Ductwork noise calculations: main outputs of AcouReVe project

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ABSTRACT

The AcouReVe Project aimed to improve the knowledge and the quality of acoustic calculation in ventilation ductworks. Such calculations are based on simplified models and the main issue is the input data. For each component of the ductwork, acoustic insertion loss and/or sound generation due to air velocity has to be known. Some components are well described by manufacturers, such as terminal devices, silencers, but others are not known. Sometimes, literature exists and can help to assess the input data, but the values may be out of date or no longer reflect current practices. This paper focuses on several component characterisations, such as attenuation in T or Y-shaped junctions, attenuation in straight ducts, attenuation and generation in bends and dampers and finally the acoustic behaviour of a manifold.

Noise remains a key issue for most building occupants, who wish to live in a relatively quiet indoor environment. Ventilation is one of the noise sources in buildings and efforts are made by most manufacturers to design silent solutions, both for components and ductwork. Acoustical consultants implement calculations to predict the sound levels in the rooms taking into account the ventilation system and ductwork but they often face to several issues:

- Lack of information about the acoustical characteristics of ductwork components, both for noise attenuation and regeneration
- Lack of confidence in the calculation process: it usually uses a simplified approach in which each ductwork component is considered independently from the others without interaction,
- Lack of confidence in the literature results: simplified tables or empirical relations are used but their field and limits of application are not well known.

The goal of the AcouReVe research project (2015-2018) was to make the ductwork noise calculations more reliable by providing answers to many of these issues.
Several ductwork geometries for bends and branches have been investigated to assess their acoustical characteristics (generation of noise and insertion loss), both numerically and experimentally. Losses in straight rigid circular ducts have been measured and analysed, as well as acoustical characteristics of components, such as for example acoustical behaviour of manifolds for balanced ventilation systems.
Laboratory tests allowed to check how the assumption of independent components used in the classical noise calculation methods leads to differences between calculation and measurements.

KEYWORDS

Noise, ventilation ductwork

1 INTRODUCTION

The AcouReVe project dealt with the ductwork noise calculation. Such calculation needs to split ductwork in several parts (elements) and associates to each of these elements the two following acoustic characteristics: sound generation by the air velocity and sound attenuation by the element. The calculation is rather simple to implement as it uses an energetic approach with uncoupled elements but the key factor is the acoustic input data for each element, which are often missing or uncertain (low reliability). The main target of the AcouReVe project was to get an overview of the ductwork noise calculation in order to highlight the main issues to be investigated, and improve the reliability of the results.

The work mainly focussed on ductwork components such as junctions (T-junctions, Y-junctions), straight ducts, bends, dampers and manifolds.

Other ductwork components such as air inlet or air outlet transfer devices are expected to be accurately characterised by manufacturers as they are located at the end of the ductwork, visible, and with specific design which can greatly influence the noise behaviour. For this reason, AcouReVe did not focus on them.

The acoustic characteristics of silencers are usually known since their role if to reduce the noise level (even if their noise generation is sometimes forgotten). They were not looked at in AcouReVe.

The project also investigated the way to implement such calculation in aerodynamics software, checking which data are necessary, and if their approach is compatible with that of acoustic calculation. In addition, a database would be implemented and available to users, giving a place to manufacturers to share these specific data with users or software developers.

This presentation focuses on the missing or uncertain data as follows:

- sound attenuation in junctions,
- sound attenuation in straight ducts,
- sound generation and attenuation in 90° bends,
- sound generation and attenuation in dampers,
- acoustic behaviour of rectangular manifolds.

2 SOUND ATTENUATION IN JUNCTIONS

The junctions are used to split the air flow from one duct to two (or more) other ducts, in order to distribute the air to several sub-branches or terminal devices. This simple geometrical configuration has been widely described in the literature (e.g. Ashrae), and is easy to implement in calculations. The calculation of the sound attenuation $\Delta L_{w\,B_i}$ uses two formulas, depending on cut-off frequency (i.e. the frequency below which only plane waves travel in the duct).

- For frequencies < cut-off frequency f_0

$$\Delta L_{w\,B_i} = 10log\left[1-\left(\frac{\frac{\Sigma S_{B_i}}{S_a}-1}{\frac{\Sigma S_{B_i}}{S_a}+1}\right)^2\right]+10log\left(\frac{S_{B_i}}{\Sigma S_{B_i}}\right) \tag{1}$$

- For frequencies > cut-off frequency f_0

$$\Delta L_{w\,B_i} = 10log\left(\frac{S_{B_i}}{\Sigma S_{B_i}}\right) \tag{2}$$

where S_a is the section of upstream branch and S_{Bi}, the ith section of downstream branch.

As equations (1) and (2) do not include geometrical parameters, they can be relevant for symmetrical Y-junctions or T-junctions. But for the T-junction with a main duct and a lateral branch, the sound could be distributed unsymmetrically, and this is not taken into account by equations (1) and (2). Several tests have been performed on 3 configurations, described in Figure 1.

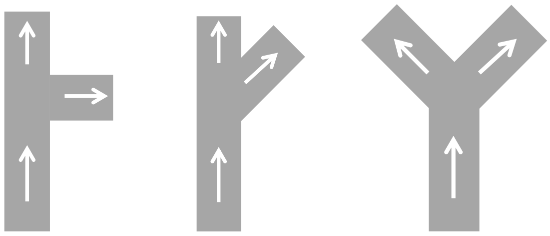


Figure 1: T90-junction, T45-junction and Y-junction

All ducts have a 250 mm diameter. They are of the round spiral steel type, as widely used in ventilation ductworks.
Sound power level is measured inside ducts (ISO 5136) at a distance of 1 m upstream of the junction and 2.5 m after the junction in each downstream branch. The sound level is generated by an upstream loudspeaker emitting a broadband noise, in axial or lateral position.
The measurements are supplemented by a noise calculation with a 2D approach.
The results are presented as the differences between the downstream and upstream sound levels, so that -3 dB means that the level in the considered downstream branch is 3 dB lower than in the incoming upstream branch.

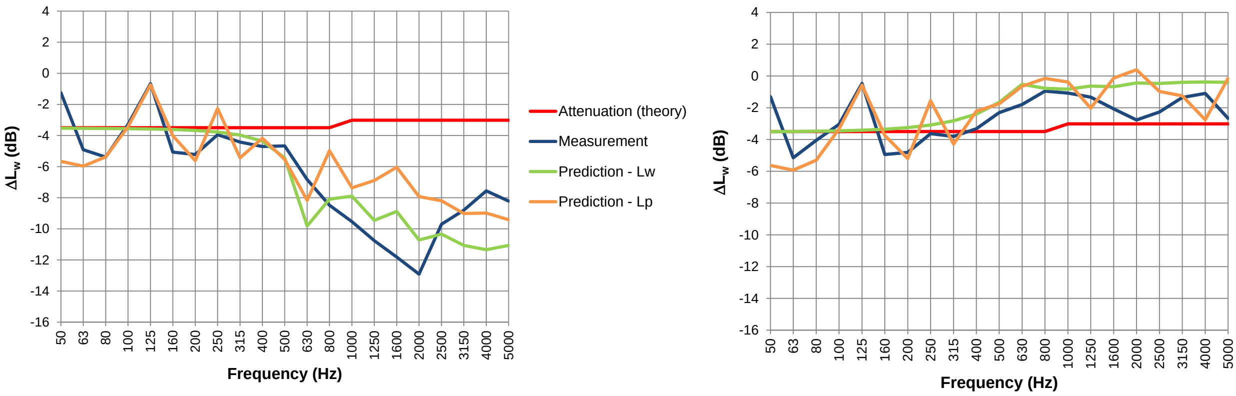


Figure 2: Attenuation for a junction T-90° (left: side branch, right: direct branch) – axial loudspeaker

Figure 2 shows the theoretical attenuation according to equations (1) or (2), the measured value of in-duct sound power level attenuation L_w, and the calculated value for in-duct sound level attenuation (considering sound pressure levels L_p and sound power levels L_w). Some standing waves remain at low frequency (despite the anechoic termination), leading to peaks, here at 125 Hz.
Anyway, the left figure shows that for frequencies below 500 Hz, the plane wave theory is quite well respected, with a -3.5 dB attenuation. Above this frequency (approx. two one-third octave band below the cut-off frequency (800 Hz), the theory is far away from the measurement and calculation results, which show a big difference with the upstream duct around 8-10 dB. By contrast, the right figure shows a smaller attenuation[1] in the direct upstream duct, around -1 dB, instead the -3 dB given by theory. It seems obvious that it is easier for noise to continue straight ahead in the duct than to take a way with right angle.
Same tests have been done on a T-45 junction, presented in Figure 3.

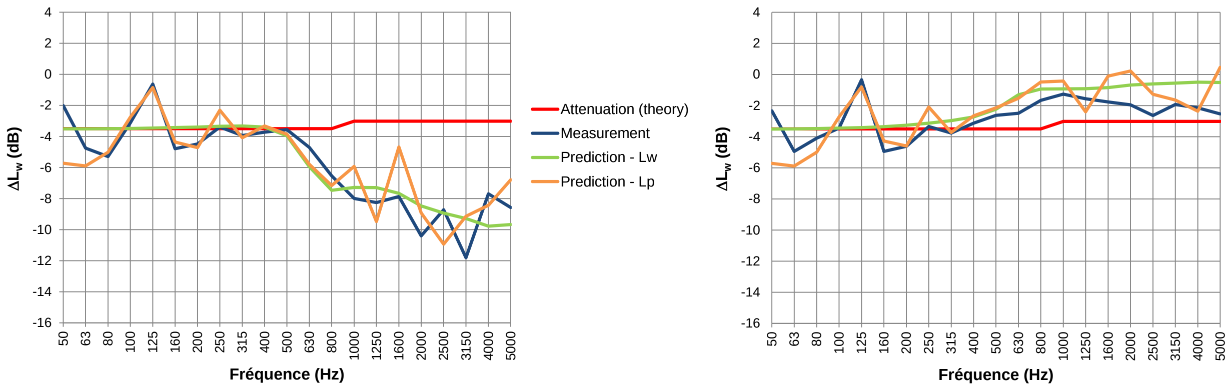


Figure 3: Attenuation for a T-45° junction ° (left: side branch, right: direct branch) – axial loudspeaker

[1] Throughout the document, negative values in the figures representing sound attenuation must be read as reducing the sound

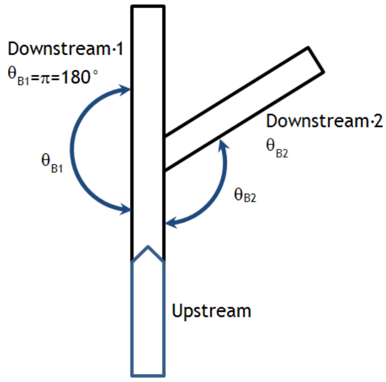
The same conclusions as for T-90° junction apply for low frequencies. For higher frequencies, the same dissymmetric results occur, with only a little lower amplitude with T-45° compared to T-90°.
It has to be noted that the dissymmetrical behaviour is reduced when the exciting loudspeaker is positioned on the lateral side of the upstream duct, instead of at the axial end. This position leads to excite resonance modes of a higher order.
The Y-junction shows that the symmetrical effect is achieved. The difference with the theory decreases especially around the cut-off frequency. On the higher range (1600-3150 Hz), higher attenuation is seen for experimental results, probably due to the losses in the ducts. The prediction obtained by the calculation fits with the literature theory.

The literature theory does not include a parameter to adjust the attenuation in the downstream branches according to the geometry of the junction.
A proposal is to add to equation (2) a new term taking into account the angle between the upstream and the downstream branches:

$$\Delta L_{B_i} = 10log\left(\frac{S_{B_i}}{\sum S_{B_i}}\right) + f(\theta_{Bi})$$

$$\text{with} f(\theta_{Bi}) \begin{cases} 1 & if\ branch\ 1 : \theta_{B1} = 180° \\ -2.5 \sin \theta_{B2} & for\ side\ branch\ 2 \end{cases} \quad (3)$$

$$\text{and } \frac{\pi}{2} \leq \theta_{Bi}$$



3 SOUND ATTENUATION IN STRAIGHT ROUND DUCTS

The total length of a ductwork can be more than several tens or hundred meters. The sound travels on long distances and can be damped by air relaxation effects and the acoustic losses on the duct walls (mainly noise transmitted to the duct and to its surrounding). This part only deals with circular ducts, which are known to produce few losses.
A typical data from literature for ducts around Ø 200 mm is a loss of 0.1 dB/m for the low frequencies and 0.3 dB/m for high frequencies. These values are small but can lead to significant attenuations for long ducts. Practice shows that design contractors often neglect this phenomenon in calculation, probably for conservative reasons.
Several tests were performed to assess these losses in the well-known galvanized steel spiral duct used in ventilation.
The test set-up consists in a loudspeaker, an upstream duct with in-duct sound power measurement (used as reference), the ducts under test (length 6 m, resulting from the assembly of 2 ducts of 3 m), and downstream an anechoic termination. A microphone is moved into the duct to measure the sound pressure levels every 20 cm, located at the ¼ of the diameter.
Tests were done on 4 diameters: Ø 80/100/125/160 mm. For the 160 mm diameter, one additional case consists of a 5 m length duct made of 5 sections of 1 m.

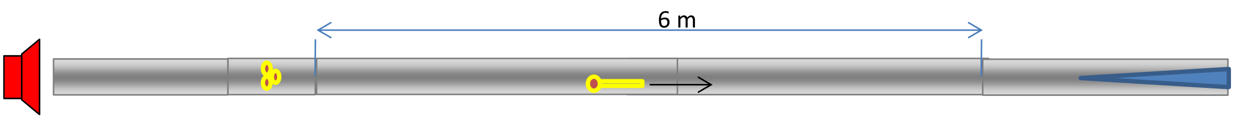


Figure 4: test set-up to characterize the attenuation in a straight duct

The Figure 5 shows the attenuations in dB/m for the 5 tested configurations. All of them show a low attenuation for frequencies below 1600 Hz. The value of literature, ~ - 0.1 dB/m, seems to be realistic. For frequencies higher than 1600 Hz, the 4 diameters with 2 x 3 m give comparable results, between - 0.5 and - 1.5 dB/m, that can be rounded to - 1 dB/m. The case of the Ø 160 mm with 5 x 1 m leads to a different result, with higher attenuation, locally around -3 dB/m at 2000 and 2500 Hz and between -1 and -2 dB/m above (the peak of +1.3 dB/m at 1600 Hz is unrealistic and unexplained). This difference is obtained although the numerous duct connections were tightened with care, using adhesive tape around the male connectors.

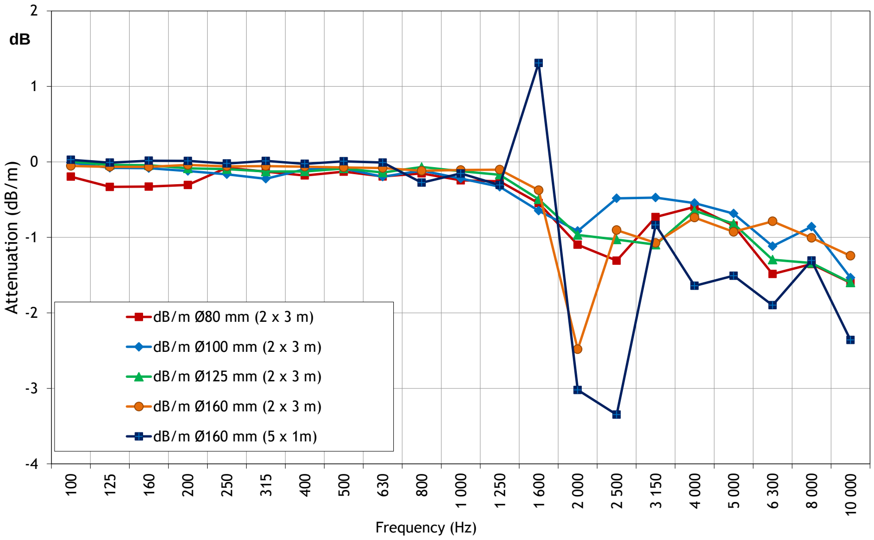


Figure 5: Attenuation in dB/m in a straight duct

The frequency from which the acoustic behaviour changes should be related to the cut-off frequency when the travelling waves are no longer plane waves (80 mm ⇨ 2530 Hz, 100 mm ⇨ 2030 Hz, 125 mm ⇨ 1624 Hz, 160 mm ⇨ 1270 Hz), but the Figure 5 does not show such differences, as the frequency of change appears to be always around 1600 Hz.
Note: the use of a loudspeaker in axial position does not favour the creation of non-plan modes

4 SOUND GENERATION AND ATTENUATION IN BENDS

4.1 Sound generation

Bends, as any obstacle in a duct with air velocity, can produce noise. The level of this noise greatly depends on the air velocity itself and as a first approximation, as the pressure loss coefficient (i.e. the shape of the obstacle). Most of bends used in ventilation have a soft radius of curvature, approx. around 1 diameter, even if they can be built differently, such as pressed steel (round bend) or assembly of several straight parts.

Several types of bends with a 250 mm diameter have been tested to determine their sound power level, using a reverberant chamber equipped with a silent wind tunnel. Upstream of the bend is an inlet horn, a convergent duct from Ø 630 mm to 250 mm and an upstream straight duct. Downstream of the bend is a straight duct and a horn that reduces the standing waves, which ends on the downstream reverberant room (ISO 5135).

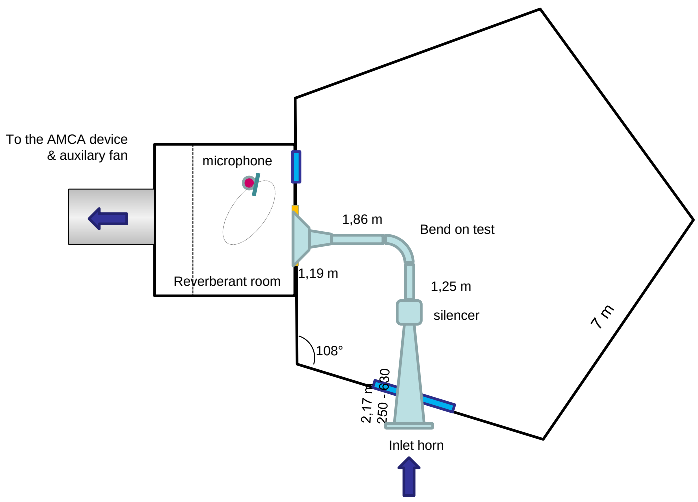


Figure 6: experimental set-up for noise generated by bend / junction

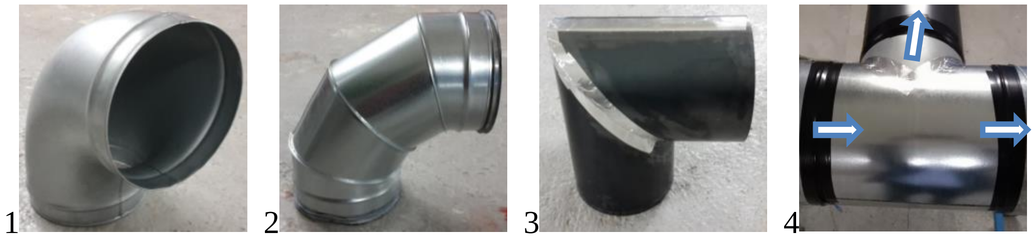


Figure 7: tested elements (l. to r.): 1. round bend, 2. bend by straight parts, 3. sharp bend and 4. T-90 junction

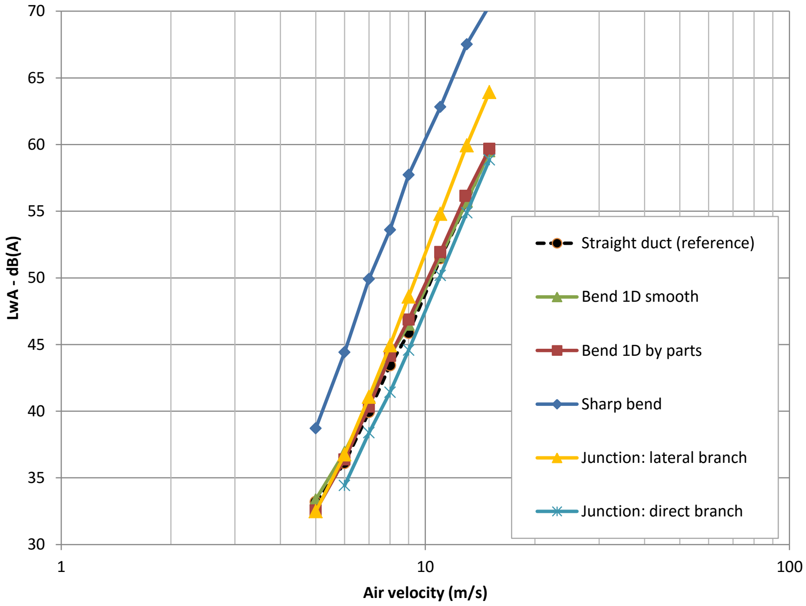


Figure 8: tests results for bends and junction. Overall sound power level vs air velocity

The Figure 8 shows the overall dB(A) sound power level for all configurations. Curves show identical results except for sharp bend and lateral branch of the junction. The noise level of the other configurations is the same as the straight duct reference configuration. This means that the 1D curvature radius bends do not generate noise in ventilation ductworks.
The sharp bend is unrealistic, tested to give a "worst case" result, around + 10 dB(A). The lateral branch of the junction has a radius curvature much smaller than 1D. It does not generate additional noise for low air speeds (< 8 m/s) and around + 4 dB(A) at 15 m/s; this latter velocity is not encountered in ductwork designed according to good practice rules.

4.2 Sound attenuation

A bend represents an impedance change in the duct for the traveling waves. This generates sound reflexions to upstream which reduces the sound energy transmitted to downstream, resulting in some attenuation. Tests were performed for 2 bends of Ø 160 mm, round bend and sharp bend (without airflow). Test set-up consists from upstream to downstream in a loudspeaker (axial or lateral with reference microphone), a straight duct, the tested bend, a straight duct with a section to measure the average sound pressure level and an anechoic termination. In the same way as before, the results are the difference of downstream sound level with the straight (without bend) configuration.
The Figure 9 shows the attenuation for the 2 bends, compared to literature results. Round bends have a lower pressure loss; their noise attenuation is small, equal to zero at low frequencies,

and around 2 dB at higher frequencies. On the contrary, the sharp bend presents a greater pressure loss, thus a higher attenuation, very close to the values given by literature.

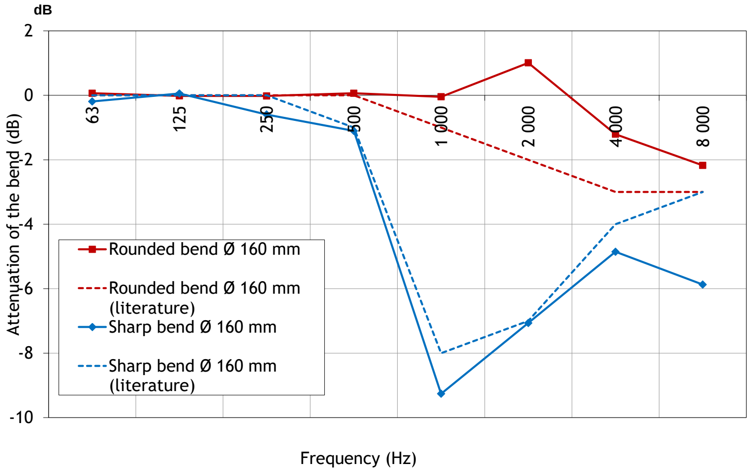


Figure 9: Attenuation of sharp and round bends (measurement and literature)

Note: when the loudspeaker is located on a lateral position of the upstream duct, then generating more non-axial waves, the attenuation is a little bit lower for the sharp bend.

5 SOUND GENERATION AND ATTENUATION IN DAMPERS

5.1 Sound generation

Dampers are used to adjust air flow rates in the various branches of the ductwork. Dampers create pressure loss, which can create noise. Five types of dampers of Ø 160 mm have been investigated, shown in Figure 10: iris, blade (full or perforated), blast gate, flexible opening foam (made with agglomerated PU foam) with removable pieces to adapt pressure drop.

Figure 10: tested dampers (l. to r.): iris damper, damper blade (full or perforated), blast gate, flexible opening foam

The dampers are installed in a reverberant room equipped with a silent wind tunnel, an airflow meter and pressure loss measurement, associated with sound power level measurement (ISO 5135) in a second room. Tests are performed for global air velocity around 10 m/s (the range depends on the pressure drop of the damper and its set-up).

The literature (VDI 2081) gives an assessment of the overall sound power level dB(A) with a coefficient $K_v = 51$ for logarithm of air velocity and another $K_x = 17$ for pressure loss coefficient.

$$L_{wA} = K_v * \log v + K_x * 17 \log \xi + 10 \log D + 10 \quad (4)$$

where v: air velocity, ξ: pressure loss coefficient, D: diameter.

For the blade damper and the foam damper, the direct application of equation (4) with $K_v = 51$ and $K_x = 17$ (default parameters of VDI 2081) leads to be best results compared to

experimental sound power levels, with respectively -3 dB error in average for the full blade and around 0 dB for the foam damper.
For the other types of dampers, the K_x parameter can be adjusted to give a smaller error between calculation and measurement. For iris damper and blast gate, the best K_x parameter is 13 instead of 17, leading to -2 dB error in average. For perforated blade damper, the best K_x is 8, with 0 dB error in average.
This means that the VDI 2081 equation is a correct basis to estimate the overall value of noise generated by dampers. Although a relationship exists between pressure loss coefficient and sound power level, the geometry of the blade has also a strong influence on sound generation and the K_x factor value can be optimized for each kind of damper.
This calculation only gives an estimate of the overall sound power level, without providing information about the spectrum shape.

5.2 Attenuation

As for bends, the damper is an obstacle in the duct that changes the impedance, creating reflexion and reducing transmission. For the downstream point of view, the damper creates a sound attenuation (insertion loss). The tests use an upstream loudspeaker to generate broadband noise. The attenuation is calculated by comparing sound pressure level in the reverberant room with and without the damper. Figure 11 compares 3 cases of iris, blade and foam dampers in a configuration close to a pressure loss coefficient $\xi = 10$. Metal dampers have no effect between 125 and 500 Hz, before to reach -2 dB (blade full) or – 4 (iris). The perforated blade has no effect, although it is in closed position (90°!). On the contrary, the foam damper, made of absorbing agglomerated PU foam, creates a true absorption for the high frequency range, up to -14 dB at 8000 Hz.

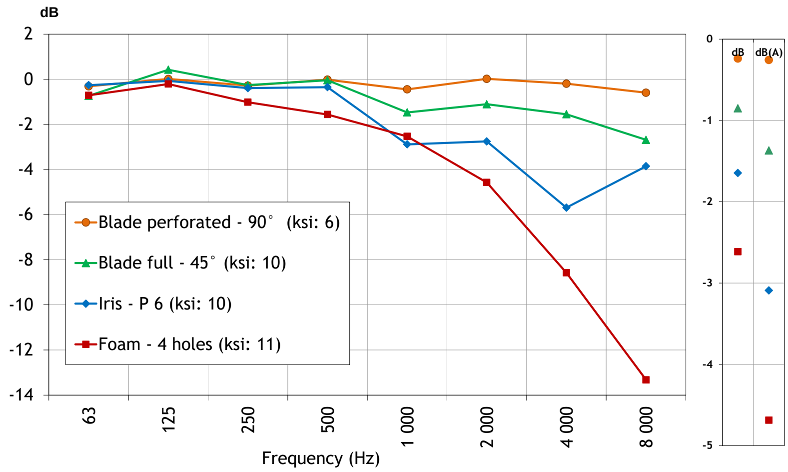


Figure 11: insertion loss of iris damper, damper blade full and perforated, flexible opening foam for $\xi \simeq 10$

6 ACOUSTIC BEHAVIOUR OF A MANIFOLD

Manifolds are used in ventilation to distribute the air coming from the ventilation unit to the downstream branches bringing air to different rooms. Their size and shape vary, from rectangular to circular, with spigots on lateral sides or on all sides as well. One could be tempted to consider them as a plenum, but their small volume makes this not suitable. One manifold has been tested and results compared with calculations. For practical reasons, the tested manifold is in wood with an inlet of Ø 250 mm and 6 outlets of Ø 160 mm. Internal size is L = 88 cm, w = 50 cm, d = 25 cm.

Figure 12: manifold in test

The experimental set-up consists in a lateral loudspeaker on the inlet side, with a measurement of sound power level at the inlet, and the measurement of sound power level at 4 of the 6 outlets. All branches with sound level measurement are equipped with an anechoic termination.
The test result is the attenuation of the sound power level observed in one branch compared to the incoming sound power level.

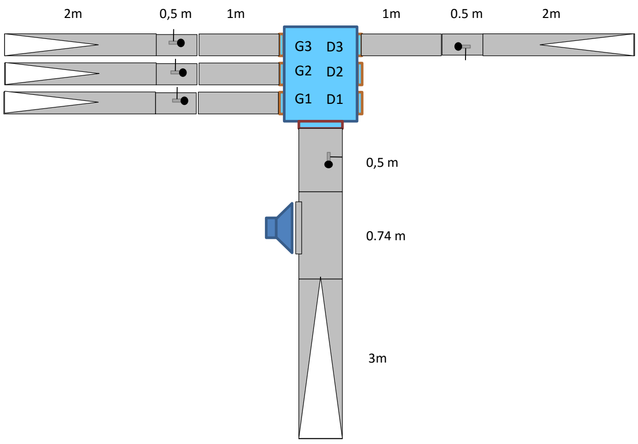


Figure 13: set-up for the acoustic characterisation of manifold

The numerical simulation is based on a 2D-approach. Many configurations have been investigated, for 1 to 4 (resp. 1 to 6) connected outlets in the experiments (resp. in the calculations).

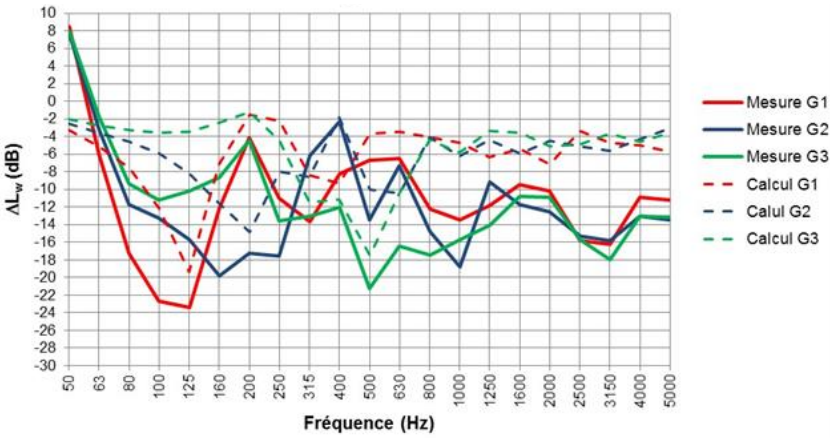


Figure 14: Attenuations obtained by measurement and calculation for only one connected outlet

The Figure 14 shows the experimental and calculated attenuations from inlet to outlet. Both approaches results are rather consistent, at least for the frequencies where peaks and holes appear. They are related to the modal situation of the volume and the branches. For high frequencies, calculation results are higher probably because no damping is taken into account. The general level of attenuation appears to be between -2 and -20 dB for the experiment, with high differences depending on frequency regions.
The first outlet (G1) exhibits a high attenuation at 125 Hz whereas the second outlet G2 has a maximum attenuation at 200 Hz.
Figure 15 illustrates the modal composition in the system at a frequency of 125 Hz. It shows that, whatever the outlet connected, a mode exists in the volume. For the first outlet (close to inlet), an anti-node leads to a high disruptive era, then an attenuation (see left figure). On the contrary, the 3rd outlet is on a pressure node, and the sound energy is easily transferred from inlet to outlet.
The analysis of all calculations (Figure 16) results for one outlet shows that all the possible combinations of all other outlets are less important than the position of the considered outlet. Improving the number of connected outlets reduces the amplitude, acting like a damping factor.

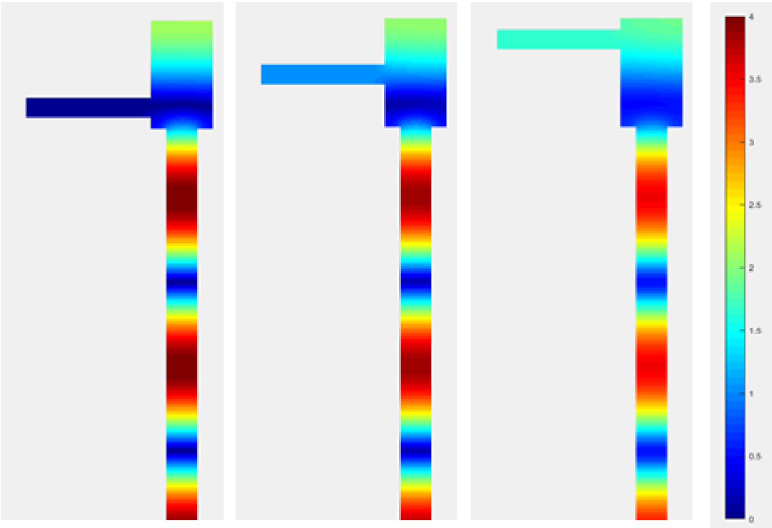


Figure 15: calculated modal composition in the system with one outlet for 125 Hz
(red: high acoustic pressure, blue: low acoustic pressure)

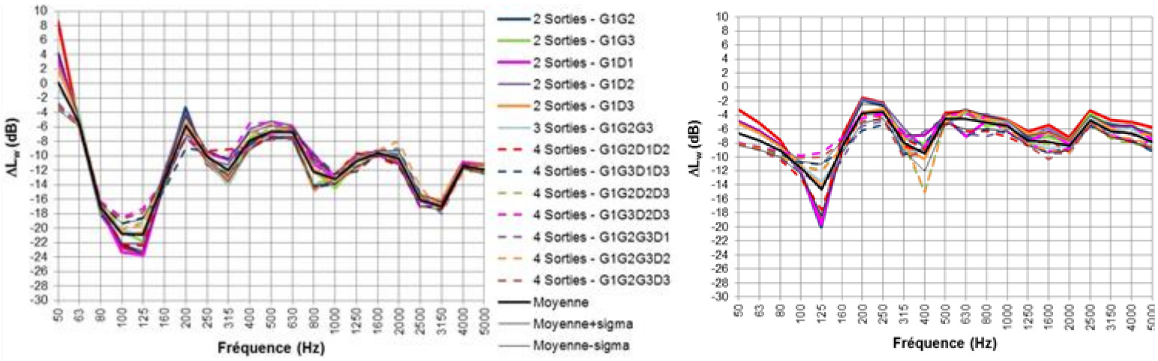


Figure 16: attenuation for G1 outlet (downstream – upstream), for 1 to 4 outlets connected
left measurement, right calculation

7 CONCLUSIONS

The acoustic calculation for ventilation ductworks is a challenge as it requires numerous input data, e.g. attenuation and sound generation for all the components of the ductwork. The method of calculation is based on a simplified energetic approach for which the crucial point is the quality of input data. Some are usually well known such as for terminal diffusers, but some others are rarely available, such as the noise generated and/or attenuated by junctions, straight ducts, bends, dampers and manifolds. This study has experimentally investigated several components. The main conclusions are:

- The literature about junctions is correct for symmetrical configurations. For T-90° or T-45° branches, the calculation requires a different treatment for each branch, the lateral one receiving less sound energy than the straight one.
- Bends provide a few sound attenuation, as given in the literature. If the curvature radius is at least 1D, no noise is generated compared to a similar straight duct.
- Dampers have to be considered as a bigger source of noise than of attenuation, which remains. Their overall sound power level can be estimated from the proposed VDI 2081 calculation, provided that the factor linked to the pressure loss coefficient is refined, according to the damper geometry. Tests showed that iris or perforated blade dampers are quieter.

- For low frequencies, losses in straights duct are consistent with literature and are very low, but the circular ventilation ducts provide higher losses than expected for high frequencies, with more than 1 dB/m for the considered galvanized spiral steel duct.
- Manifolds provide an attenuation whose frequencies are related to their geometry (modal behaviour). The number of connected outlets slightly influences the attenuation levels.

8 ACKNOWLEDGEMENTS

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