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Acoustic validation of calculation software for ducts, panels and room acoustics

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ABSTRACT

Alara-Lukagro is a project based manufacturer of acoustic enclosures, components and attenuators. These are often placed in projects where they are part of a number of components that have a strong effect on the acoustics. In order to predict such a variety of situations software was developed in line with international noise mitigation standards. The attenuation values are based on laboratory measurements and measurements in situ. Low frequency attenuation rarely results in shortcomings, a larger spread in results is found in the high frequency range.. This paper shows how the attenuation is determined on a geometric chain model, and how it can be engineered in order to design a good attenuator in complex realistic acoustic environments.

Keywords: Attenuators, Directivity dependence, Noise attenuations

1. INTRODUCTION

As a manufacturer of acoustic solutions, the received information is limited. Mostly however this information satisfies to calculate simple and complex projects based on calculation rules and test results from laboratories. Noise requirements have become more stringent and there is a lot of competition. Due to variable rotation velocities and the way noise source data is often only available in octave band values, splitter silencers are often a more suitable option than resonator silencers. Fitting a broadband solution to a set of data and geometries leads to a design meant to beat the competition. How certain is this outcome, and does the wat we calculate lead to systematic errors that can or should be avoided in order to make better predictions on the end result of a noise control design? The evaluation considers a number of aspects:

- How accurate can absorption be predicted?
- How well does the division into near field and far field Eyring(1) work?
- How accurate is data provided by the suppliers?
- How well can transmission loss from laboratories be compared with transmission loss in the field?

2. Model

2.1 General design

The main philosophy is to use a standard calculation program for critical paths and for bulks. The calculation program for critical paths is most interesting. It creates a chain of steps, that can be mere algebraic, or unfold from 1D into 3D (room) acoustic models, always respecting calculation rules, room acoustics and the energy balance. Some boundary assumptions for reversed sound apply.

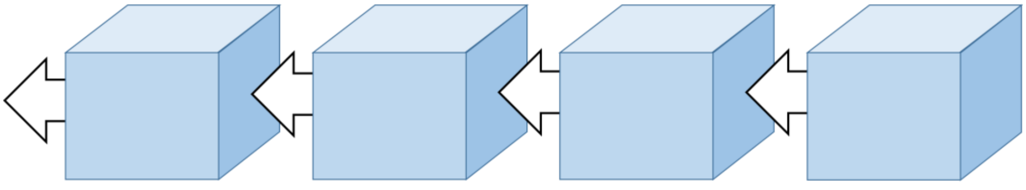

Figure 1 – Chain of elements 1 D information potentially unfolded into 3D blocks

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2.2 Boundary conditions

In the current model, sound that is reflected back into the source channel opening is assumed to disappear completely. Multiple bent reflections therefore do not lead to a theoretical increase at the receiver side. Boundary conditions and algebraic components were evaluated, but are not included in this report.

2.3 Room acoustics

The direct noise, the ground reflections and the corner or edge reflections are summed up as the direct sound. The reverberant noise is reduced by the direct source contribution, when direct sound can be calculated. The sound acting upon a surface is the sum of the reverberant and the direct sound.

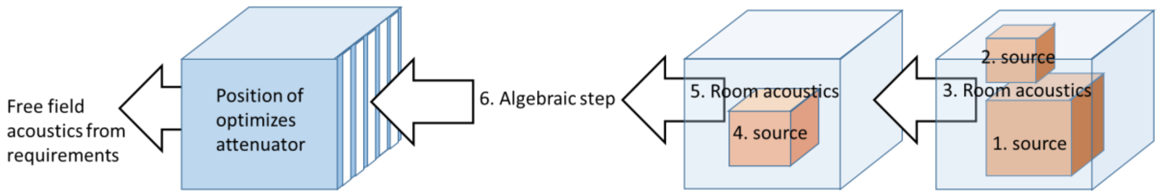

Figure 2 – Example of a model with 6 elements, optimization and free field problem

2.4 Flow generated noise

For each step (change from room to room, or channel to room, or room to channel, or room/channel into free field) the flow generated noise power per area is calculated in accordance with VDI 2081(2) 'worst case' this noise is added to the sound power. The levels are always based upon the highest average velocity and smallest area. If there is any relevant throttling, that will generate flow generated noise, the program will automatically account for it at the end of each block.

2.5 Geometric expansion

As a manufacturer, our solutions are generally evaluated at close proximity. The noise is still quite directional and is likely less under a 90° angle from the source plane than on a small angle from the radiating surface. This led to the implementation of a safety margin, This safety margin is in line with the expansion of a theoretical cube (see figure 3). Further away the curvature of noise leads to a wider expansion. Because as a manufacturer we are rarely informed about the specifics of the environment (reflections) currently there is no transition into normal halve bulb expansion (yet).

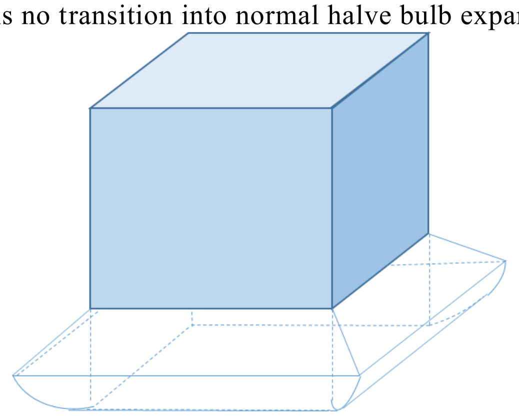

Figure 3 – Expansion in close proximity of the source

Besides expansion near field correction may apply. Because the calculation is done automatically near field correction always applies. Since there are so many steps in any given calculation, it was decided to use a more accurate continuous equation for the near field correction, which is based on a best fit with the numbers in the ILHR13.01(3).

2.6 Available elements

Numerous restraints, requirements and conditions can be set. More interesting are the currently available elements. The elements that are available in the calculation tool are:

Table 1 – Elements (phenomena)

A	Rounded bents
B	Sharp bents
C	Expansions
D	Room acoustics
E	Sound insulation elements
F	Splitter attenuators (optimization)
G	Louvres
H	Acoustic louvres
I	Sound sources (L_w, L_p)
J	Algebraic user defined equations

This means any situation that was evaluated was built up in the calculation tool only with the above mentioned elements.

2.7 Evaluated projects

A large number of projects ware evaluated. Projects that were measured after the introduction of the calculation tool were measured more extensively with the aim to carry out this evaluation. The project that were included to evaluate the evaluation tool are in table 2 (anonymized).

Table 2 - Projects

No.	description
1	Energy plant in residential area with ten 2.5 MW el. Gas engines (United Kingdom)
2	Test facility with three test cells for diesel and gas engines for large vessels (Germany)
3	Energy plant in residential area with one 2.5 MW el. Gas engine (France)
4	Heat and power mini plant 1.6 MW el.in a greenhouse (Belgium)
5	Silencer on a chimney (Netherlands)
6	Power pack test 500 kW el. (Netherlands)
7	Mobile Pump installation test (Netherlands)
8	Heat and power mini plant 1.6 MW el. In a greenhouse (Netherlands)
9	Emergency power with 9 diesel engines 2 MW el.(Germany)
10	Existing silencer on a production facility (Netherlands)
11	Enclosure around an electrical pump installation (Netherlands)
12	New silencer in an underground concrete structure (Netherlands)
13	Gas power plant with more than three 10 MW el. gas engines (Germany)
14	Power containers for emergency power with 3.5 MW el. diesel engines (Netherlands)

For privacy reasons of our customers no more details than mentioned above will be made available. This list is to illustrate the nature and the magnitude of this evaluation.

3. Evaluation

3.1 Different reasons for measurements

The projects that were evaluated were not chosen randomly. In general projects are not evaluated, unless there is a suspicion, or an agreed request before sales to do so. When the data of the evaluation is processed, this intention was to keep this distinction. In practice other factors can disturb the measurements. In order to maintain a significant amount of measurements all projects are included.

3.2 Absorption values

As a noise control solution firm a lot of projects involve big gas or diesel internal combustion engines, which typically require attenuation in the lower frequencies (31-250 Hz octave bands). The rate between direct noise radiated on acoustic elements and reverberant noise acting upon the element strongly favors the reverberant noise. So the absorption coefficient is a very important parameter. The absorption coefficient in an impedance tube is typically lower than in a reverberant room. The situation in practice is most often more like a reverberant room than an impedance tube. However the ISO 354 (4) allows to use sample sizes that increase the area of absorption to above the limit where Sabine applies (1).

Therefore it is important to determine the right absorption coefficient. In some projects the sound source couldn't be switched off, or the room was too big, or too noisy to determine reverberation times. Therefore not all noise measurements include reverberation measurements.

Initially the absorption coefficient used in the calculation tool was a safe estimate based on the Sabine absorption coefficient according to ISO 354 (4). Improvement by using measured Eyring absorption values already resulted in a 0.8 dB better prediction of the acoustic pressure in the machinery room. Taking into account that measurements were done manually and an acoustician was therefore present in the room another 0.05 dB of the total dB(A) level can be accounted for. The used average absorption with a different material property list is given in see table 3, the influence of manned measurements in table 6.

Table 3 – Eyring average shell Absorption values [-]

See Table 2	31 Hz	63 Hz	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz	8 kHz
1 old	0.10	0.13	0.31	0.59	0.66	0.73	0.73	0.69	0.69
1 new	0.09	0.16	0.26	0.41	0.44	0.55	0.61	0.64	0.59
3 old	0.03	0.03	0.11	0.13	0.14	0.18	0.17	0.15	0.14
3 new	0.03	0.04	0.10	0.12	0.14	0.17	0.17	0.18	0.18
4 old	0.11	0.15	0.32	0.61	0.69	0.76	0.76	0.73	0.73
4 new	0.12	0.22	0.31	0.46	0.48	0.58	0.64	0.67	0.62
8 old	0.11	0.15	0.32	0.61	0.69	0.76	0.76	0.73	0.73
8 new	0.12	0.22	0.31	0.46	0.48	0.58	0.64	0.67	0.62
9 old	0.05	0.05	0.04	0.05	0.06	0.08	0.07	0.06	0.07
9 new	0.05	0.05	0.05	0.08	0.09	0.10	0.08	0.07	0.07
11 old	0.09	0.13	0.32	0.64	0.71	0.79	0.79	0.75	0.75
11 new	0.08	0.16	0.27	0.44	0.46	0.59	0.66	0.70	0.65
13 old	0.04	0.04	0.04	0.04	0.06	0.08	0.07	0.06	0.07
13 new	0.05	0.05	0.05	0.08	0.09	0.10	0.08	0.06	0.06
14 old	0.10	0.15	0.32	0.63	0.71	0.79	0.78	0.74	0.74
14 new	0.11	0.22	0.32	0.47	0.49	0.59	0.65	0.68	0.62

A typically cluttered room has a higher absorption value than an uncluttered room. Taking into consideration the improved absorption from typical clutter in three test cells, the indirect noise can be further reduced. In table 4 the increase in average Eyring absorption in three test facilities with a shell area of 284.4 m^2 as a result of typical clutter is shown.

Table 4 – Eyring average shell Absorption values empty and cluttered measured α[-]

See Table 2	31 Hz	63 Hz	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz	8 kHz
2 empty	0.03	0.05	0.08	0.18	0.25	0.35	0.40	0.42	0.38
2 cluttered	0.07	0.11	0.15	0.23	0.30	0.34	0.32	0.32	0.37

The shell acoustic model representation is chosen, because adding absorption has its limitations. In a room it is not impossible to increase the product of the absorption times area in some octave bands to a value more than three times the initial shell area, using the ceiling only, see figure 4. However this will change the direction of the sound field and the average absorption will bot rise far above 0.2 when the ceiling area is 0.2 times the total shell area and the rest of the shell is still fully reflective.

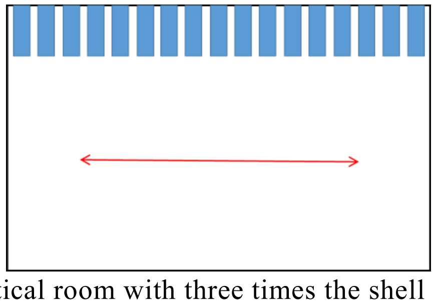

Figure 4 side view of a hypothetical room with three times the shell area as absorption on the ceiling

3.3 Source data

Based on the room acoustics model and the validation measurements on panels as used in these measured situations the sound power level that was provided by the customer or machine manufacturer was compared with the measurements taken around the engine or sound source. Before such a comparison can be made the consistency of 16 equal engine rooms with the same measurement device and procedure was done The standard deviation of this exercise as shown in table 5 gives an upper limit to the inaccuracy of the measurements of the calibrated device.

Table 5 – Standard deviation of average noise levels in equal engine rooms under equal load in dB(-)

16 Hz	31 Hz	63 Hz	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz	8 kHz	dB(A)
0.9	1.2	1.8	0.9	0.8	0.7	0.9	0.8	1.2	0.8	1.0

Initially the same 8 projects as in table 3 were compared. There is a striking inaccuracy found in a single manufacturer, another engine was not operating on the designed operating pressure. After these projects were removed 3 projects and 5 engine rooms remained representative. This improved the accuracy of the 125 Hz values, but had limited effect on the 63 Hz values. Furthermore no dB(A) value in any machinery room was at least 6 dB(-) less than predicted, see figure 5.

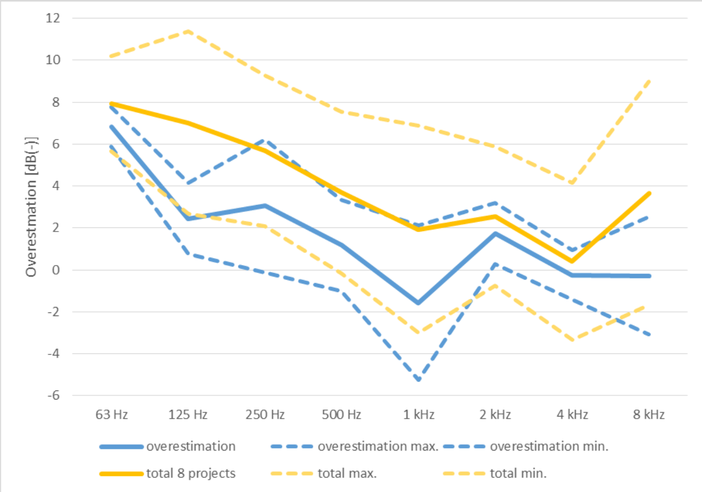

Figure 5 overestimation of sound pressure in a machinery room

When the additional absorption from clutter as shown in table 4 is assumed to be half related to the floor and half to the walls a hypothetical change in absorption as a result of clutter can be derived as shown in table 6. The room acoustics can be re-evaluated as shown in figure 6. The dotted red lines that are not mentioned in the legend show the standard deviation for identical systems from table 5. When poor predictions are left out and the average clutter is included the calculated and measured sound pressure values differ roughly as much as the standard deviation shown in table 5. Around 2 dB more sound power in the 63 Hz octave band is needed in order to create a perfect fit.

Table 6 Influence of clutter on average shell absorption values

See table 2	31 Hz	63 Hz	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 Hz	8 kHz
1 manned	0.09	0.16	0.26	0.42	0.45	0.56	0.62	0.65	0.60
1 cluttered	0.10	0.32	0.41	0.53	0.57	0.56	0.62	0.65	0.60
3 manned	0.03	0.04	0.11	0.13	0.14	0.18	0.18	0.18	0.18
3 cluttered	0.05	0.07	0.14	0.15	0.17	0.18	0.18	0.18	0.18
4 manned	0.12	0.22	0.32	0.47	0.48	0.58	0.64	0.68	0.62
4 cluttered	0.12	0.32	0.41	0.53	0.56	0.58	0.64	0.68	0.62
8 manned	0.12	0.22	0.32	0.47	0.48	0.58	0.64	0.68	0.62
8 cluttered	0.12	0.32	0.41	0.53	0.56	0.58	0.64	0.68	0.62
9 manned	0.06	0.05	0.06	0.08	0.09	0.11	0.09	0.08	0.08
9 cluttered	0.08	0.15	0.14	0.15	0.17	0.11	0.09	0.08	0.08

11 manned	0.08	0.16	0.27	0.45	0.47	0.60	0.68	0.72	0.66
11 cluttered	0.09	0.37	0.47	0.59	0.64	0.60	0.68	0.72	0.66
13 manned	0.05	0.05	0.05	0.08	0.09	0.10	0.08	0.06	0.06
13 cluttered	0.07	0.07	0.07	0.09	0.10	0.10	0.08	0.06	0.06
14 manned	0.11	0.22	0.32	0.48	0.50	0.60	0.66	0.69	0.63
14 cluttered	0.11	0.33	0.43	0.56	0.59	0.60	0.66	0.69	0.63

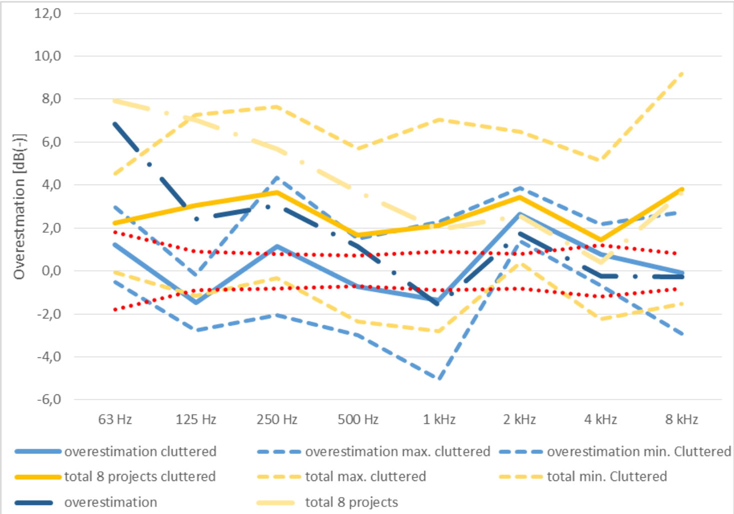

Figure 5 For clutter corrected predicted sound pressure levels in machinery rooms

3.4 Attenuator values in practice

In line with the last paragraph the attenuation was evaluated, see figure 6. Therefore the adjusted sound power spectrum was used based on a uncluttered shell in line with individual source measurements and the average shell absorption value mentioned as 'new' in table 3.

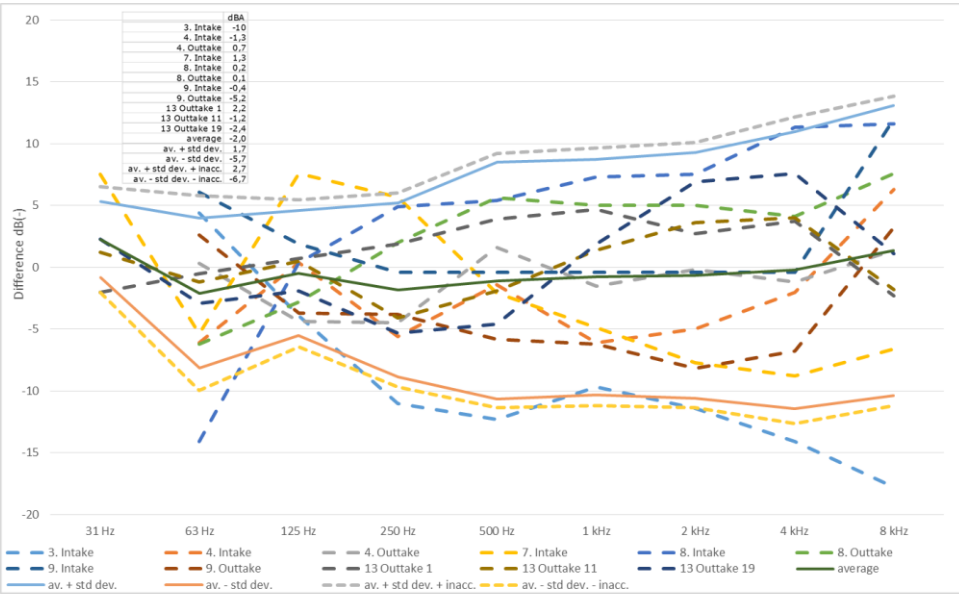

Figure 6 Overestimations on the transmission loss for splitter attenuators

Note that how poor the prediction of absorption values the sound pressure is fitted to the measurements, so the sound pressure in the room is always correct. Only the direct noise is potentially underestimated. The reverberant noise is much more than the direct noise contribution, so the potential underestimation of the noise acting upon the silencers is negligible.

Two silencers were excluded from figure 6. One of them had such high attenuation in the high frequencies that the noise didn't exceed the background noise, even not at 0.1 m. from the silencer. The other one was around 30 dB worse for frequencies of 1 kHz and above, but was not our own make and had unknown infill material. So the numbers cannot be properly compared. Because of the interesting situation a schematic of this underperforming silencer is given in figure 7. The noise in the pipe (on the right of figure 7) can be assumed omnidirectional, so the square diffusor, between the pipe bend and the attenuator directs the noise in a straight line through the splitters.

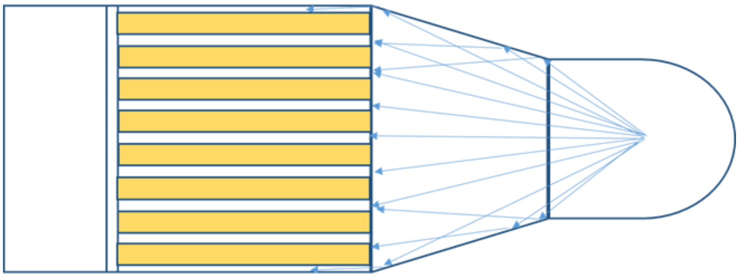

Figure 7. Silencer with serious high frequency shortcomings.

Directionality has a strong influence on the high frequency transmission loss, see figure 8.

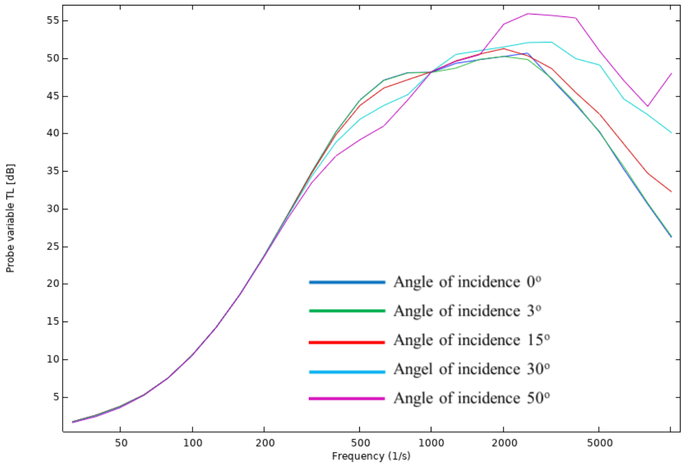

Figure 8 The effect of angle of incidence on high frequency attenuation (3, from table 2)

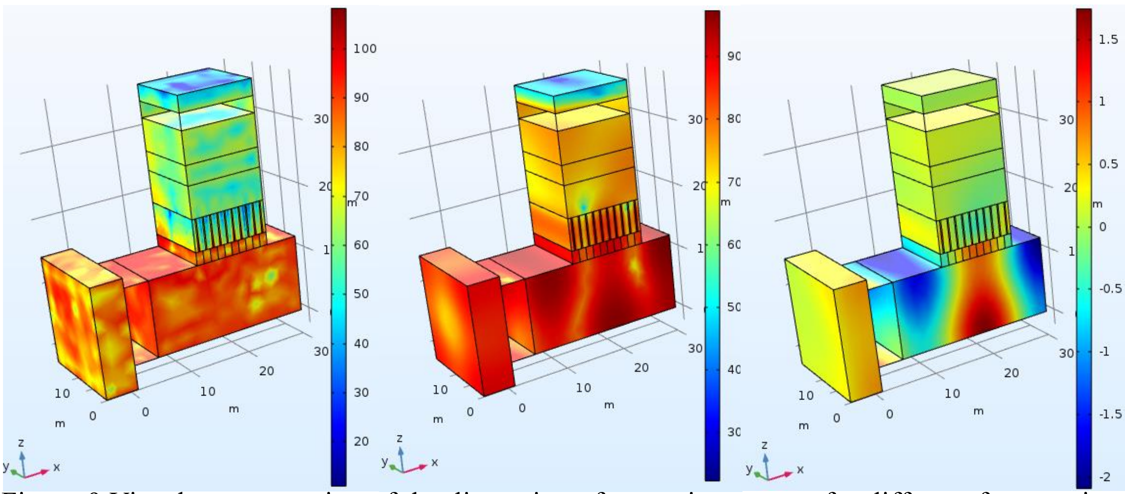

Figure 9 Visual representation of the dispersion of acoustic pressure for different frequencies

As can be seen in figure 8, the directivity of sound can have a strong effect in higher frequencies and has no effect on the lower frequency attenuation. The effect in high frequencies can easily exceed 10-20 dB above 2 kHz. In the low frequencies the position of the attenuator can have a strong effect in relation to standing waves. This effect only occurs in lower frequencies.

In figure 6 the spread in attenuation is high in the 63 Hz, than reduces and keeps increasing above 250 Hz. The average shortcoming in 63 Hz in figure 6 is again 2 dB, the same value as in figure 5. The measurements are done by a pressure measurement device only, the procedure (5) prescribes, measurements are not done closer than 0.5 m from a hard surface. Therefore measurements are potentially carried out in a standing wave were sound propagates as velocity rather that pressure. A 2 dB underestimation of the sound pressure is consistent with physics. Thus when deriving a sound power number in 63 Hz from measurements in an echoic room, a 2 dB safety margin should be applied.

The numbers are yet to be fit to a new theoretical model, but as a first impression from typical machinery rooms, the performance of the splitter attenuator can differ in accordance to figure 10. Differences work both ways, positive and negative. Which is a good motivator in order to deduct simplified algebraic equations that predict these design features.

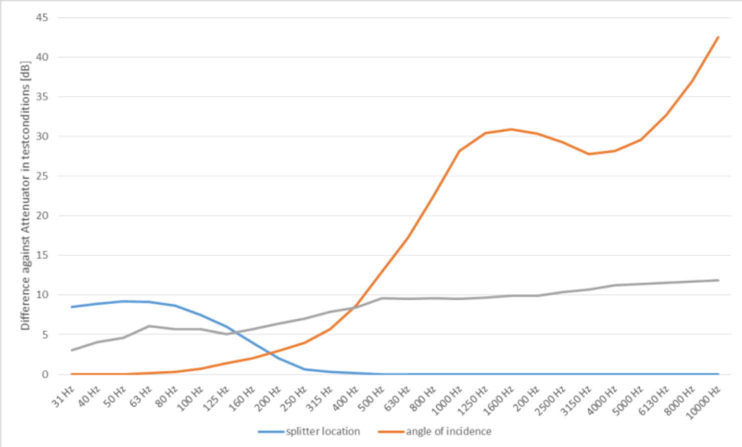

Figure 10. Typical design effects for Transmission loss values

3.5 Risk from vibrations

There is limited space I n this paper to go into detail. In practice a 1.7 dB margin for constructions with a good vibration insulation is sufficient. When vibration is an issue no single safety margin can realistically account for the potential defects.

4. Conclusions

When using only the shell and the average absorption value for room acoustics, it is important to use the Eyring absorption values and to account for the clutter. If this is not done the sound pressure level in a room as a result if the engine power is overestimated by respectively around 0.8 and 3.5 dB.

Using a sound pressure meter and a room acoustic model to evaluate the sound power emitted in a machinery room will likely result in a 2 dB underestimation of the sound power in the 63 Hz octave band.

Splitter attenuators are highly unpredictable but behave on average the same as under laboratory circumstances. Location affects the low frequencies and angle of incidence the high frequencies. It is worthwhile to deduct equations and differentiate direct and reverberant noise to predict these effects.

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REFERENCES

1. van Dijk, CC, De analyse van isolatie- en absorptiewaarden in absorberende ruimten, Blad geluid no. 4 demeber2015; Wolters Kluwer; p. 24-27.
2. VDI 2081 Noise generation and noise reduction in air-conditioning systems, Verein Deutscher Ingenieure, Düsseldorf 2001. ICS 17.140.20; 91.140.30; 2003-08.
3. ILHR13.01, Handleiding meten en rekenen industrielawaai, ir W.C. Reij; Interdepartementale commissie geluidhinder; 1981
4. ISO 354. Acoustics — Measurement of sound absorption in a reverberation room. INTERNATIONAL STANDARD; 2003(E).
5. ISO 11957, Acoustics - Determination of sound insulation performance of cabins - Laboratory and in situ measurements. INTERNATIONAL STANDARD; 2009(E).