usage in equation (4.4) is given. The information given is lacking somewhat as it simply states that if one want to be on the safe side then y=0 downstream. There is no further mention of this factor or the effect of flow, making it difficult to assess the usefulness of the equation.

An additional method of calculating the effect of flow is through the following equation:

$$Attenuation = A_0[1 - 1{,}5M + M^2] \tag{4.5}$$

It is derived from equation (4.4) as that equation can be written to look like equation (4.5) given a few initial conditions. This equation is only valid for values -0,3<M<0,3 as it becomes increasingly inaccurate for larger values (Ray, 2010). For most cases this is sufficient as flows with M>0,1 will result in material deterioration, high pressured losses and flow induced noise (Galaitsis &Vèr, 1992).

This equation, being easy to use and not missing any variables, is most suited to use when correcting for flow.

The flow induced noise, sometimes called self-noise, is always present when dealing with flow and can never really be ignored. If the self noise proves to be too large the baffle silencer must be redesigned as there is little that can be done to reduce the noise in other manners. How much self-noise sound power that is too much is dependent on the situation and the limitations imposed on the silencers. There is currently no universally agreed upon method of calculating the sound power generated by flow induced noise though there are empirically developed equations made to predict the generated sound power. One, presented by Galaitsis and Vér, is expressed in the following way:

$$L_{w,self} = 8{,}4 + 55\log_{10} V_f + 10\log_{10} A_f - 45\log_{10}\frac{P_{OA}}{100} - 25\log_{10}\frac{273+T}{293}\, dB\ re\ 10^{-12}\ W \quad (4.6)$$

Where:

V_f=Face velocity of the silencer(volume flow/face area)	[m/s]
A_f=Face area of the silencer	[m^2]
P_{OA}=Percentage open area	[%]
T=Temperature	[C]

An additional method is presented by Munjal in the following form:

$$L_{w,self} = 10\log_{10}\left(2{,}16 * 10^5 * V^{5,4} * \frac{S_t}{T^{2,27}P^4}\right) \quad (4.7)$$

Where:

V=Velocity	[m/s]
St=Face area	[m^2]
T=Temperature	[K]
P=Percentage open area	[%]

The sound power in these two equations is the octave-band sound power of the flow-generated noise which is approximately flat over the entire frequency region of interest. There is no easy way to conclude which of these is more correct due to the lack of empirical data on flow-induced noise though it should be mentioned that the method proposed by Munjal is based on theories by Vér and that the method proposed by Galaitsis and Vér are featured in a book released years after Munjal's.

4.5 Pressure drop

Pressure drop is always present in ducts. Depending on the operating conditions in the duct it may be more or less important. Volume flow or flow velocity is very important for the determination of the pressure drop as well as the assessment whether or not the pressure drop is actually important. As mentioned in the explanatory chapter, an increase in pressure drop can result in a larger need of flow speed in order to compensate for the pressure lost. For small installations with small volume flows the pressure drop can reach much larger relative values than for much larger installations. For power plant exhausts the flow velocity is usually rather high meaning that the pressure drop should be kept as low as possible. While the fans used in such power plants have the capacity to increase the flow to compensate for the loss this would cost much more energy making reduced pressure drop a much more desirable alternative. The pressure drop through a parallel baffle silencer can be calculated with the following equation (Almgren, 1978):

$$P = \frac{\xi \rho {v_b}^2}{2} + \frac{\lambda_{perf} L \rho {v_b}^2}{2 d_h} + \frac{\rho (v_b - v_0)^2}{2} \tag{4.8}$$

Where:

$\xi \approx 0{,}1$	[-]
$\lambda_{perf} \approx 0{,}05$	[-]
d_h=4A/R Hydraulic diameter	[m]
ρ=Density of air	[kg/m^3]
A=Face area between the baffles	[m^2]
R=Circumference of a channel between two baffles	[m]
L=Length of the baffles	[m]
v_o=Flow speed before the baffles	[m/s]
v_b=Flow speed between the baffles	[m/s]

This equation assumes that the incoming flow hits all the baffles evenly and that the front of the baffles is rounded. The image below shows this sequence of events.

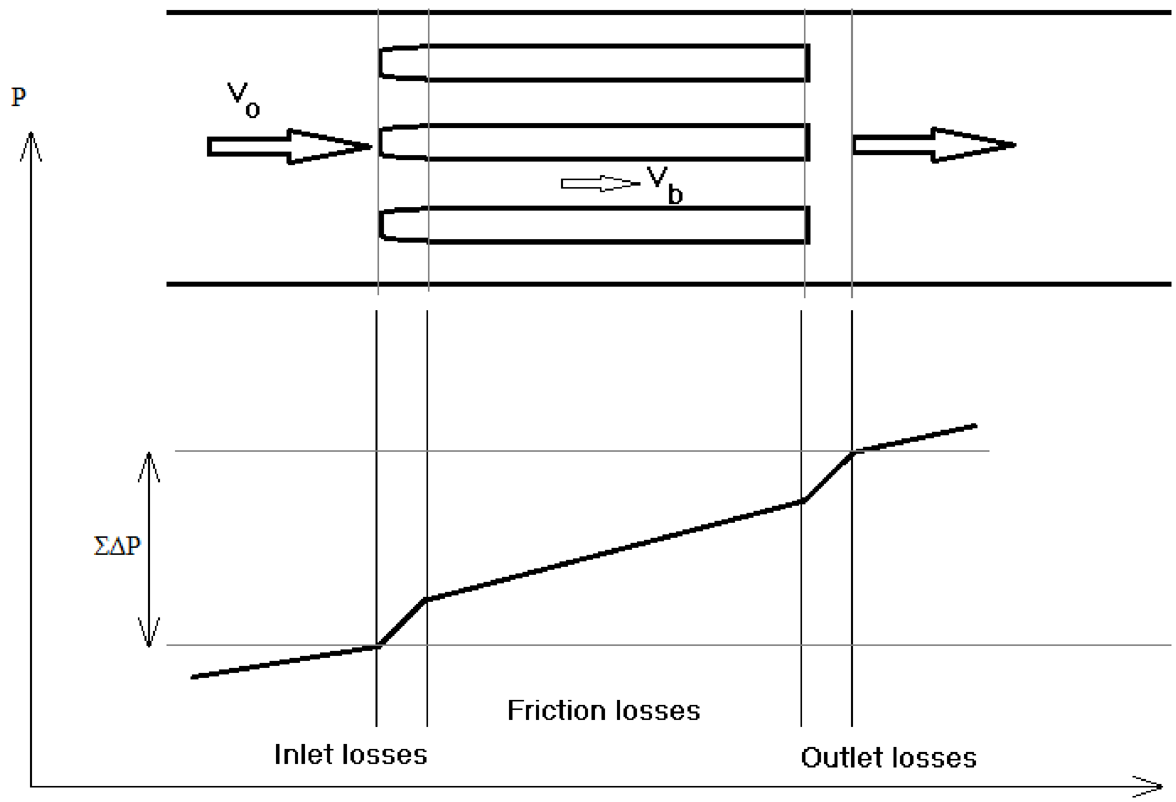

Figure 4.7: Pressure loss in a duct

The pressure drop occurring at the end of the baffles can be reduced by tapering the baffles though how this will affect the reduction values of the baffles is at present no known. It is difficult to create meaningful examples for the pressure drop since the flow speed between the baffles has to be determined which is not always easy. While approximations can be made there is a risk that they vary too much from reality and end up producing inaccurate results for the pressure drop.

4.6 Conclusion

Of the two presented methods for attenuation prediction, Embleton's sticks out as being easier to use and has a workflow that is simple and concise. Calculating attenuation with Galaitsis and Vér is a slower process that gives results that can easily be erroneous, though this will have to be checked in the comparison chapter. The presented methods cover most of the parameters involved in predicting silencer attenuation in ducts with temperature, flow, self-induced noise and pressure drop represented. Though there are other aspects of duct acoustics, such as expansion chambers, bend and plenum chambers, they are not directly related to parallel baffle silencers and are thus not a necessity to include.

5 COMPARISON BETWEEN MEASURED AND THEORETICAL VALUES

In order to validate the theoretical reduction values obtained in this report they will be compared to measured reduction values for silencers with the same size. Finding measurement data is somewhat difficult as the majority of measured data comes from silencer manufacturers. The issue here is that the only publicly available information about the silencers are the product catalogues which are often available online. These catalogues seldom contain all necessary information needed to make reasonable comparisons to the methods used in this report. Typically a product catalogue will contain reduction values for a set of mass-produced silencers while sometimes also including values for pressure drop or flow induced noise. The main issue is that there is no information on the size of the baffles themselves, thus making comparison a guessing game. A few published works do include sizing in their measurement and it is mainly these values that will be compared to the theory.

5.1 Comparisons

Laboratory measurements for baffle silencers have been found where the measurements were done according to ISO standard 7235 in a commissioned report for the Polish ventilation company, Centrum Klima. The reduction values found in the report were all centered on the same model of baffle absorber with a baffle width and spacing of 100mm but with different length. The figure below shows the insertion loss obtained in the text for the various lengths.

Test series	Description	Insertion loss D_e in dB							
		63	125	250	500	1000	2000	4000	8000
A	100/100/600	0	2	5	11	24	22	12	8
B	100/100/1000	0	5	9	17	33	33	19	12
C	100/100/1200	1	6	10	21	36	37	21	13
D	100/100/1500	1	7	14	24	35	38	26	17

Figure 5.1: Measured reduction values for a baffle silencer with various lengths

Inserting the same sizes into the calculation methods employed by Embleton the following reduction values are obtained:

Table 5.1: Calculated total attenuation using Embleton's method. Size corresponds to baffles from ISO 7235 measurement

	63 Hz	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	8000 Hz
A	0	1.8	5.4	11.0	18.0	19.8	16.2	9.0
B	0	3.0	9.0	17.5	29.5	32.0	28.0	16.0
C	0	3.6	10.2	21.0	35.4	38.4	33.6	19.0
D	0	4.5	12.7	26.3	44.2	48.0	42.0	24.0

The laboratory tests were performed in a controlled environment with normal room temperature and no flow through the baffles. As such no modifications have to be made to Embleton’s method in order to fully compare these numbers. The calculated values are close to the measured ones for lower frequencies while starting to become more inaccurate for higher ones. The longer baffles also produce a bigger error due to the length factor involved with Embleton's method. Of some interest is the fact that the theory produces attenuation values of 0 dB for some configurations, this is a limiting factor of the method as it cannot be used for very low frequencies in some cases. This has been mentioned earlier with an explanation on how to circumvent this problem.

The dimensions for case B in Figure correspond to the exact same curve as in Figure 3, thus the same graph can be used here as well. An assumption is made that the same value for flow resistivity can be used here as well.

Table 5.2: Calculated total attenuation using Galaitsis and Vér's method. Size corresponds to baffle B from ISO 7235 measurement

	63 Hz	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	8000 Hz
B	-	-	5.5	15	19	31	55	-

The limited nature of the method provided results for only five of the used frequencies. Though the attenuation at the other frequencies could have been approximated this was avoided here to showcase the values that are directly obtained by using Galaitsis and Vér's method. This method provides fairly accurate results for frequencies at and below 2000 Hz, being very close to the measured values and the results from Embleton's method. At 4000 Hz the reduction value differs greatly form measurements and becomes completely unreliable. It is unclear whether this method is supposed to be used at higher frequencies or if that is to be avoided as Galaitsis and Vér make no mention of the limitations of their method.

In a report for Acta Acoustica the material TROLIT was used to make parallel baffle silencers in order to test its absorptive properties (Kohn, et al., 2003). TROLIT is a material that is said to be non-flammable and has properties that make it useful for sound absorption given that it is created with sufficient pore size. The report states that TROLIT has very low flow resistivity though it is not specified what the exact value was for the measurements performed. In the report graphs are presented showing the insertion loss for the TROLIT baffles.

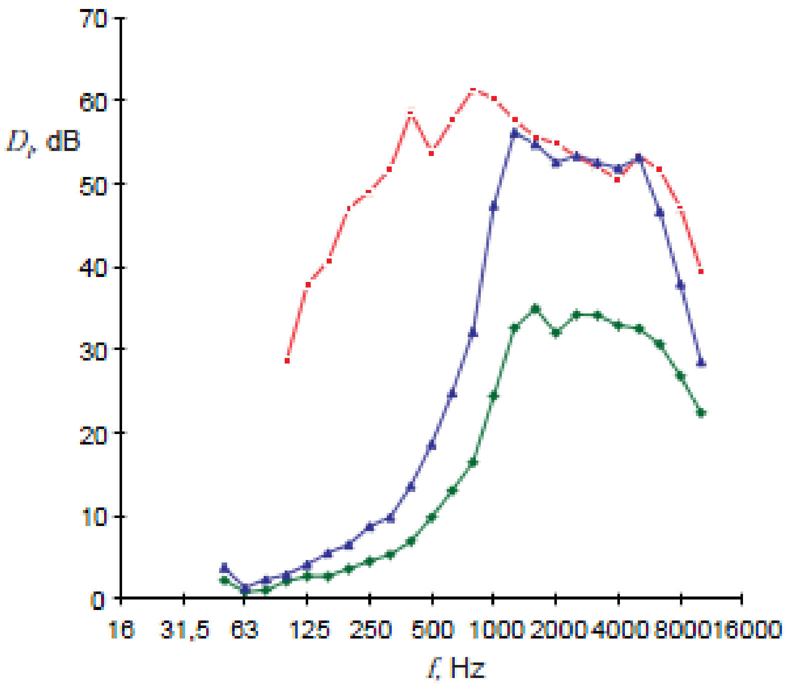

Figure 5.2: Insertion losses for laboratory measurements of TROLIT baffles with baffle width 80mm and 70mm spacing. Red line represents an acoustic plug of unspecified length filling the whole duct, blue line a length of 1200mm and the green line a lenght of 600mm

Since the report also provides the sizing and spacing of the baffles a comparison can be made to Embleton's method. Calculations performed provided the following attenuation for the same size and length. The calculations correspond to the blue line with triangles.

Table 5.3: Calculated total attenuation with Embleton's method. Baffle dimensions the same as the TROLIT baffles

Hz	100	200	500	1000	2000	4000
A_{ly}	0.2	6.8	24.0	44.6	56.6	51.4

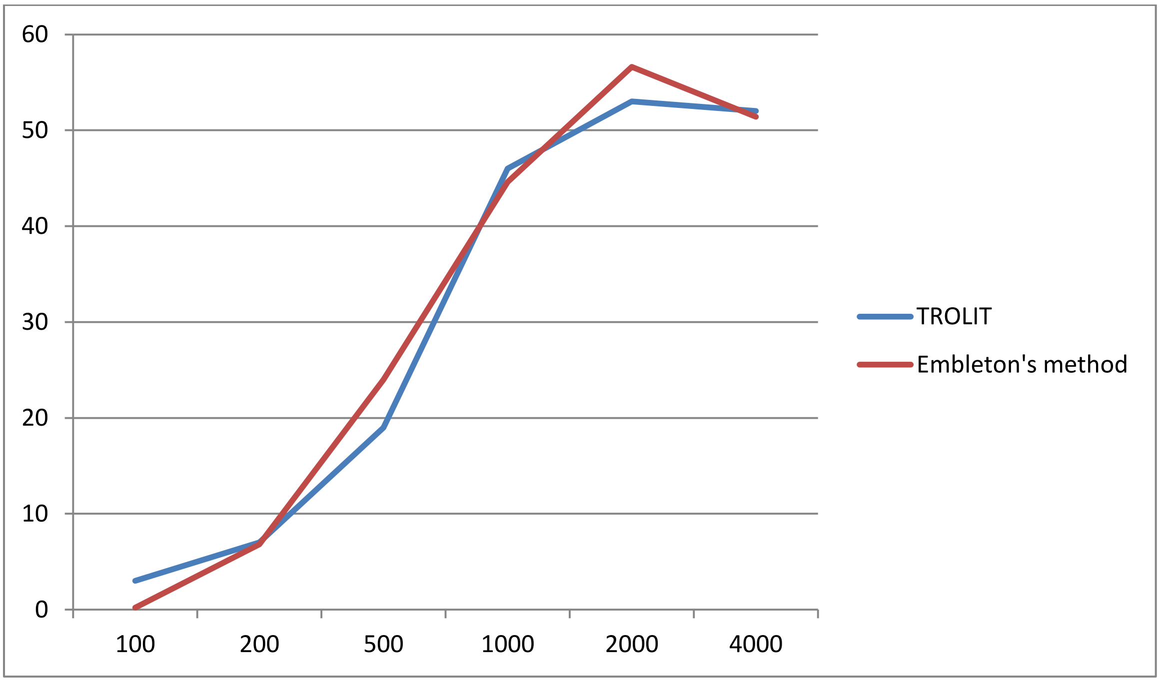

Figure 5.3: Comparison between the total attenuation for TROLIT baffles and calculated values using Embleton's method

Comparing the insertion loss with the calculated attenuation using Embleton’s method shows that the prediction is fairly accurate, varying by a few dB at the frequencies of interest. The error can depend either on the properties of the TROLIT material or due to human error when it comes to reading the reduction values from Embleton’s graph.

More comparisons have been made by Almgren where the Embleton method was compared to measured values (Almgren, 1978). As seen in appendix A, the predictions are fairly accurate at predicting the attenuation; however there are some errors that become quite large at certain circumstances. All of the charts show that with an open area of approximately fifty percent the attenuation of the theory and from in situ measurements will be reasonably close, giving a conservative approximation of the attenuation. With lower percentages, as seen in appendix A4 with thirty percent open area, the theoretical reduction greatly overestimates the actual reduction. Almgren suggests an upper ceiling for these reduction values and limits the maximum attenuation to half. Large open areas instead cause the prediction estimate to become too conservative, producing results that cannot be viewed as efficient predictions. All of the measurements show the same pattern in that the predictions are fairly accurate for frequencies below 500-1000Hz while becoming increasingly less accurate for higher frequencies. This is most likely due to the beaming effect mentioned by Embleton where higher frequency sound will become more directional and start “beaming” though the gaps between the baffles. While there is a solution to reduce the effects of this phenomenon, through the use of a staggered baffle arrangement, there are currently no calculation methods available that can be used to predict the attenuation for such an arrangement, nor has any measurement data been found.

One aspect of the comparison between predicted and measured values that has not been covered in this chapter is a proper comparison using manufacturer data. Several companies developing parallel baffle absorbers have been contacted with questions on how they estimate the attenuation of their products. The majority of these companies provided no answers, being unwilling to share any information on the specifics of their products, instead referring to their product catalogues. The issue with these catalogues is that they lack information that can be used with the calculation methods presented earlier, though some of the companies state that they measure the insertion loss of their products. An example of a product catalogue for a commercially available baffle silencer can be seen below, showing the various parameters for an IAC Acoustics manufactured silencer.

Clean Flow™ Rectangular

Page	Silencer Type	Face Velocity	Self Noise Lw	DIL, dB at 250 Hz				Pressure Drop in N/m²		Application
				Length (mm)				Length (mm)		
		m/s	dB	900	1500	2100	3000	900	3000	
50	HLFS	5.0	45	14	23	22	30	142	177	Fill protected silencers for low, medium and high velocity applications where cleanliness is critical such as hospitals, clean rooms, or laboratories. 'LF' series units are designed for increased low frequency attenuation.
52	HLFM	5.0	36	10	20	23	27	80	100	
54	HS	5.0	49	13	18	19	27	90	122	
56	HMS	10.0	52	8	11	16	23	25	47	
58	HLFL	5.0	30	10	14	16	22	20	25	
60	HL	10.0	51	3	7	9	11	12	17	
62	HML	10.0	52	6	10	12	17	12	22	

Figure 5.4: Typical product catalogue showing reduction values as well as a few other parameters (from IAC Acoustics)

The most important aspect that is missing from these catalogues is the width of the baffles and the spacing between them, only the outer dimensions for the entire silencer and its length are presented. While this may be good for a customer wishing to buy a silencer for their installation it does not provide any means of making a proper comparison with theoretical calculations.

Comparisons between calculated values for the influence of temperature and flow have not been possible due to lack of information regarding the subject. While there are a few charts that explore how different flow speeds affect attenuation of various baffle silencers, these charts lack details that would be needed to make a proper comparison such as baffle thickness, spacing or flow resistivity. As such their values cannot be directly compared the values obtained by calculating the predicted attenuation though they can be used to gauge the general behavior of baffle attenuation with various flows.

Model no.	Octave band, Hz	1 63	2 125	3 250	4 500	5 1K	6 2K	7 4K	8 8K
	Silencer face velocity, fpm	Dynamic insertion loss, dB							
3Es	−2000	6	10	19	29	36	34	23	15
5Es	−2000	11	19	24	40	52	49	27	17
7Es	−2000	11	21	40	53	54	54	38	24
10Es	−2000	14	30	40	57	58	56	47	29
3Es	−1000	5	10	17	28	36	34	23	16
5Es	−1000	11	17	23	39	50	49	32	20
7Es	−1000	11	22	38	50	55	54	43	27
10Es	−1000	12	31	44	54	55	56	50	31
3Es	+1000	4	8	15	25	34	34	24	17
5Es	+1000	8	13	20	35	50	49	36	24
7Es	+1000	8	18	35	48	54	53	46	32
10Es	+1000	10	26	42	55	54	57	51	38
3Es	+2000	3	7	13	23	32	34	24	18
5Es	+2000	7	12	19	33	48	49	36	25
7Es	+2000	7	16	32	46	54	51	48	34
10Es	+2000	8	23	39	55	54	57	52	41

Figure 5.5: Attenuation as affected by various flow velocities

Attenuation with flow, + denotes flow in the same direction as the sound waves and – denotes flow in the opposite direction (Bell, 1994)

The above chart shows the how important it is that flow is not ignored, its direction making the biggest difference in total attenuation. 1000 and 2000 feet per minute correspond to approximately 5m/s and 10m/s respectively. These are flow speeds that are relatively small and would result in Mach numbers of 0,015 and 0,030 respectively, being quite far from the limit for self noise at M=0,1. Despite being small these flows still manage to affect the attenuation quite a bit. Using equation (4.5) on this chart with the flow velocities 5m/s and 10m/s does not produce the same results with the chart producing much greater reduction values. When used backwards to calculate the original reduction values before flow was applied to the silencer the results do not match, meaning that either the theory is wrong or that the values in the chart are incorrect.

Table 5.8: Comparative calculations for original reduction values

125 Hz	+1000fpm - +5m/s	Calculated A_0 $_{+}$5m/s	+2000fpm - +10m/s	Calculated A_0 $_{+}$10m/s
3E	8.0	8.18	7.0	7.3
5E	13.0	13.3	12.0	12.5
7E	18.0	18.4	16.0	16.7

It is difficult to state which one of these is more correct, the theoretical values or the measured ones, due to a lack of further information for the measured values and the way that the theoretical approach seems to be very simplified. As it stands there is simply a difference between the methods described in the previous chapter and the measured values presented here.

5.1 Conclusion

Initial comparisons show reasonably good agreement between calculated reduction values and experimental measurements for both prediction models. As is stated by Embleton himself his method becomes increasingly inaccurate for higher frequencies though the same applies for increasing baffle lengths. This is most probably caused by the nature of the length factor, being dependent on both the length and width of the baffles. Small baffle widths result in a larger length factor, leading to higher reduction values. This may not be entirely the case as the comparison with the TROLIT baffles shows much better correlation between measurements and calculations despite having baffles that are thinner. The reason for this behavior is not certain though it can be caused due to a different flow resistance parameter in the baffles. The other method, by Galaitsis and Vér, shows the same behavior as Embleton's method, becoming increasingly more inaccurate for higher frequencies. This method became much more inaccurate however as it produced results that were not even close to the experimental measurements. Of these two the best choice for approximating baffle attenuation would be the Embleton's method. The comparisons performed by Almgren further consolidate this. The lack of experimental data for baffles in high temperature conditions means that no verification on the temperature calculations has been made. While this is undesirable, the calculation method used can still be considered viable as it is recommended by several authors who themselves make no attempts to verify the accuracy of the proposed method. Ascertaining the accuracy of the chosen method for predicting the influence of flow has proven to be more troublesome as the calculated results do not match the data found. As mentioned earlier there is a lack of comprehensive data regarding the effect of flow on attenuation and the one source found is simply not enough to verify whether the proposed flow correction method is wrong or right. This ties in to the general lack of certain specific information from product catalogues that makes it impossible to make valid comparisons to commercially available products.

6 ATTENUATION CALCULATION SCRIPT

The methods previously presented have been written into a MatLab script to ease calculations and to make it possible to study how various parameters affect the attenuation through baffle silencers.

6.1 Script functionality

The Matlab script is made to work as a supplement to the methods presented in earlier chapters. It is assumed that a user would have read the theory chapter and would thus be familiar with the workings behind the script. This is not necessary however as anyone can easily use the script by simply inputting the parameters that are required. The script will ask the user to input required parameters either once or twice depending on how much information is needed. The image below shows how this conceptually works.

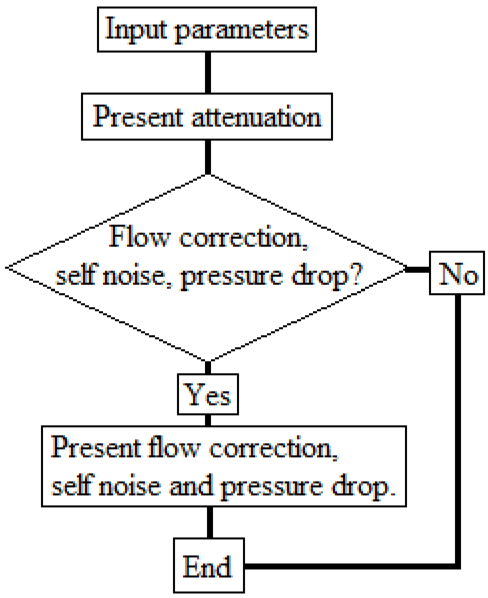

Figure 6.1: Work flow of the Matlab script

After the first series of input the script presents attenuation values for the baffle dimensions given. At this point the user can choose to input more parameters in order to calculate flow correction, self noise and the pressure drop or to just end the script.

6.2 Choice of methods

The calculations used in the script are based directly the theories from the earlier chapters. Due to how Matlab works some methods were more appropriate than other, additionally some of the methods have been proven to yield more accurate results. The method used to calculate attenuation is based on Embleton's theory. Partially because it yields more accurate results compared to measured data and partially because it was easier to implement than the other options. The curves in Figure are not as easily represented in Matlab as one would wish. The numerical values of the curves had to be entered manually for each point on the graph. This resulted in five different curves for the five different percentages of open area in Figure

3.1, with the others needing to be interpolated. Interpolation was done using preexisting commands in Matlab, resulting in curves for each percentage from 17% to 80%. An added benefit of the interpolation was that it increased the amount of points used for drawing the curves. Where the original curves had 211 points, the interpolated ones have closer to 4000. Occasionally the reduction graph obtained from the scrip will show zero attenuation at lower frequencies. This is due to the nature of the source curves from Figure 3.1 where some of them simply have no attenuation for low values of l_y/λ.

It should be noted that when the script asks for the width of the baffles the entire width has to be put in, not just the half that belongs to one channel. A first draft of the script did use half the width in the same way that Embleton does. Test subjects testing the script found this difficult to understand and repeatedly used the entire width, hence the change to the script. The effect of temperature on the sound of speed was taken from Galaitsis and Vèr. This is used in conjunction with Embleton's method in order to properly account for temperature. Additionally the equation for estimating the generated self-induced noise was also taken from Galaitsis and Vèr. The effect of flow on attenuation has been presented in several different equations in chapter 4 but most of them had information lacking, hence the method chosen was the one presented by Ray. Though it has the limitation of being inaccurate for flow speeds that exceed M=0,3 it is sufficient for the script. The final calculation, pressure drop, was taken from Almgren.

6.3 Instructions for use of script

The following are instructions for how the script should be used and how the data should be interpreted. The procedure is presented step by step and will also explain the calculations that take place behind the script.

1) Run the script
   The script is written in such a way so that the user only has to run it in Matlab. There is no need to change anything inside the script as the user puts in the needed parameters when the script starts

2) Input the required parameters
   The script will use these parameters to determine which curve to interpolate and how large the attenuation will be.

3) Attenuation is presented
   A graph is presented showing the attenuation values for the current baffle configuration. Depending on the input parameters the attenuation curve will shift along the frequency axis and will not represent all available frequencies. This is due to the interpolated curves having clear limits for the amount of points that can be drawn.

4) Stop the script or continue for flow corrections, self generated-noise and pressure drop
   At this point the user can choose to either stop the script or continue to obtain additional information. The script gives a prompt to either write y or n, denoting that the user wants to continue or stop. Should they choose to continue they will be

informed that additional parameters will be needed, hence the option to stop the program should these parameters be unavailable.

5) Input the required parameters
The script prompts the user to input values for the flow velocity between the baffles, the height and the width of the duct as well as the density of the gas propagating in the duct.

6) Attenuation, generated noise and pressure drop is presented
The script presents a new graph with attenuation values corrected for flow. The self generated-noise and the pressure drop are presented in the Matlab command window.

6.4 Parameter study with Matlab script

The Matlab script enables quick calculations of the attenuation produced by baffle silencers. This makes it possible to evaluate the parameters involved in the calculation process so that their importance can be determined, making decisions regarding which parameters can be sacrificed while still maintaining good attenuation easier. A standard case has to be determined before any studies can be made with the parameters. This standard case will feature parameters of a baffle silencer placed in a duct under simple operating conditions i.e. small downstream flow and room temperature. An assumption is made that the width and height of the duct before the silencer are the same as silencer itself, illustrated by the figure below.

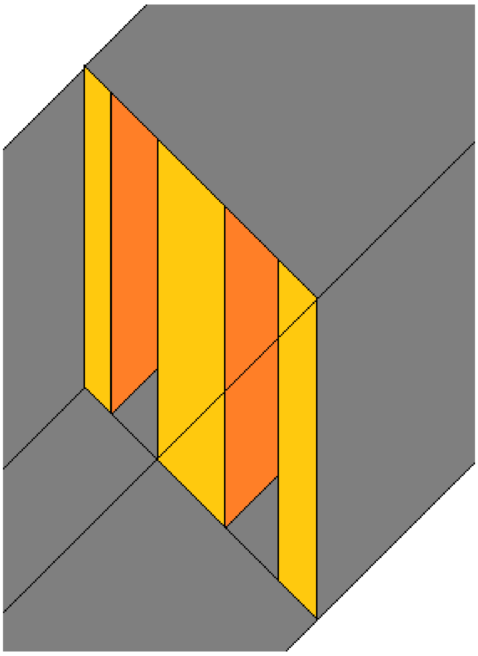

Figure 4.2: The width and height of the duct are assumed to be the same as the silencer

This is mainly done to ease calculations regarding the sound power as the size of the duct must be known for the volume flow to be calculated. These assumptions are not based on any real duct. Flow is included in the standard case so that self induced-noise can be generated for better comparisons.

For cases regarding baffle spacing and baffle width the pressure drop will occasionally show a pattern where a lower percentage of the open area results in a lower pressure drop, despite this being unreasonable. This is due to equation (4.8) using the flow speed between the baffles. When changing the ratio between baffle spacing and baffle width, not only does the open area change but the cross-sectional area of the channels between the ducts changes as well, resulting in changes to the volume flow. For cases where the open area percentage is low, equation (4.8) will lower the total volume flow so that the flow speed between the baffles will be kept the same. This can clearly be seen in the baffle width study where the spacing is kept the same but the width is increased, the pressure drop gives lower values despite the open area being lower. The frequencies among the presented attenuation tables do not go below 63Hz as Embleton's method is unreliable at lower frequencies. The limits for this change with the speed of sound and the baffle spacing, thus 63Hz is kept despite the fact that the prediction method is unreliable even at that frequency. The table below shows the parameters for the standard case:

Table 6.1: Input parameters for the script

Baffle width, t	0.2m
Baffle spacing, l_y	0.1m
Temperature, T	20 C
Gas	Air
Flow velocity between baffles	+10 m/s
Duct width	0.6m
Duct height	0.5m
Silencer length	2m

Baffle spacing here means the width of the open duct between the baffles.

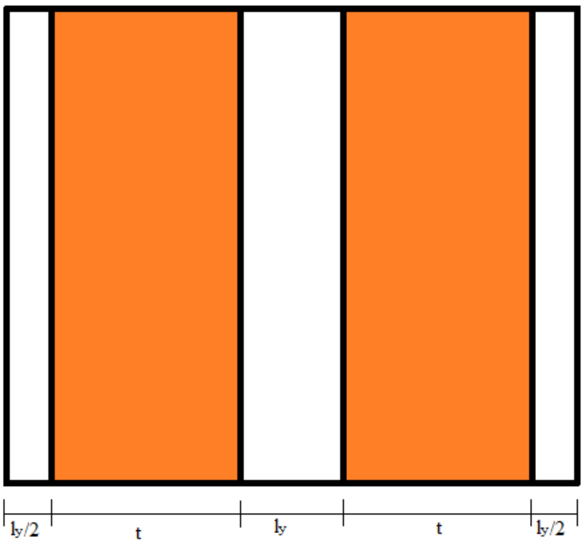

Figure 6.3: Principal figure of script example

Standard case

Table 6.2: Reduction values for the standard case

	63 Hz	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	Self-noise	Pressure drop
Attenuation	6.0	14.5	32.8	48.1	59.8	60.8	51.7	85.0 dB	29.6 Pa

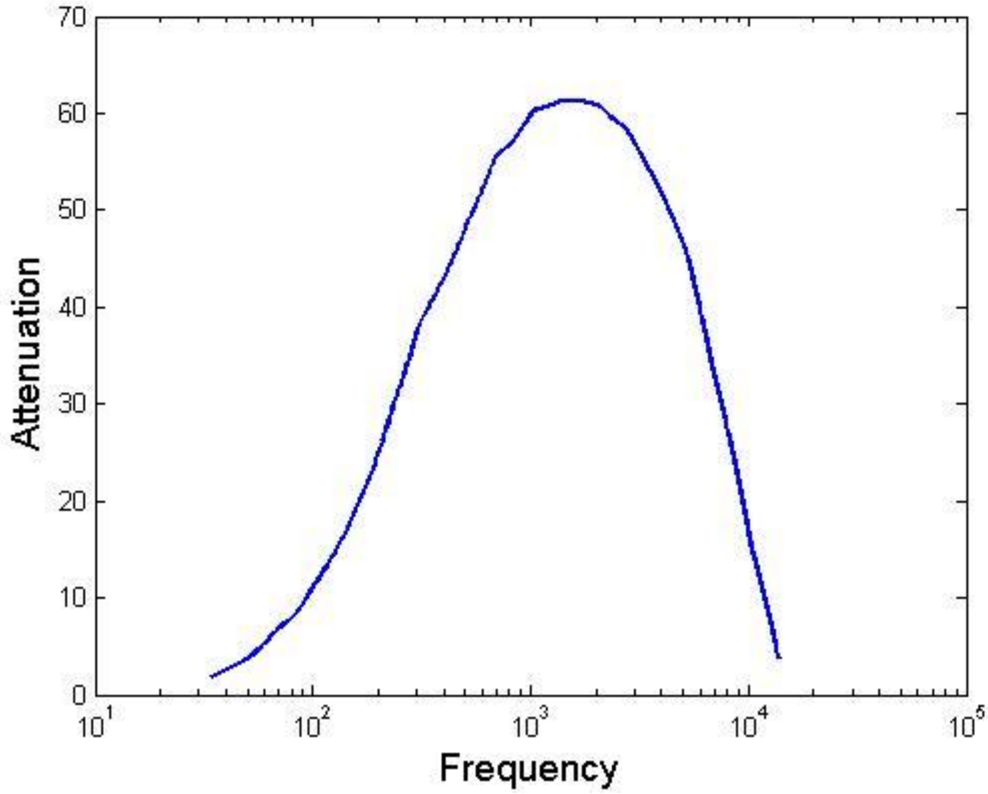

Figure 6.4: Standard reduction curve used for comparison

Temperature comparison

Table 6.3: Reduction values with respect to temperature

	63 Hz	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	Self noise	Pressure drop
T_{norm}	6.0	14.5	32.8	48.1	59.8	60.8	51.7	85.0 dB	29.6 Pa
T_{50}	5.5	13.5	30.3	46.9	59.0	60.9	52.6	84.2 dB	29.6 Pa
T_{100}	4.7	12.6	28.2	43.0	57.9	61.0	54.0	82.1 dB	29.6 Pa
T_{150}	4.2	11.7	26.3	43.9	57.1	61.2	55.1	81.0 dB	29.6 Pa
T_{200}	3.6	10.8	24.6	42.7	56.6	61.2	56.1	80.0 dB	29.6 Pa
T_{500}	2.3	7.6	18.6	38.2	52.7	60.8	59.1	74.1 dB	29.6 Pa

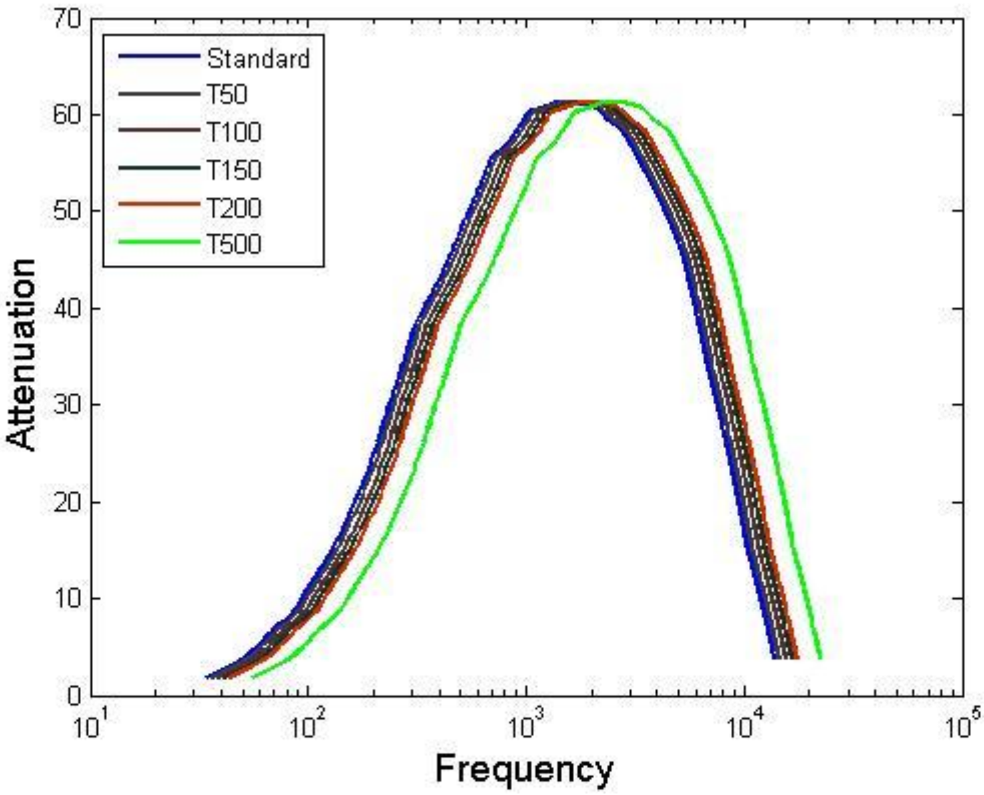

Figure 6.5: Calculated attenuation curves affected by temperature, standard=20°C

Flow comparison

Table 6.4: Reduction values with respect to flow

	63 Hz	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	Self noise	Pressure drop
F_{norm}	6.0	14.5	32.8	48.1	59.8	60.8	51.7	85.0 dB	29.6 Pa
F_{+25}	5.5	13.6	29.7	45.0	56.0	57.0	48.4	107 dB	185 Pa
F_{+50}	4.8	12.1	26.6	40.3	50.1	51.0	43.3	123 dB	741 Pa
F_{+75}	4.3	11.0	23.9	36.2	45.0	45.8	38.9	133 dB	1670 Pa
F_{-25}	6.9	17.0	37.0	56.0	69.7	70.9	60.2	107 dB	185 Pa
F_{-50}	7.5	19.0	41.1	62.3	77.5	78.8	67.0	123 dB	741 Pa
F_{-75}	8.4	20.9	45.7	69.2	86.0	87.5	74.3	133 dB	1670 Pa

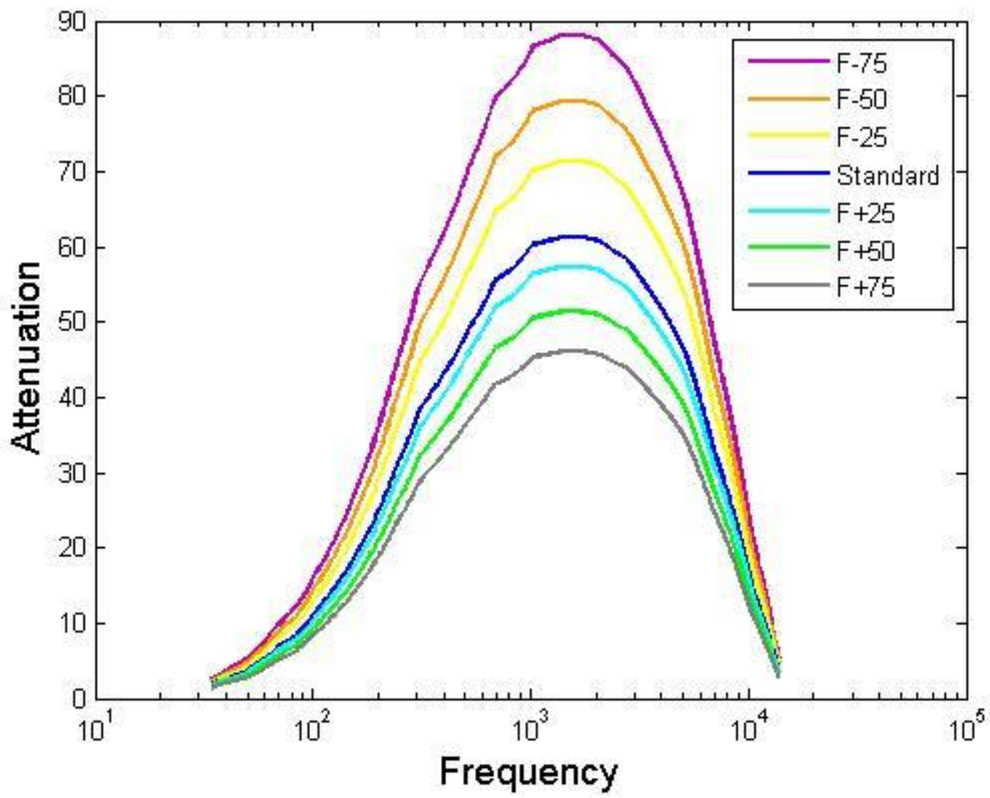

Figure 6.6: Calculated attenuation curves affected by flow, standard=10m/s

Baffle width

Table 6.5: Reduction values with respect to baffle width

	63 Hz	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	Self noise	Pressure drop
t_{norm}	6.0	14.5	32.8	48.1	59.8	60.8	51.7	85.0 dB	29.6 Pa
$t_{0,15}$	3.3	10.9	25.6	42.5	58.1	61.4	53.1	73.2 dB	36.4 Pa
$t_{0,1}$	0	5.3	16.8	35.2	55.7	62.2	54.9	59.3 dB	44.6 Pa
$t_{0,25}$	8.2	17.2	33.9	49.3	60.0	60.5	51.0	93.1 dB	25.3 Pa
$t_{0,3}$	10.7	20.0	36.0	50.5	60.3	60.2	50.2	102. dB	20.6 Pa
$t_{0,4}$	13.7	23.2	38.7	52.1	60.7	59.9	49.3	116 dB	15.1 Pa

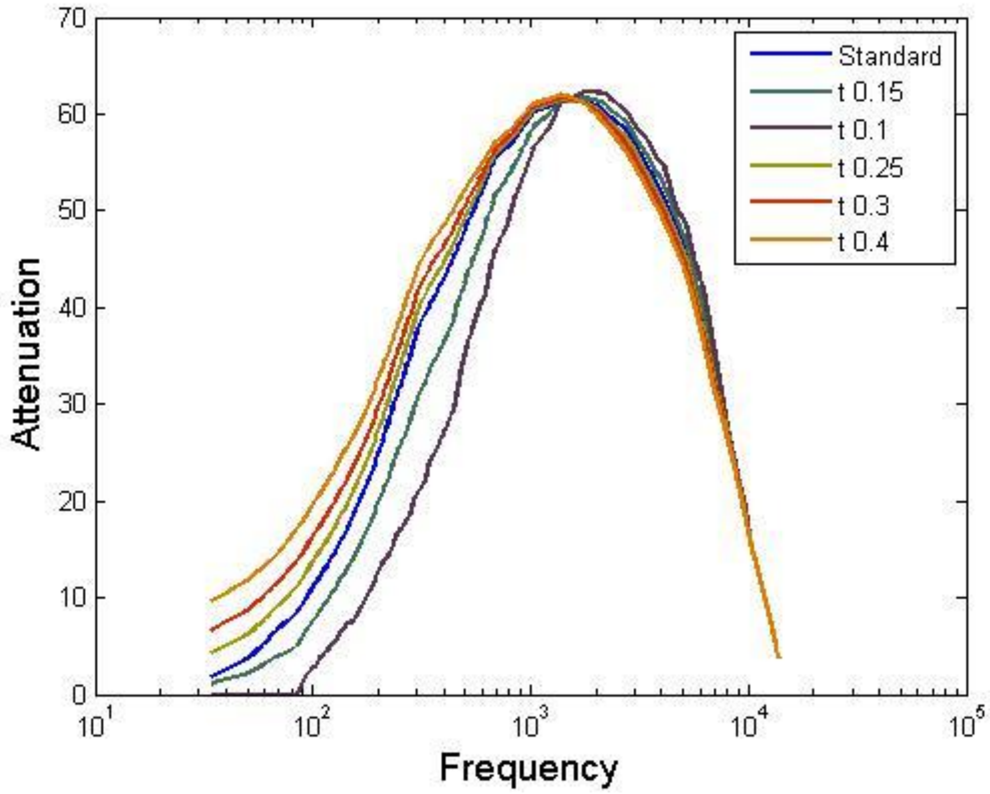

Figure 6.7: Attenuation curves for various baffle widths, standard=0.2m

Note that with these changes to the baffle width, the duct width no longer matches the sizing stated for the standard case. Hence the calculations regarding the flow-generated noise and the pressure drop are not entirely correct. They manner in which these two values change, by altering the parameters used to calculate them, is still interesting and thus they are kept.

Spacing

Table 6.6: Reduction values with respect to baffle spacing

	63 Hz	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	Self noise	Pressure drop
$l_{y,norm}$	6.0	14.5	32.8	48.1	59.8	60.8	51.7	85.0 dB	29.6 Pa
$l_{y,0.05}$	-	27.1	46.2	77.5	104	121	119	100 dB	33.3 Pa
$l_{y,0.15}$	3.7	10.9	21.6	34.1	41.0	38.7	27.8	78.2 dB	27.9 Pa
$l_{y,0.2}$	2.7	8.4	17.6	27.8	31.1	27.4	13.7	75.3 dB	26.6 Pa
$l_{y,0.3}$	1.9	5.8	12.2	19.5	20.0	14.6	3.2	73.1 dB	27.6 Pa
$l_{y,0.4}$	1.3	3.9	10.6	15.2	14.4	7.5	-	73.1 dB	33.2 Pa

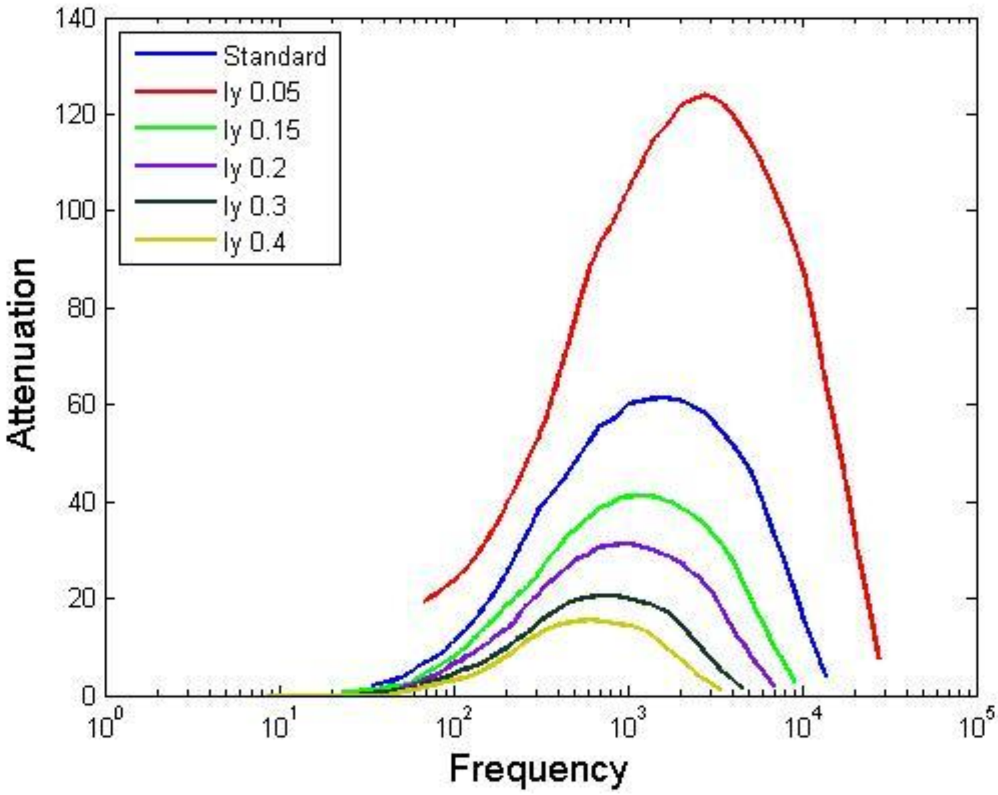

Figure 6.8: Attenuation for various baffle spacing, standard=0.1m

As with the previous parameter the change in spacing causes the duct width to no longer accurately represent the new actual duct. The flow-generated noise and the pressure drop are once again erroneous though they can still be used for analysis.

6.5 Conclusion

The script performs as expected with attenuation curves closely resembling the ones from Figure 3.1, the main difference being that the curves produced by the script look jagged and less smooth. This is due to the values from the curves being manually inserted into the code at each point. Coupled with the interpolation that occurs in the script the curves look rough and unappealing though they function exactly as needed. The interpolation itself is a useful tool for approximating attenuation for baffle width and spacing rations that do not fit the curves from Figure 3.1. This kind of interpolation can be done manually, though such an approach is prone to human errors and one could easily make mistakes regarding the actual reduction values. The interpolated curves from the script represent every percentage of open area between their limits, making them sufficiently accurate for all predictions. Since the interpolation is done step wise between each curve from Figure 3.1 the interpolated curves can be considered accurate approximations for the reduction values they represent. Interpolation could have been done with only the curves for the lowest and highest percentages, making the script shorter and more effective though this would probably reduce the overall accuracy of the predictions.

The parameter study shows patterns that agree with the theories describing changes in baffle attenuation with different configurations and operating conditions. The temperature study shows a clear trend where higher temperatures result in the attenuation curve shifting upwards in frequency. This has the effect of lowering the attenuation at lower frequencies and increasing it for higher ones. In reality a change in temperature would also change the flow resistance parameter which would change the shape of the curve, though as mentioned several times earlier this would only result in small changes that can be ignored. Another interesting aspect of the temperature study is that the self-induced noise is reduced for higher frequencies. As there are no experimental values with which to compare the results the accuracy of the self-induced noise calculations cannot be ascertained. Assuming that the calculations are correct would mean that the self-induced noise is more easily controlled in exhaust ducts where operating temperature is high. The results from the flow speed study also shows expected results as the attenuation becomes much smaller for high downstream flows and higher for upstream ones. The study also clearly shows how high the pressure drop becomes with large flows as well as the self-induced noise, which reaches values above 100 dB. In most cases this would need to be addressed immediately, however flow velocities exceeding 75 m/s is not common among power plant exhausts. In Figure 6.4 all attenuation curves have the same lower and upper limit in frequency, differing from the temperature curves in Figure 6.3. It would not be entirely unexpected if large flow velocities would change the speed at which sound propagates in the duct, changing the sound speed parameter and thus also changing the position of the attenuation curve along the frequency axis. Since no useful experimental data regarding baffle attenuation under the effects of flow have been found the reduction values obtained from the flow velocity study cannot be verified.

The baffle thickness study is important as the size of the baffles is something that can be determined beforehand, unlike the operating conditions. Changes in baffle thickness show that thicker baffles result in greater attenuation for lower frequencies and smaller for higher frequencies. For very high frequencies, above 10000 Hz, the baffle thickness becomes less important. Should the operating temperature of a duct be very high, thicker baffles can be chosen to offset the reduction in attenuation brought by temperature. Unfortunately this also has the effect of increasing the self-induced noise as the face area of the baffles increases. This is one of many contradictory demands that come with designing a baffle silencer. A peculiar part of the thickness study is that the pressure drop becomes smaller with larger thicknesses. This should not be possible as the open area becomes smaller with thicker baffles. While this may look like an error it is actually a problem with the algorithm of the script and the original equation for calculating the pressure drop. The pressure drop equation depends on the flow velocity between the baffles. When the velocity is kept constant while changing the baffle width and spacing, the overall volume flow will change as well. For the thicker baffles the volume flow in the duct is actually lower than for the thinner ones, making it seem like the pressure drop becomes lower for thicker baffles. The same can be seen in the spacing study where larger baffle spacing results in larger pressure drop, despite the open area increasing. No pressure drop calculations could be verified as no measured data have been found with which to compare the results. While it is interesting to see this behavior with the calculations for the pressure drop it is best if the parameter is disregarded, partially due to the lack of verification data and partially due to the misleading nature of the results.

The spacing study is the most interesting one as it shows great variations in attenuation despite small changes to the parameter. One can notice that there are no results for the lowest frequency for the smallest baffle spacing and the highest frequency for the largest spacing. This is due to the position of the attenuation curve being dependent on the l_y/λ factor, hence there are no results in those two extreme cases. The importance of baffle spacing can be seen between the standard curve for l_y=0.1m and the curve for l_y=0.15 where the attenuation is much lower. In order to keep good attenuation the spacing should be kept low though this also results in an increase in self-induced noise as well as increased pressure drops. To keep the pressure drop and self-induced noise below an acceptable limit, the number of parallel baffles and ducts must be sufficiently increased. A large attenuation with limited pressure drop and self-induced noise leads to a silencer with a large cross-sectional area.

Overall the script provides useful means to quickly calculate attenuation values for baffle silencers using a simple method. The accuracy of the script is ensured by the fact that the script reduction values comply with the reduction values obtained from manually using Embleton's method, which itself has proven to be accurate. While the pressure drop and self-noise calculations are unverified the correction for flow speed can be considered as reliable, considering that several authors recommend the method used in the script. Regarding the pressure drop, the script could be modified so that a volume flow is used at the beginning which would then translate to a flow velocity between the baffles. The results for the pressure drop calculations would likely be much less misleading by implementing this change. However, these changes to the script would detract from the simplicity of the script in its current form. The way the script is written at the moment makes it possible to calculate different parameters with them being separate from each other. Changing the script to use volume flow instead of flow velocity would integrate several parameters into each other, making it less clear to see their individual effects.

7 CONCLUSIVE CHAPTER

7.1 Results

Calculation methods for predicting attenuation for parallel baffle absorbers have been presented. Comparisons have been made between theoretical results and measured values that have shown good agreement between theory and reality. Of the methods found only two were presented in detail. The rest were either based on one of these two or required numerical solutions for them to yield results regarding attenuation. The scope of this thesis does not include solving for such equations. Calculation methods for the influence of temperature and flow on attenuation have been presented as well as calculations for pressure drop in ducts with baffle silencers. No new simple theories have been found which can aid in making more accurate attenuation predictions nor have any theories been found that can be used to predict matters such as varying baffle width with respect to length. Measured experimental results for cases where the attenuation is affected by temperature or flow have not been found and thus the comparisons have been made with other calculated and computed results. Comparison with manufacturer data has not provided any results that can be considered useful. Contact with companies manufacturing baffle silencers resulted in no further data being provided.

The presented methods were written into a Matlab script to facilitate repeated calculations. Using the script, a parameter study was performed and the various parameters involved were tested. The results of the study showed the effect on attenuation with changing parameters, with some parameters being more important than others. The study also showed some of the contradictory demands regarding baffle design, such as the improvement with increased baffle width and the simultaneous increase in self-generated noise.

7.2 Conclusion

The results of this report indicate that new information regarding the prediction of parallel baffle silencer attenuation is limited. The majority of the prediction methods available are several decades old, some of them reaching back to the 1950's with Cremer's theory. Despite their age the prediction methods are still reliable and produce values that are close to the real attenuation that a baffle silencer would produce, as shown by the comparisons to experimental data. Despite being mostly reliable they are still not optimal, producing results that are sufficiently close in most cases but sometimes lacking in other cases, such as attenuation at very low frequencies. The newer prediction methods that can be found are mainly based on the chapter by Galaitsis and Vér in Beranek's *Noise and Vibration Control Engineering: Principles and Applications* from 1992. Other authors such as Munjal and Ray base their calculations on the theory and methodology of Galaitsis and Vér. While there is newer literature on the subject of baffle absorbers, such as the theories by Munjal, these are more focused on the behavior of sound in ducts with baffle absorbers rather than the attenuation of the absorbers themselves. Additionally these theories are apparently not developed with engineering use in mind since they often involve complicated differential equations which would take large amounts of time to solve. It is likely that these theories are directed more towards academia than towards engineers needing a formula in order to predict the attenuation of a silencer. The prediction methods regarding the effect of flow and temperature are easy to use and provide results that are easy to understand. The accuracy of these prediction methods remains unknown since no measured data could be found in order to make a valid comparison. As it stands the reliability of these methods is based entirely on the authors' credibility. Further comparisons could have been made with commercially available baffle silencers had the necessary parameters been available. The lack of detailed information from product catalogues and answers from manufacturers meant that no concrete comparisons could be made to commercial products. The general unwillingness of manufacturers to answer questions on how their products are manufactured or how their final attenuation is predicted indicates that the information regarding this is something that the companies do not wish to part with.

7.3 Discussion and future work

It is stated among the goals for this thesis, that existing theories for predicting baffle silencer attenuation would be presented in a manner so that engineers from other disciplines than acoustics could use them and that more recent theories on the subject would be searched for and presented as well. Of these only the first was fully accomplished. The calculations methods for predicting the attenuation of baffle silencers presented in this report are simple enough to use and provide the user with a quick and effective way of predicting attenuation given a few key parameters. The script included in the thesis further makes it easy to use the various methods obtained. Good agreement with measured values means that the prediction methods are reliable and can be trusted for those cases where the agreement was sufficient. Calculation methods for estimating the effect of temperature presented in this report have not been tested in any way due to the lack of any experimental values with which to compare calculated values. The same applies to changes in the flow velocity where no experimental values have been found to support the calculated values. While their influence is described in literature the lack of verification means that there exists a possibility where actual reduction values deviate significantly from prediction. Measured data for high temperatures is interesting since it not only changes the speed of sound but also the flow resistivity of the medium, thus changing the shape of the reduction curve. Despite the fact that temperatures in power plant exhausts rarely reach the upper limit used in the parameter study it would still be of interest to evaluate how much a reduction curve changes with extreme temperatures.

Measured data for the influence of flow was found but did not comply to calculated values at all. Equation (4.5), as presented by Ray, has its origin in equation (4.4) by Embleton and is also featured in reports by Almgren, indicating that it has a solid theoretical grounding. Assuming that the equation is correct would mean that the chart in Figure 5.4 is incorrect or that there is additional information not presented in the literature. Without additional measurement data it is not possible to fully ascertain whether the theory or the chart is correct. The other method for predicting the effect of flow on attenuation, as presented by Galaitsis and Vér, proved to be impossible to replicate, despite several attempts. Finding out how to actually use this method would be useful as it is still of interest to see how it compares to Embleton's method as well as measured values. Noise caused by flow is a bit special as there is a lot of measured data for the parameter, unfortunately all of it stems from commercial product catalogues. As has been stated earlier, the information from these catalogues is generally lacking one or two key parameters, prohibiting proper analysis. Another parameter commonly found in catalogues is the pressure drop. Being one of the more important aspects of baffle design, aside from the attenuation itself, it is quite easy to find information regarding pressured drops from baffle silencers, especially among commercial silencers. The downside being that manufacturers tend to use their own terms when describing pressure drops which may not be useful for a comparison in this thesis. Pressure drop in literature is rarely fully explored, with the majority merely mentioning its importance and not including ample instructions on how to actually calculate the pressure drop. A reason for this could be that pressure drop is not inherently a part of acoustics but rather belongs to fluid mechanics and other similar disciplines within engineering. Empirical testing on baffles with known size would be of great interest for confirming the calculation method present in this thesis.

Of the two main methods presented for predicting baffle silencer attenuation, by Embleton as well as by Galaitsis and Vér, the first one is recommended for use. The simplicity and short amount of time needed to perform the calculations coupled with the fact that it gives fairly accurate results makes it a desirable option. One should be aware of the limitations of the method however, as comparisons have shown that it can overestimate attenuation greatly in some cases. As was mentioned when discussing Almgren's reports, the method by Embleton can occasionally produce much larger attenuation values than what a baffle silencer actually would produce. The red curve from Figure 6.6 is probably one such case as it produces attenuation values above 120 dB. This amount of reduction from baffles 20 cm thick and 2 m long is unreasonable, meaning that this is one such case where the predicted attenuation values are grossly overestimated. In his report, Almgren suggests a method for these extreme cases where the maximum attenuation is reduced. The same can be assumed to work here as well. The other method of predicting attenuation, developed by Galaitsis and Vér, also proved to give reasonably good results compared to measurement. The method itself, however, is cumbersome and takes considerably more time to utilize than the one by Embleton. The need of transparent paper further makes this method unwieldy to use since calculations today are mostly done with computational software. This procedure was probably more useful at the time the source material was written, over twenty years ago. Despite its age and the cumbersome nature of this prediction method it is well referenced to in other texts while the method by Embleton is barely mentioned anywhere aside from the investigations made by Almgren. Seeing as how both Embleton's and Galaitsis and Vér's methods are both featured in two of Beranek's books, both called *Noise and Vibration Control Engineering* with Galaitsis and Vér's method appearing in a later edition, it is possible that only the more recent book is used by newer authors, thus missing Embleton's method. With the exception for small modifications and changes for convenience, the majority of recent literature references to one or both of these two methods. The end results remain the same, as such there is no need to include the methods from every source found, and only the originals are needed. The fact that Galaitsis and Vér's method is heavily referenced among more recent literature shows a pattern that has appeared during the work on this thesis. There are no new simple methods for calculating baffle attenuation and it seems as if there is little research done on the subject. While there is literature on the subject, such as the report by Ochmann and Donner, it requires numerical solutions that may or may not be easy to solve. It would be of great interest to solve the equations used by Ochmann and Donner, as well as those used by Ingard, to see if a simple method can be derived from them. As mentioned earlier, Ochmann and Donner do make mentions of numerical solutions to their equations in a later report. However, such a text has not surfaced since the claim was made. While it is entirely speculation, it is possible that their solutions were acquired by a baffle silencer manufacturer, not wanting the information to become accessible by everyone. This is supported by the general attitude of manufacturers, not wishing to reveal their methods of sizing of their baffles or anything else involving their production. One could also speculate that manufacturers are acquiring research regarding baffle silencers for themselves in order to gain an advantage over their competitors, which could explain the lack of newer theories for predicting attenuation. Ochmann and Donner, as well as Ingard, write about the possibility of predicting silencer attenuation where the thickness of the baffles varies with their length, though no readily available method has been found. The possibility of predicting attenuation for baffles of varying thickness would immensely aid in baffle design. The fact that there exists no simple method for doing this could indicate that such knowledge is bought and classified by manufacturers, though this is only speculation. The theories by Ingard and Ochmann and Donner can still be solved numerically by anyone who would like to attempt such a task, though the amount of effort required to find these solutions is at present unknown and has not been evaluated.

Parallel baffle silencers belong to a small area of acoustics, being somewhat of a combination between duct acoustics and absorbers, along with some fluid dynamics. As such it is not entirely unexpected that there is limited information on the subject and that relatively little research is done. Coupled with the fact that baffle silencer manufacturers have historically depended mostly on their own measurements on their products and not prediction models creates an environment where there is little need for extensive research to be done. This brings up the question of whether or not newer prediction models are even needed as manufacturers seem content with the methods they are presently using. A problem with being content for too long is that technology and procedures may change without one noticing. There are already several concerns that are not readily addressed by manufacturers such as breakout noise and duct rumble. Though these subjects have not been addressed in the thesis they still exist as a part of the problem with noise in ducts and exhausts. There is also the possibility that the older measurements that manufacturers depend on may not match actual field performance. It could therefore be economically beneficial to keep pace with changes in technology and incentivize research regarding baffle silencers. Since manufacturers are not willing to openly speak of their procedures and inner workings it is possible that this is already the case, albeit on an internal level within the manufacturer companies.

The theories and methods presented in this thesis can be used as a useful basis for further work within the subject of parallel baffle silencers. Solutions to the equations by Ingard, Mechel and Ochmann and Donner would all be greatly beneficial to further improve prediction of silencer attenuation. Solutions to these equations hold probably the most potential to yield prediction models for a wide range of baffle silencers. Most useful would be the ability to predict attenuation for varying baffle thicknesses in the same duct. If this thesis is used as groundwork for further investigation the solutions to these equations should be the primary goal. Another point of interest would be the finding of experimental data for such things as pressured drop and influence of flow as well as temperature on attenuation. If no such data can be found performing these measurements oneself would be a suitable alternative. Obtaining such data would not only make for better comparisons with the prediction methods but would also make it possible to make comparisons with manufacturer data. One aspect of baffle silencers that were looked upon as a third objective was the shape of the baffles themselves. Mentioned only briefly in chapter 3, staggered baffle arrangements are one such shape. Baffles shaped as staves, called brick or rod silencers, would also be an interesting find as no information regarding them has been found during the work on this thesis. Having knowledge about various kinds of silencers creates more tools for engineers to use when reducing sound levels in ducts and exhausts. Further improvements to the thesis can be made by improving the Matlab script used for chapter 6. Modifying the script to use volume flows rather than flow velocities would create less confusing results for the pressure drop, which could be beneficial. Beyond this a new script which incorporates numerical solutions to equations from Ingard, Mechel or Ochmann & Donner would probably yield better predictions, though how advanced the script would have to be or how much effort this would take is unknown.

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APPENDIX

A – Graphs from Almgren

<u>1 - Comparison between measured and calculated values, baffle dimensions in image</u>

TM V-78.108 bilaga 2 sidan 1(10)

PROGRAM BF8010
ADK9.M ALMGREN EDITION NO.2. JULY 1978

SOUND ATTENUATION IN SPLITTER SILENCER
WITH PARALLELL BAFFLES

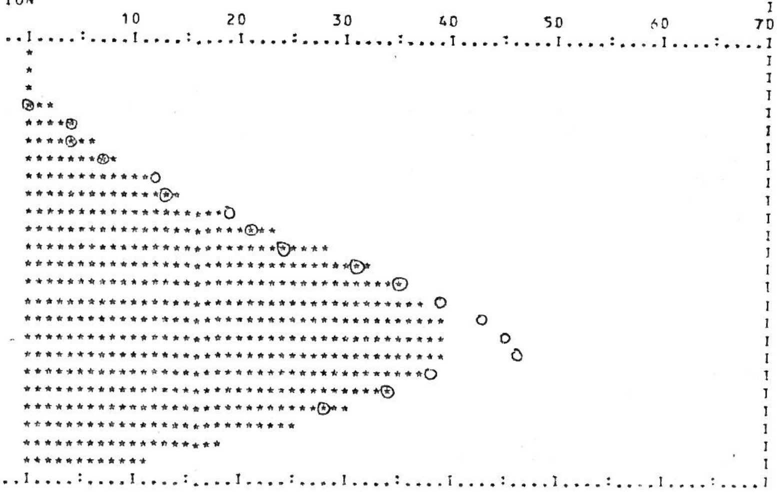
TODAY IS: 1978-07-17

TEST 11

DISCHARGE SILENCER

O = uppmätt

FREQUENCY (HZ)	ATTENUATION (DB)
50	0.0
63	0.0
80	0.0
100	1.8
125	3.5
160	5.6
200	8.0
250	10.8
315	14.2
400	18.1
500	23.1
630	27.9
800	31.9
1K	35.1
1.25K	37.3
1.6K	38.6
2K	38.9
2.5K	38.5
3.15K	36.6
4K	33.6
5K	29.8
6.3K	24.8
8K	18.2
10K	10.8



DESIGN: WIDTH = 1.000 M
HEIGHT = .300 M
LENGTH = 1.200 M

5 PARALLELL BAFFLES WITH THICKNESS: 100. MM
AND WITH SEPARATION DISTANCE: 100.MM

OPEN AREA = 50.0 PERCENT
GAS TEMPERATURE = 20.0 DEG. CENTIGRADE
VOLUME FLOW RATE = 0.0 M3/S
GAS VELOCITY = 0.0 M/S

APPROXIMATE PRESSURE LOSS (FOR AIR AT 1 ATM)
IN SILENCER, WHEN CONNECTED TO DUCT WITH THE
SAME HEIGHT AND WIDTH, HAS BEEN CALCULATED TO: 0.0 PA

2 - Comparison between measured and calculated attenuation, baffle dimensions in image

```
                         TM V-78.108                                   bilaga 2   sidan 3

PROGRAM BF8010
ADK9,M ALMGREN EDITION NO.2, JULY  1978

SOUND ATTENUATION IN SPLITTER SILENCER
WITH PARALLELL BAFFLES

TODAY IS: 1978-07-17

TEST 12
                                                  O = uppmätt
DISCHARGE SILENCER

IIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIII
I FREQUENCY I ATTENUATION                                                                    I
I   (HZ)    I    (DB)            10        20        30        40        50        60        70
I...........I...........I....:....I....:....I....:....I....:....I....:....I....:....I....:....I
I   50      I     0.0   *                                                                    I
I   63      I     0.0   *                                                                    I
I   80      I     0.0   *                                                                    I
I   100     I     1.8   ***O                                                                 I
I   125     I     3.5   ****                                                                 I
I   160     I     5.6   ******                                                               I
I   200     I     8.0   ********                                                             I
I   250     I    10.8   ***********                                                          I
I   315     I    14.2   **************                                                       I
I   400     I    18.1   ******************                                                   I
I   500     I    23.1   ***********************                                              I
I   630     I    27.9   ****************************                                         I
I   800     I    31.9   ********************************  O                                  I
I   1K      I    35.1   *********************************** O                                I
I   1.25K   I    37.3   *************************************  O                             I
I   1.6K    I    38.6   *************************************** O                            I
I   2K      I    38.9   ***************************************   O                          I
I   2.5K    I    38.5   *************************************** O                            I
I   3.15K   I    36.6   *************************************                                I
I   4K      I    33.6   **********************************                                   I
I   5K      I    29.8   ******************************                                       I
I   6.3K    I    24.8   *************************                                            I
I   8K      I    18.2   ******************                                                   I
I   10K     I    10.8   ***********                                                          I
I.......................I....:....I....:....I....:....I....:....I....:....I....:....I....:....I

DESIGN: WIDTH  =  1.000 M
        HEIGHT =  1.000 M
        LENGTH =  1.200 M

         5 PARALLELL BAFFLES WITH THICKNESS: 100. MM
        AND WITH SEPARATION DISTANCE:  100.MM

OPEN AREA          =  50.0 PERCENT
GAS TEMPERATURE    =  20.0 DEG. CENTIGRADE
VOLUME FLOW RATE =    0.0 M3/S
GAS VELOCITY       =    0.0 M/S

APPROXIMATE PRESSURE LOSS (FOR AIR AT 1 ATM)
IN SILENCER, WHEN CONNECTED TO DUCT WITH THE
SAME HEIGHT AND WIDTH, HAS BEEN CALCULATED TO:      0.0 PA
```

3 - Comparison between measured and calculated attenuation, baffle dimensions in image

TM V-78.108 bilaga 2 sidan 3

```
PROGRAM BFR010
ADK9.M ALMGREN EDITION NO.2, JULY 1978

SOUND ATTENUATION IN SPLITTER SILENCER
WITH PARALLELL BAFFLES

TODAY IS: 1978-07-17

TEST 13
                                             O = uppmätt
DISCHARGE SILENCER

IIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIII
I FREQUENCY I ATTENUATION                                                                          I
I   (HZ)    I    (DB)             10        20        30        40        50        60        70
I...........I...........I....:....I....:....I....:....I....:....I....:....I....:....I....:....I
I   50      I     0.0    *                                                                      I
I   63      I     0.0    *                                                                      I
I   80      I     0.0    *                                                                      I
I   100     I      .3    *                                                                      I
I   125     I     1.3    **                                                                     I
I   160     I     2.3    ***                                                                    I
I   200     I     3.6    ****                                                                   I
I   250     I     5.1    ******                                                                 I
I   315     I     7.0    *******                                                                I
I   400     I     9.4    **********                                                             I
I   500     I    11.9    ************                                                           I
I   630     I    13.7    **************                                                         I
I   800     I    15.1    ****************                                                       I
I   1K      I    16.0    ****************                                                       I
I   1.25K   I    16.2    *****************                                                      I
I   1.6K    I    15.7    ****************                                                       I
I   2K      I    14.5    ***************                                                        I
I   2.5K    I    12.5    *************                                                          I
I   3.15K   I     9.3    **********                                                             I
I   4K      I     5.8    *******                                                                I
I   5K      I     2.7    ****                                                                   I
I   6.3K    I     0.0    *                                                                      I
I   8K      I     0.0    *                                                                      I
I   10K     I     0.0    *                                                                      I
I.......................I....:....I....:....I....:....I....:....I....:....I....:....I....:....I

DESIGN: WIDTH  =   .999 M
        HEIGHT =   .600 M
        LENGTH =  1.200 M

          3 PARALLELL BAFFLES WITH THICKNESS: 100. MM
          AND WITH SEPARATION DISTANCE:  233.MM

OPEN AREA          =  70.0 PERCENT
GAS TEMPERATURE    =  20.0 DEG. CENTIGRADE
VOLUME FLOW RATE   =   0.0 M3/S
GAS VELOCITY       =   0.0 M/S

APPROXIMATE PRESSURE LOSS (FOR AIR AT 1 ATM)
IN SILENCER, WHEN CONNECTED TO DUCT WITH THE
SAME HEIGHT AND WIDTH, HAS BEEN CALCULATED TO:     0.0 PA
```

4 - Comparison between calculated and measured attenuation, baffle dimensions in image

TM V-78.108

bilaga 2 sidan 6

PROGRAM BFR010
ADK9.M ALMGREN EDITION NO.2. JULY 1978

SOUND ATTENUATION IN SPLITTER SILENCER
WITH PARALLELL BAFFLES

TODAY IS: 1978-07-17

TEST 16

DISCHARGE SILENCER

O = uppmätt

FREQUENCY (HZ)	ATTENUATION (DB)
50	.3
63	2.5
80	4.7
100	6.9
125	9.7
160	12.7
200	16.6
250	21.1
315	26.6
400	34.0
500	42.9
630	52.9
800	62.1
1K	69.6
1.25K	76.0
1.6K	81.1
2K	85.5
2.5K	88.2
3.15K	89.3
4K	89.6
5K	87.8
6.3K	84.3
8K	78.4
10K	70.4

Scale (DB): 10 20 30 40 50 60 70

DESIGN: WIDTH = 1.001 M
HEIGHT = .600 M
LENGTH = 1.200 M

7 PARALLELL BAFFLES WITH THICKNESS: 100. MM
AND WITH SEPARATION DISTANCE: 43.MM

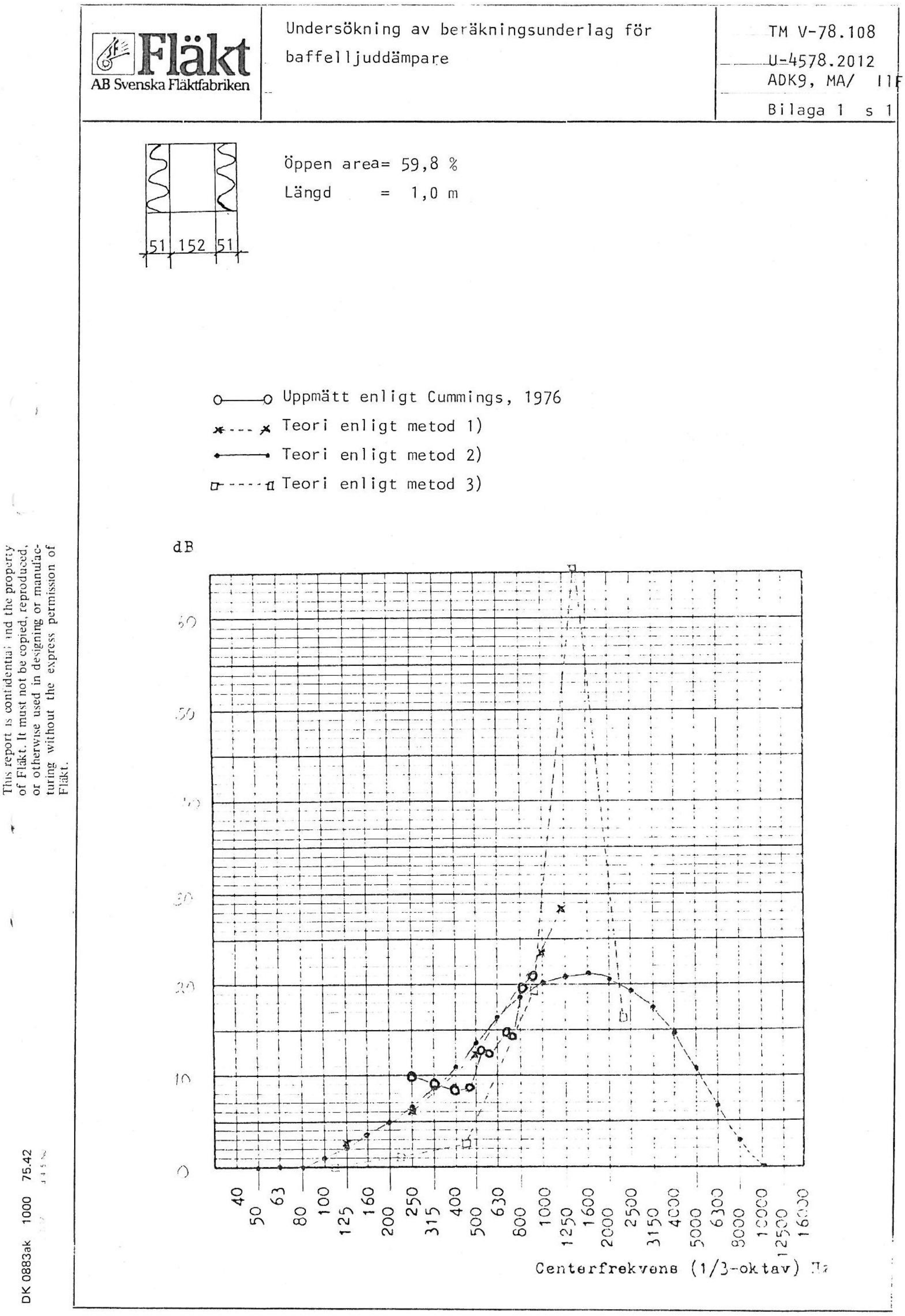
OPEN AREA = 30.1 PERCENT
GAS TEMPERATURE = 20.0 DEG. CENTIGRADE
VOLUME FLOW RATE = 0.0 M3/S
GAS VELOCITY = 0.0 M/S

APPROXIMATE PRESSURE LOSS (FOR AIR AT 1 ATM)
IN SILENCER, WHEN CONNECTED TO DUCT WITH THE
SAME HEIGHT AND WIDTH, HAS BEEN CALCULATED TO: 0.0 PA

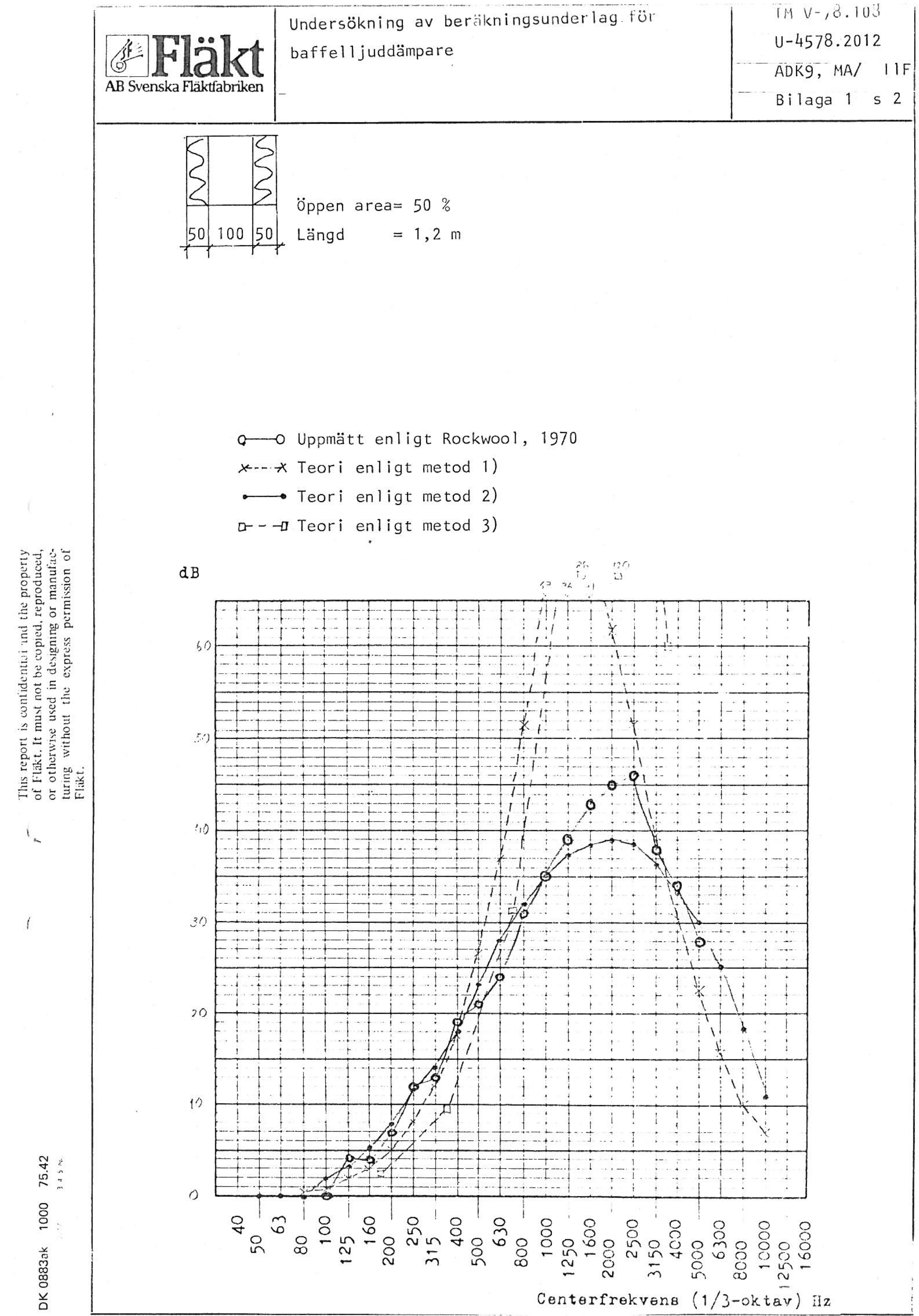
5 - Comparison between calculated and measured attenuation, baffle dimensions in image

```
                                                  TM V-78.108                      bilaga 2  sidan 8
 PROGRAM BFR010
ADK9,M ALMGREN EDITION NO.2, JULY  1978

SOUND ATTENUATION IN SPLITTER SILENCER
WITH PARALLELL BAFFLES

TODAY IS: 1978-07-17

TEST 20
                                                 O = uppmätt
DISCHARGE SILENCER

IIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIIII
I FREQUENCY I ATTENUATION                                                                    I
I   (HZ)    I   (DB)              10        20        30        40        50        60        70
I...........I...........I....:....I....:....I....:....I....:....I....:....I....:....I....:....I
I   50      I     .5    **                                                                   I
I   63      I    1.2    **                                                                   I
I   80      I    2.0    ***                                                                  I
I   100     I    2.8    ****                                                                 I
I   125     I    3.9    *****                                                                I
I   160     I    5.1    *****                                                                I
I   200     I    6.5    *******                                                              I
I   250     I    8.0    *********                                                            I
I   315     I    9.8    **********                                                           I
I   400     I   11.3    ***********                                                          I
I   500     I   12.2    **************                                                       I
I   630     I   12.7    ***************                                                      I
I   800     I   12.8    **************                                                       I
I   1K      I   12.5    ***************                                                      I
I   1.25K   I   11.7    *************                                                        I
I   1.6K    I   10.6    ************                                                         I
I   2K      I    9.0    **********                                                           I
I   2.5K    I    6.7    ********                                                             I
I   3.15K   I    4.2    *****                                                                I
I   4K      I    1.9    ***                                                                  I
I   5K      I    0.0    *                                                                    I
I   6.3K    I    0.0    *                                                                    I
I   8K      I    0.0    *                                                                    I
I   10K     I    0.0    *                                                                    I
I.......................I....:....I....:....I....:....I....:....I....:....I....:....I....:....I

DESIGN: WIDTH  =  1.000 M
        HEIGHT =   .600 M
        LENGTH =  1.200 M

         2 PARALLELL BAFFLES WITH THICKNESS: 200. MM
        AND WITH SEPARATION DISTANCE:  300.MM

OPEN AREA          =  60.0 PERCENT
GAS TEMPERATURE    =  20.0 DEG. CENTIGRADE
VOLUME FLOW RATE   =   0.0 M3/S
GAS VELOCITY       =   0.0 M/S

APPROXIMATE PRESSURE LOSS (FOR AIR AT 1 ATM)
IN SILENCER, WHEN CONNECTED TO DUCT WITH THE
SAME HEIGHT AND WIDTH, HAS BEEN CALCULATED TO:     0.0 PA
```

6 – Comparison calculation by Almgren. Circle line represents measurement, dotted line represents Emblton’s method. 60% open area and baffle length of 1m. Spacing and baffle width in image

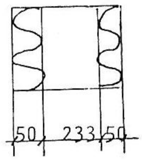


7 - Comparison calculation by Almgren. Circle line represents measurement, dotted line represents Emblton's method. 50% open area and baffle length of 1.2m. Spacing and baffle width in image



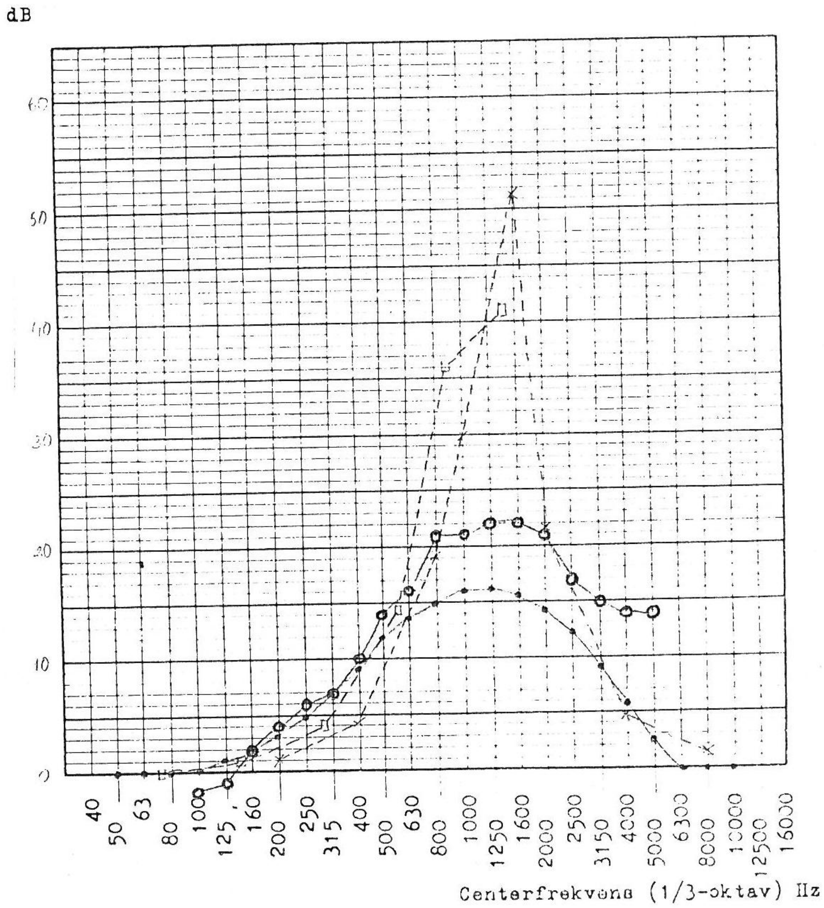
8 - Comparison calculation by Almgren. Circle line represents measurement, dotted line represents Emblton’s method. 70% open area and baffle length of 1.2m. Spacing and baffle width in image

Fläkt AB Svenska Fläktfabriken	Undersökning av beräkningsunderlag för baffelljuddämpare	TM V-78.108 U-4578.2012 ADK9, MA/ 11F Bilaga 1 s 3



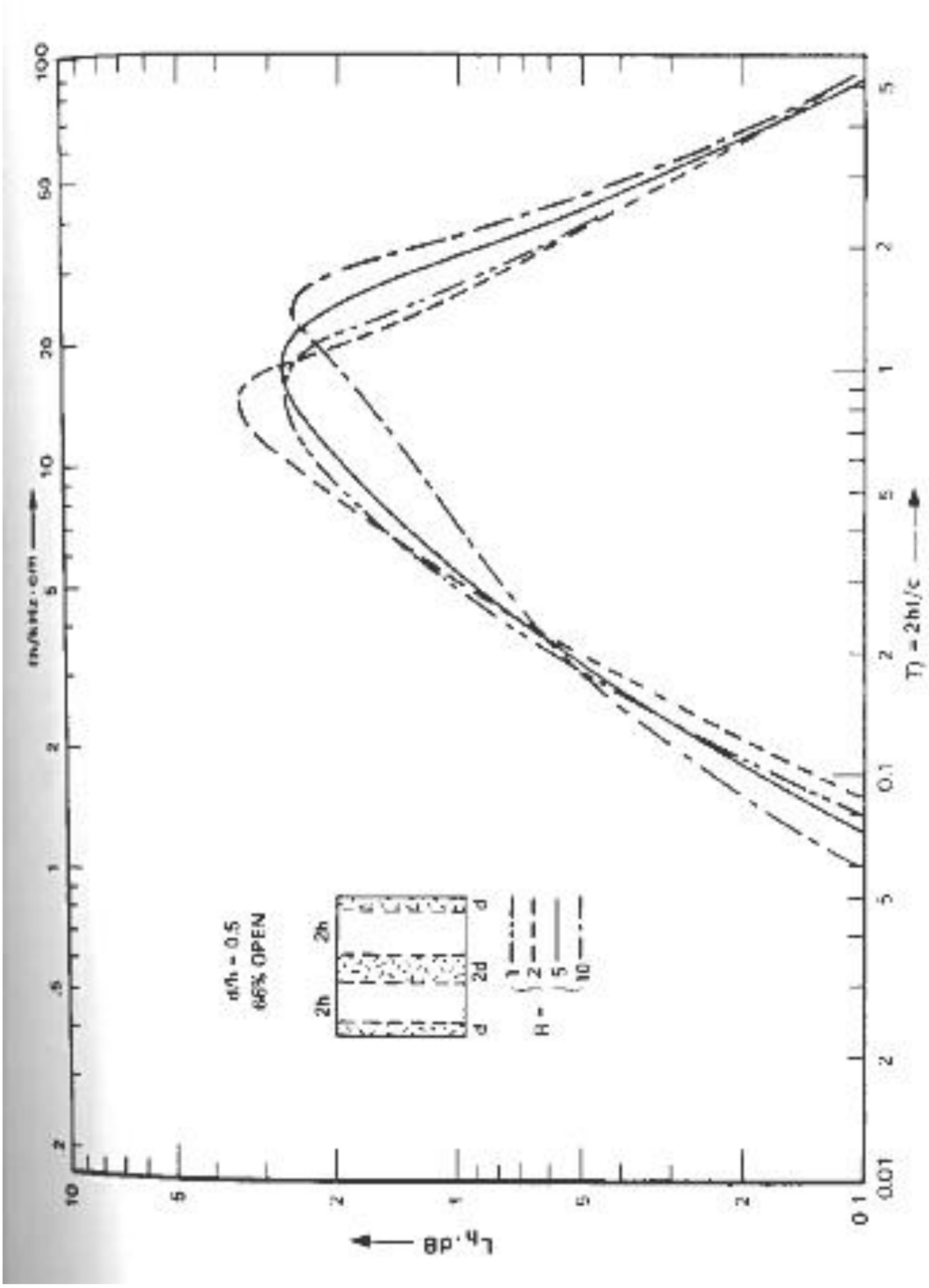
Öppen area = 70 %
Längd = 1,2 m

o——o Uppmätt enligt Rockwool, 1970
x- - - -x Teori enligt metod 1) (75 % öppen area)
•——• Teori enligt metod 2)
□- - -□ Teori enligt metod 3)

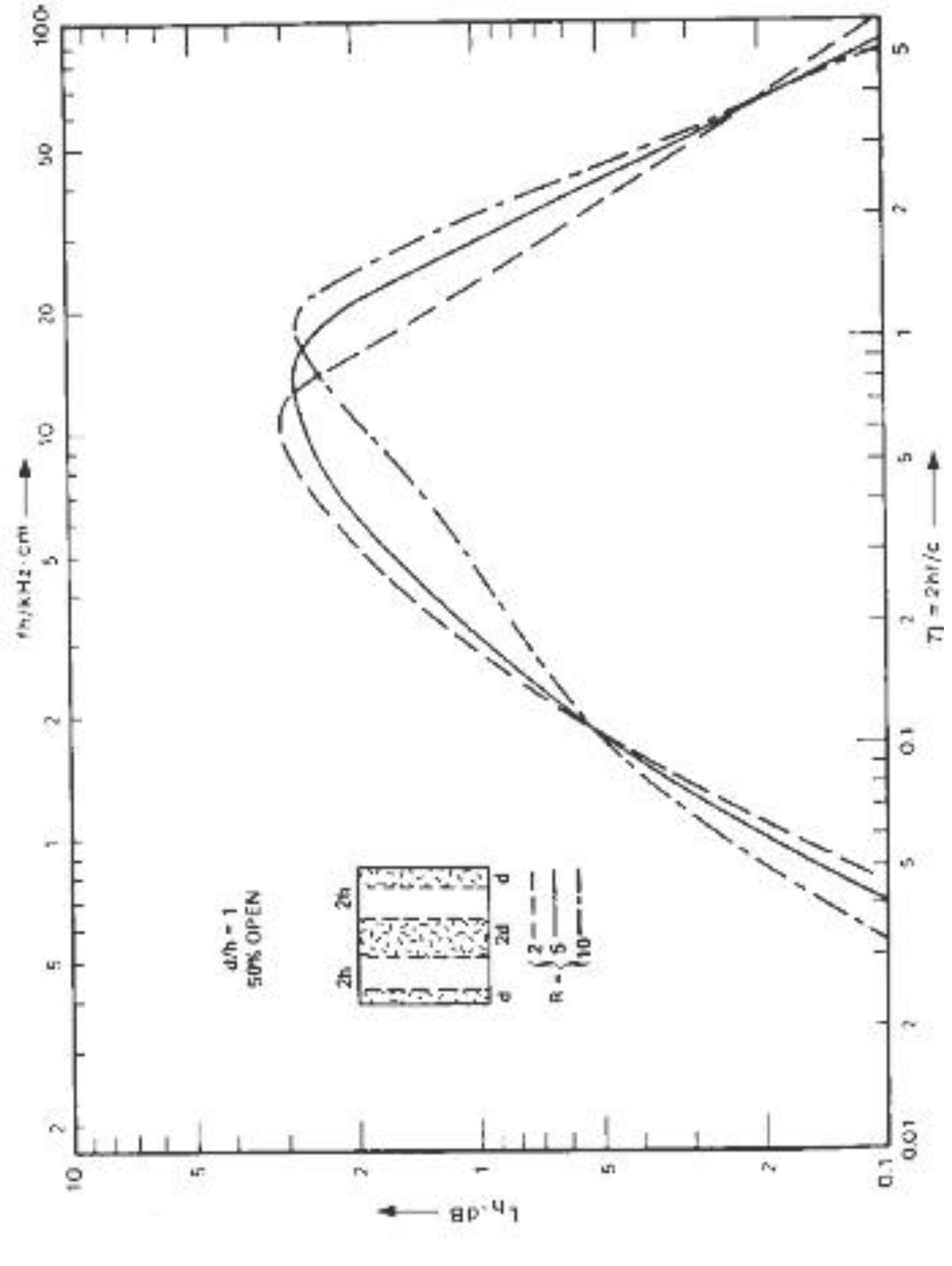


B - Graph from Galaitsis and Vér

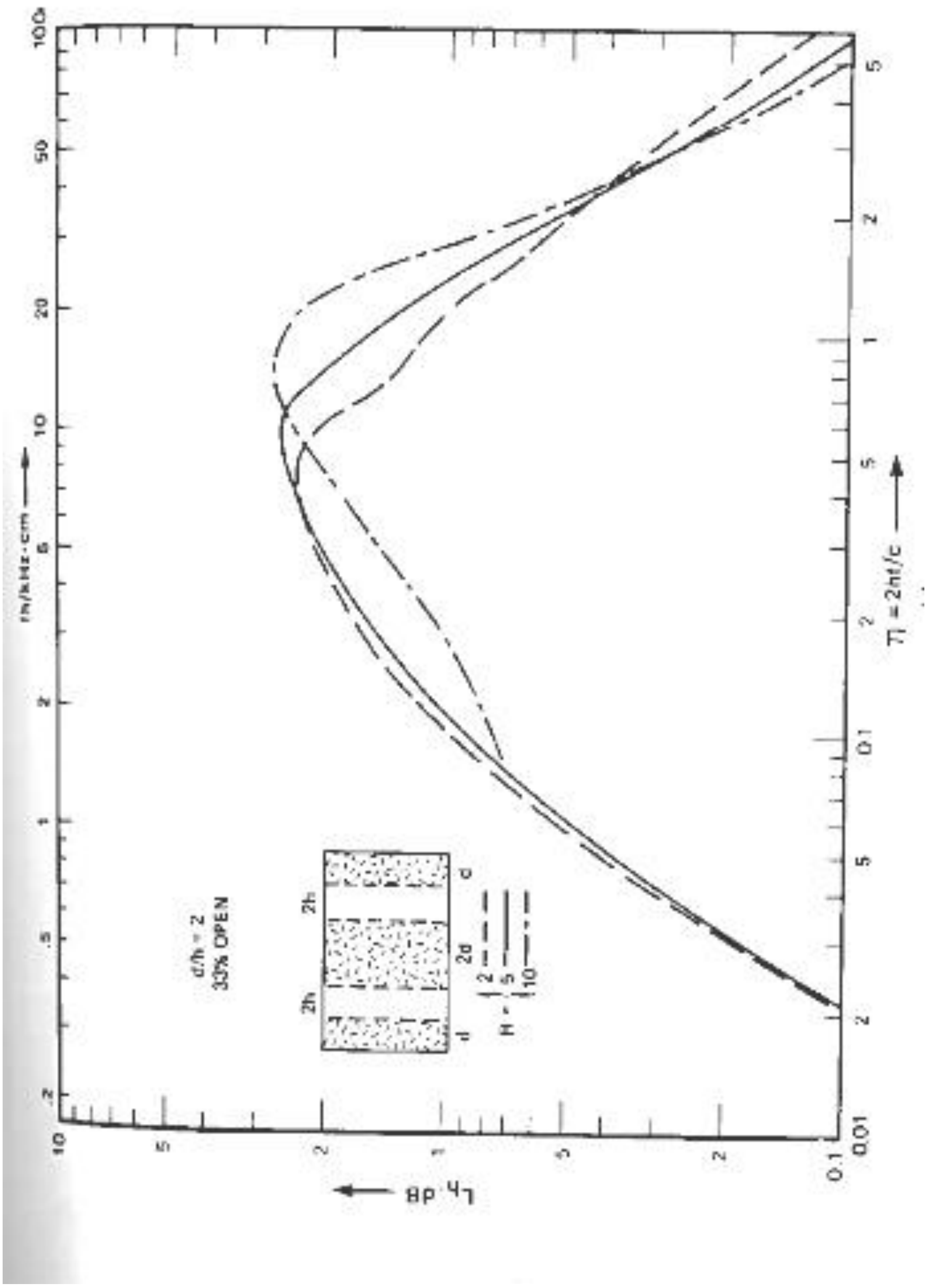
1 - Graph for predicting baffle attenuation, for use with 66% open area



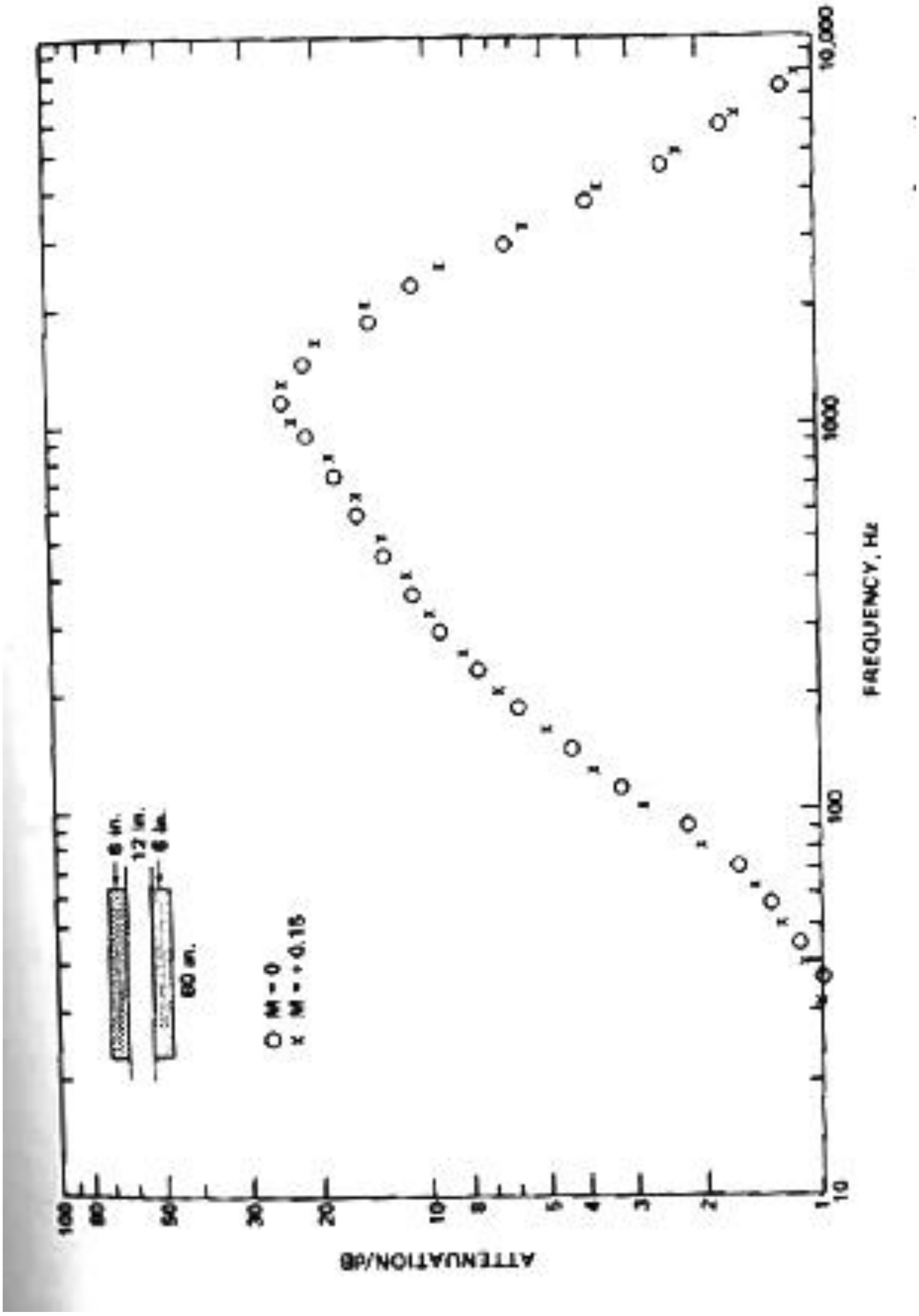
2 - Graph for predicting baffle attenuation, for use with 50% open area



<u>3 - Graph for predicting baffle attenuation, for use with 33% open area</u>



4 - Graph comparing computed and shifted values for attenuation influenced by flow, (x) computed (o) shifted



C – Matlab script

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%   Attenuation calculation program  %
%       by Dario Bogdanovic          %
%   Latest day changed: 2014-09-09   %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%------------------------------------%
close all
clear all
clc

% Parameters for baffle dimension, temperature and flow velocity input by
% the user
T=input('Input temperature in degrees Celsius \n');
t=input('Input width of the baffle in [m] \n');
ly=input('Input the spacing between baffles in [m] \n');
L=input('Input baffle length in [m] \n');

%Length factor to be used later
Lly=L/ly;
%Open area
OA=ly/(ly+2*(t/2));
OA=round(OA*100);
%Speed of sound
c0=343;
% Speed of sound with respect to temperature
c=c0*sqrt((273+T)/293);



%Manually inserted reduction values for each of the five curves as well as
%to corresponding horizontal axis
hor=[0.010 0.011 0.012 0.013 0.014 0.015 0.016 0.017 0.018 0.019 0.020
0.021 0.022 0.023 0.024 0.025 0.026 0.027 0.028 0.029 0.030 0.031 0.032
0.033 0.034 0.035 0.036 0.037 0.038 0.039 0.040 0.041 0.042 0.043 0.044
0.045 0.046 0.047 0.048 0.049 0.050 0.051 0.052 0.053 0.054 0.055 0.056
0.057 0.058 0.059 0.060 0.061 0.062 0.063 0.064 0.065 0.066 0.067 0.068
0.069 0.070 0.071 0.072 0.073 0.074 0.075 0.076 0.077 0.078 0.079 0.080
0.081 0.082 0.083 0.084 0.085 0.086 0.087 0.088 0.089 0.090 0.091 0.092
0.093 0.094 0.095 0.096 0.097 0.098 0.099 0.10 0.11 0.12 0.13 0.14 0.15
0.16 0.17 0.18 0.19 0.20 0.21 0.22 0.23 0.24 0.25 0.26 0.27 0.28 0.29 0.30
0.31 0.32 0.33 0.34 0.35 0.36 0.37 0.38 0.39 0.40 0.41 0.42 0.43 0.44 0.45
0.46 0.47 0.48 0.49 0.50 0.51 0.52 0.53 0.54 0.55 0.56 0.57 0.58 0.59 0.60
0.61 0.62 0.63 0.64 0.65 0.66 0.67 0.68 0.69 0.70 0.71 0.72 0.73 0.74 0.75
0.76 0.77 0.78 0.79 0.80 0.81 0.82 0.83 0.84 0.85 0.86 0.87 0.88 0.89 0.90
0.91 0.92 0.93 0.94 0.95 0.96 0.97 0.98 0.99 1.0 1.1 1.2 1.3 1.4 1.5 1.6
1.7 1.8 1.9 2.0 2.1 2.2 2.3 2.4 2.5 2.6 2.7 2.8 2.9 3.0 3.1 3.2 3.3 3.4 3.5
3.6 3.7 3.8 3.9 4.0];
Aly_80=[0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0.05 0.15 0.25 0.35 0.45 0.55
0.6 0.65 0.70 0.75 0.80 0.85 0.90 0.95 1.0 1.05 1.1 1.14 1.18 1.22 1.26 1.3
1.35 1.40 1.45 1.50 1.55 1.60 1.65 1.70 1.75 1.8 1.84 1.88 1.92 1.96 2 2.05
2.10 2.15 2.20 2.25 2.28 2.31 2.34 2.37 2.4 2.44 2.48 2.52 2.56 2.6 2.64
2.68 2.72 2.76 2.8 2.82 2.84 2.86 2.88 2.9 2.92 2.94 2.96 2.98 3 3.01 3.02
3.03 3.04 3.05 3.06 3.07 3.08 3.09 3.1 3.105 3.11 3.115 3.12 3.125 3.13
3.135 3.14 3.145 3.15 3.20 3.17 3.15 3.12 3.05 3.0 2.75 2.6 2.45 2.15 2.015
1.88 1.745 1.61 1.475 1.34 1.205 1.07 0.935 0.8 0.74 0.68 0.62 0.56 0.5
0.44 0.38 0.32 0.26 0.2];
Aly_67=[0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
0 0 0 0 0 0 0 0.015 0.030 0.045 0.060 0.075 0.090 0.105 0.120 0.135 0.15
```

```
0.16 0.17 0.18 0.19 0.20 0.21 0.22 0.23 0.24 0.25 0.26 0.27 0.28 0.29 0.30
0.31 0.32 0.33 0.34 0.35 0.36 0.37 0.38 0.39 0.40 0.41 0.42 0.43 0.44 0.45
0.46 0.47 0.48 0.49 0.50 0.51 0.52 0.53 0.54 0.55 0.60 0.66 0.73 0.79 0.85
0.95 1.05 1.15 1.25 1.35 1.44 1.53 1.62 1.71 1.8 1.9 2.0 2.1 2.2 2.3 2.36
2.42 2.48 2.54 2.6 2.65 2.70 2.75 2.80 2.85 2.88 2.91 2.94 2.97 3.0 3.02
3.04 3.06 3.08 3.1 3.11 3.12 3.13 3.14 3.15 3.16 3.17 3.18 3.19 3.2 3.205
3.21 3.215 3.22 3.225 3.23 3.235 3.24 3.245 3.25 3.245 3.24 3.235 3.23
3.225 3.22 3.215 3.21 3.205 3.2 3.195 3.190 3.185 3.180 3.175 3.170 3.165
3.160 3.155 3.15 3.1425 3.135 3.1275 3.12 3.1125 3.105 3.0975 3.09 3.0825
3.075 3.05 3.0 2.92 2.84 2.75 2.59 2.43 2.27 2.11 1.95 1.835 1.72 1.605
1.49 1.375 1.26 1.145 1.03 0.915 0.8 0.74 0.68 0.62 0.56 0.5 0.44 0.38 0.32
0.26 0.2];
Aly_50=[0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0.05 0.1 0.12 0.14 0.15 0.2 0.2
0.22 0.24 0.25 0.27 0.29 0.31 0.33 0.35 0.37 0.38 0.39 0.39 0.40 0.42 0.44
0.46 0.48 0.50 0.52 0.54 0.56 0.58 0.60 0.62 0.64 0.66 0.68 0.70 0.71 0.72
0.73 0.74 0.75 0.77 0.79 0.81 0.83 0.85 0.86 0.87 0.88 0.89 0.90 0.91 0.92
0.93 0.94 0.95 0.97 0.99 1.01 1.03 1.05 1.06 1.07 1.08 1.09 1.10 1.11 1.12
1.13 1.14 1.15 1.17 1.19 1.21 1.23 1.25 1.35 1.45 1.55 1.75 1.9 2.0 2.1
2.15 2.30 2.40 2.45 2.50 2.55 2.63 2.70 2.75 2.80 2.85 2.90 2.95 2.97 2.99
3.01 3.03 3.05 3.08 3.11 3.14 3.17 3.2 3.205 3.210 3.215 3.220 3.225 3.230
3.235 3.240 3.245 3.25 3.25 3.25 3.25 3.25 3.25 3.25 3.25 3.25 3.25 3.25
3.25 3.25 3.245 3.245 3.24 3.23 3.22 3.21 3.205 3.2 3.2 3.19 3.19 3.18 3.18
3.17 3.17 3.16 3.16 3.15 3.14 3.13 3.12 3.11 3.10 3.09 3.08 3.07 3.06 3.05
3.05 3.04 3.04 3.03 3.03 3.02 3.02 3.01 3.01 3.0 2.9 2.85 2.7 2.6 2.55 2.4
2.3 2.2 2.05 1.9 1.76 1.62 1.48 1.34 1.25 1.17 1.10 1.03 0.9 0.8 0.74 0.68
0.62 0.56 0.5 0.44 0.38 0.32 0.26 0.2];
Aly_33=[0.1 0.12 0.14 0.16 0.18 0.2 0.23 0.26 0.29 0.32 0.35 0.37 0.39 0.41
0.43 0.45 0.48 0.51 0.54 0.57 0.6 0.625 0.65 0.675 0.7 0.725 0.75 0.775 0.8
0.825 0.85 0.875 0.9 0.925 0.95 0.975 1.0 1.025 1.05 1.075 1.1 1.125 1.15
1.175 1.2 1.225 1.25 1.275 1.3 1.325 1.35 1.375 1.4 1.425 1.45 1.475 1.5
1.525 1.55 1.575 1.6 1.62 1.64 1.66 1.68 1.7 1.72 1.74 1.76 1.78 1.8 1.82
1.84 1.86 1.88 1.9 1.92 1.94 1.96 1.98 2 2.01 2.02 2.03 2.04 2.05 2.06 2.07
2.08 2.09 2.1 2.19 2.28 2.37 2.46 2.55 2.62 2.69 2.76 2.83 2.9 2.92 2.94
2.96 2.98 3 3.03 3.06 3.09 3.12 3.15 3.155 3.16 3.165 3.17 3.175 3.18 3.185
3.19 3.195 3.2 3.2 3.2 3.2 3.2 3.2 3.2 3.2 3.2 3.2 3.2 3.1975 3.195 3.1925
3.19 3.1875 3.185 3.1825 3.18 3.1775 3.175 3.1675 3.16 3.1525 3.145 3.1375
3.13 3.1225 3.115 3.1075 3.1 3.095 3.09 3.085 3.08 3.075 3.07 3.065 3.06
3.055 3.05 3.04 3.03 3.02 3.01 3.0 2.99 2.98 2.97 2.96 2.95 2.94 2.93 2.92
2.91 2.9 2.89 2.88 2.87 2.86 2.85 2.76 2.67 2.58 2.49 2.4 2.27 2.14 2.01
1.88 1.75 1.655 1.56 1.465 1.37 1.275 1.18 1.085 0.99 0.895 0.8 0.74 0.68
0.62 0.56 0.5 0.44 0.38 0.32 0.26 0.2];
Aly_17=[0.6 0.625 0.65 0.675 0.7 0.725 0.75 0.7750 0.8 0.825 0.85 0.88 0.91
0.94 0.97 1.0 1.03 1.06 1.09 1.12 1.15 1.175 1.2 1.225 1.25 1.275 1.3 1.325
1.35 1.375 1.4 1.42 1.44 1.46 1.48 1.5 1.52 1.54 1.56 1.58 1.6 1.625 1.65
1.675 1.7 1.725 1.75 1.775 1.8 1.825 1.85 1.87 1.89 1.91 1.93 1.95 1.97
1.99 2.01 2.03 2.05 2.07 2.09 2.11 2.13 2.15 2.17 2.19 2.21 2.23 2.25 2.265
2.28 2.295 2.31 2.325 2.34 2.355 2.37 2.385 2.4 2.41 2.42 2.43 2.44 2.45
2.46 2.47 2.48 2.49 2.5 2.56 2.62 2.68 2.74 2.8 2.84 2.88 2.92 2.96 3 3.02
3.04 3.06 3.08 3.1 3.12 3.14 3.16 3.18 3.2 3.205 3.21 3.215 3.22 3.225 3.23
3.235 3.24 3.245 3.25 3.245 3.24 3.235 3.23 3.225 3.22 3.215 3.21 3.205 3.2
3.19 3.18 3.17 3.16 3.15 3.14 3.13 3.12 3.11 3.1 3.09 3.08 3.07 3.06 3.05
3.04 3.03 3.02 3.01 3.0 2.99 2.98 2.97 2.96 2.95 2.94 2.93 2.92 2.91 2.9
2.89 2.88 2.87 2.86 2.85 2.84 2.83 2.82 2.81 2.8 2.79 2.78 2.77 2.76 2.75
2.74 2.73 2.72 2.71 2.7 2.61 2.52 2.43 2.34 2.25 2.13 2.01 1.89 1.77 1.65
1.565 1.48 1.395 1.31 1.225 1.14 1.055 0.97 0.885 0.8 0.74 0.68 0.62 0.56
0.5 0.44 0.38 0.32 0.26 0.2];


% The interpolation is made with the 'griddata' command, the following rows
% of code are preperation for the command
HORT=transpose(hor);
```

```
horid=[HORT;HORT];              %Transposed ly/lambda vector
ALY17T=transpose(Aly_17);       %Transponing the attenuation curves
ALY33T=transpose(Aly_33);
ALY50T=transpose(Aly_50);
ALY67T=transpose(Aly_67);
ALY80T=transpose(Aly_80);
lin17=ones(length(hor),1)*17;   %Vertical vectors with the same value, used
to identify the interpolation limits
lin33=ones(length(hor),1)*33;
lin50=ones(length(hor),1)*50;
lin67=ones(length(hor),1)*67;
lin80=ones(length(hor),1)*80;

%The script now checks inbetween which two preexisting curves the current
%open are percentage is located. It then interpolates all curves between
%these two curves and selects the one that is correct

if 17 <= OA & OA < 33
    disp('Open area is'),disp(OA)
    Alyid=[ALY17T;ALY33T];                  %Determine the two attenuation
curves used
    linid=[lin17;lin33];                    %Identify the limits
    [TI,XI]=meshgrid(17:33, 0.01:0.001:4);  %Create a grid of points
covering the area in which the curves are located
    YI=griddata(linid,horid,Alyid,TI,XI);   %Interpolate using 'griddata'
to obtain all curves between the limiting
    for k=1:17                              %Extract the correct curve
        if TI(1,k) ==OA
            var=k;
        end
    end
elseif 33 <= OA & OA < 50
    disp('Open area is'),disp(OA)
    Alyid=[ALY33T;ALY50T];
    linid=[lin33;lin50];
    [TI,XI]=meshgrid(33:50, 0.01:0.001:4);
    YI=griddata(linid,horid,Alyid,TI,XI);
    for k=1:18
        if TI(1,k) ==OA
            var=k;
        end
    end
elseif 50 <= OA & OA < 67
    disp('Open area is'),disp(OA)
    Alyid=[ALY50T;ALY67T];
    linid=[lin50;lin67];
    [TI,XI]=meshgrid(50:67, 0.01:0.001:4);
    YI=griddata(linid,horid,Alyid,TI,XI);
    for k=1:18
        if TI(1,k) ==OA
            var=k;
        end
    end
elseif 67 <= OA & OA < 80
    disp('Open area is'),disp(OA)
    Alyid=[ALY67T;ALY80T];
    linid=[lin67;lin80];
    [TI,XI]=meshgrid(67:80, 0.01:0.001:4);
    YI=griddata(linid,horid,Alyid,TI,XI);
    for k=1:14
        if TI(1,k) ==OA
```

```
            var=k;
        end
    end
else
    disp('Bad input dimensions, try again')
    break
end

f=XI(:,var)*c/ly;
Attenuation=Lly*YI(:,var);
figure(1)
semilogx(f,Attenuation,'Linewidth',2)
xlabel('Frequency','Fontsize',14)
ylabel('Attenuation','Fontsize',14)

y=1;
n=0;
cont=input('Would you like to calculate pressure drop and add corrections
for flow and self noise? \nThis will require the following input: \nFlow
velocity between the baffles in [m/s] \nDuct width in [m] \nDuct height in
[m] \nDensity of the gas in the duct in [kg/m^3]. Normal value for air=1,2
kg/m^3 \ny/n (use lowercase) \n');

if y==cont
    %Input for the self noise parameters
    flow=input('Input flow velocity between baffles in [m/s], \npositive
for downstream and negative for upstream \n');
    duct_width=input('Input the width of the baffle duct in [m] \n');
    duct_height=input('Input baffle height in [m] \n');
    %Input for the pressure drop
    air_den=input('Input density of the gas in [kg/m^3], normal value for
air=1.2 kg/m^3) \n');

    %The flow from the input is the flow between baffles, the volume flow
    %of the duct has to be calculated in order to obtain the face velocity
    %of the baffles

    %First the volume flow through between two baffles is calculated
    channel_area=ly*duct_height;
    v_flow_channel=flow*channel_area;
    %With the volume flow in a channel the total volumeflow of the duct is
    %calculated
    v_flow=v_flow_channel/(OA/100);

    %Face area of the baffles
    facea=duct_height*duct_width*OA/100;
    %Face area of the open area between the baffles
    faceoa=duct_width*duct_height-facea;
    %Circumference of a channel between two baffles
    circum=2*ly+2*duct_height;
    %Face velocity of the baffles
    facev=abs(v_flow/facea);

    %Mach number
    M_num=flow/c0;
    if M_num > 0.3
        disp('Flow too large, cannot compute')
        break
    elseif M_num < -0.3
        disp('Flow too large, cannot compute')
```

```
        break
    end

    %Correction scalar
    flow_corr=1-1.5*M_num+M_num^2;
    %Corrected attenuation
    Flow_corrected_attenuation=Attenuation*flow_corr;

    figure (2)
    semilogx(f,Flow_corrected_attenuation,'Linewidth',2)
    xlabel('Frequency','Fontsize',14)
    ylabel('Attenuation','Fontsize',14)
    %Self noise generated in sound power

    Lw_self=8.4+55*log10(facev)+10*log10(facea)-45*log10(OA/100)-
25*log10(273+T/293);
    disp('Generated sound power'),disp(Lw_self)

    %Calculate pressure drop
    disp('Pressure drop')

pressd=0.1*air_den*flow^2/2+0.05*L*flow^2*air_den/(2*4*(faceoa/circum))+air
_den*(flow-(v_flow/(duct_width*duct_height)))^2/2;
    disp(pressd)
    disp('All done')
else
    disp('All done')
end
```