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# Comparison between prediction and measurement of sound attenuation associated to ventilation network elements

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## ABSTRACT

The AcouReVe Project (2015-2018) 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. The present work concentrates on sound attenuation across different ventilation network elements, such as bends, branches and manifold. A 2D prediction method was developed; it involves a multi-modal model for elements associated to guided waves propagation and a finite elements model for complex elements. Measurements have been performed at CETIAT laboratory on different elements; the experimental setup is described along with the different evaluated configurations. The comparison between the measured results, the predicted results as well as the results from simple standard formulas applicable for the considered elements is presented and discussed in detail.

Keywords: Ductwork, Noise, Ventilation

## 1. INTRODUCTION

Indoor noise remains a key issue for most building occupants. 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 rooms taking into account the ventilation system and ductwork but they often face 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 scope 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. 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 then rather simple to implement if 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).

Part of the work then focused on characterizing ductwork components such as junctions, straight ducts, bends, dampers and manifolds for balanced ventilation systems. Other ductwork components such as air inlet or air outlet transfer devices are expected to be accurately characterized by manufacturers as they are located at the end of the ductwork, visible, and with specific design which can greatly influence the noise behavior. For this reason, they were not targeted in the AcouReVe project. Acoustic characteristics of silencers are usually known since their role is to reduce the noise level (even if their noise generation is sometimes forgotten), so they were not considered in the project.

Laboratory tests allowed to check how the assumption of independent components used in the

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classical noise calculation methods leads to differences between calculation and measurements.

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, the implementation of a database was started and made available to users, giving a place to manufacturers to share these specific data with users or software developers.

This paper focuses on sound attenuation in junctions, along straight ducts, in 90° bends, in dampers and rectangular manifolds.

## 2. JUNCTIONS

The junctions are used to split the air flow from one duct to two (or more) other ducts, i.e., to distribute the air flow to several sub-branches or terminal devices. Several tests have been performed on 3 main configurations, described in Figure 1. The measurements are supplemented by a noise calculation with a 2D approach, as well as with the common simple expression prediction model.

The results are presented as in terms of sound level variation, i.e. the difference 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.

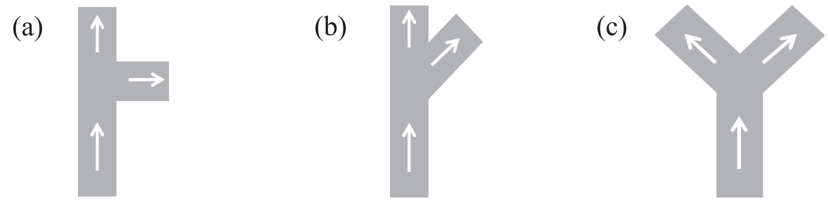



Figure 1 – Evaluated junctions; (a) T90-junction, (b) T45-junction and (c) Y-junction.

### 2.1 Simple prediction model

This simple geometrical configuration has been widely described in the literature (e.g. Ashrae (1)) and is easy to implement in calculations. The calculation of the sound variation $\Delta L_{W\,Bi}$ uses two formulas, depending on cut-off frequency $f_0$ (i.e. the frequency below which only plane waves travel in the duct) of the upstream branch

$$\text{For } f<f_0 \quad \Delta L_{W\,B_i} = 10log\left[1-\left(\frac{\frac{\sum S_{B_i}}{S_a}-1}{\frac{\sum S_{B_i}}{S_a}+1}\right)^2\right] + 10log\left(\frac{S_{B_i}}{\sum S_{B_i}}\right) \tag{1}$$

$$\text{For } f>f_0 \quad \Delta L_{W\,B_i} = 10log\left(\frac{S_{B_i}}{\sum S_{B_i}}\right) \tag{2}$$

where $S_a$ is the section of upstream branch and $S_{Bi}$, the i$^{th}$ section of downstream branch. The cut-off frequency $f_0$ for a circular cross-section duct is given by $f_0=(0.59\ c_0)/D$, with D the diameter in m and $c_0$ the sound speed in air expressed in m/s ; and for a rectangular section duct $f_0=(0.5\ c_0)/H$, with H the largest dimension in m.

For frequencies below the cut-off frequency, the first term of Equation (1) is associated to a reflection effect of the plane waves in the low frequency range. For frequencies above the cut-off frequency, Equation (2) only depends on the section of the considered downstream branch and the sections sum of all the downstream branches. It is obvious that since Equations (1) and (2) do not include geometrical parameters, they are most relevant for symmetrical Y-junctions or T-junctions. But for the T-junction with a main duct and a lateral branch, the sound transmission could be distributed unsymmetrically, and Equations (1) and (2) might not be appropriate.

### 2.2 2D prediction model

A 2D acoustic model was also developed. Three types of geometries were distinguished: straight duct elements, bends and elements with complex arbitrary geometry (such as junctions). A distinct modelling strategy was implemented for each of these three types of elements. For straight duct elements as well as bends, a multi-modal approach of guided waves propagation is implemented; for the complex element a finite elements model approach is preferred. The model is described in detail in (2) and is therefore not discussed further here.

Two types of attenuation are evaluated based on the developed model: either based on the acoustic power evaluated over a duct section (denoted Prediction -$L_w$), or based on pressure level obtained at 2 positions on the duct section at half the radius (denoted Prediction -$L_p$). The difference between these two evaluations is that for the method denoted Prediction -$L_p$ the pressure level $L_p$ in the upstream duct branch (sound emission) combines the incident and reflected waves due to the discontinuity inducing some interference effects; while for the method denoted Prediction -$L_w$ the power level $L_w$ in the upstream duct branch is obtained from the power associated to the incident waves only.

## 2.3 Results

All ducts are 250 mm in diameter, of the round spiral steel type, as widely used in ventilation ductworks. Sound power level is measured inside ducts (ISO 5136 (3)) 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 a upstream loudspeaker emitting a broadband noise, in an axial or lateral position. Anechoic terminations are placed at the end of each branch.

Figure 2 shows for the T90-jounction the theoretical sound level variation according to Equations (1-2), the measured sound power level variation $L_w$, and the 2D model predicted sound level variation (considering sound pressure levels $L_p$ and sound power levels $L_w$). The accident observed in the measurements and the prediction based on pressure level in the one-third octave band 125 Hz and 250 Hz is associated to the fact that the microphones in the upstream branch (incident sound) are located in a section corresponding to a sound pressure node (interference between the incident wave and reflected wave due to the junction discontinuity). Figure 2(a) shows that for frequencies below 500 Hz, the plane wave theory (see Equation (1)) is quite well respected, with a 3.5 dB attenuation. Above this frequency (approx. two one-third octave bands below the cut-off frequency (800 Hz), the theory is far away from the measurement and 2D model prediction results, showing a large attenuation around 8-10 dB. By contrast, Figure 2(b) shows a smaller attenuation, 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.

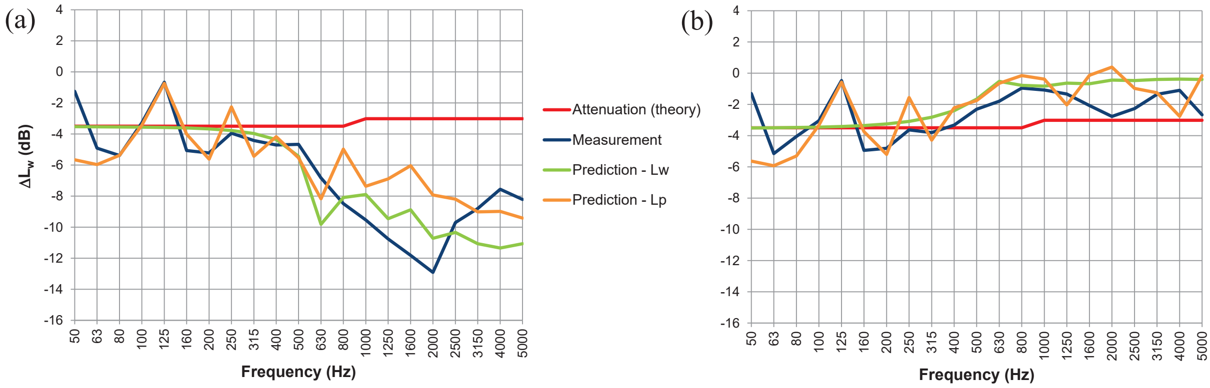



Figure 2 – Sound level variation for T90-junction; (a) side branch and (b) direct branch – axial loudspeaker.

Similar results have been obtained for a T45-junction, as presented in Figure 3. The same conclusions as for T90-junction apply for low frequencies. For higher frequencies, the same dissymmetric results occur, with only a little lower amplitude with the T45 junction compared to T90 junction. It has to be noted that the dissymmetrical behavior is reduced when the exciting loudspeaker is positioned on the lateral side of the upstream duct branch, instead of at the axial end. This position leads to higher order modes excitation and may be more representative of sound fields in real ductworks.

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 2D model 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. The following terms are proposed for a junction with a downstream branch (branch 1) in line with upstream branch and a second branch (branch 2) forming an angle $\theta_{B2}$ with the upstream branch ($\pi/2 \leq \theta_{B2}$)

$$\Delta L_{w\,B_i} = 10log\left(\frac{S_{B_i}}{\Sigma S_{B_i}}\right) + f(\theta_{Bi}) \text{ with } f(\theta_{Bi}) \begin{cases} 1 & for\ straight\ branch\ 1 \\ -2.5\ \sin\theta_{B2} & for\ \ side\ branch\ 2 \end{cases} \tag{3}$$

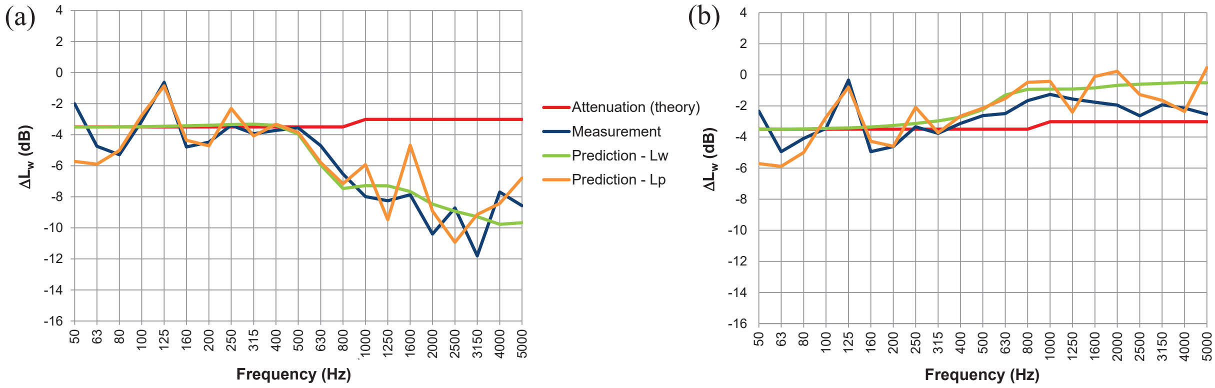



Figure 3 – Sound level variation for T45 junction; (a) side branch and (b) direct branch – axial loudspeaker.

## 3. STRAIGHT CIRCULAR DUCT

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 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 carried out on 4 different diameters: Ø 80/100/125/160 mm. For the 160 mm diameter, one additional case consisted in a 5 m length duct made of 5 sections of 1 m.

Figure 4 shows the attenuations in dB/m for the 5 tested configurations. All of them present a low attenuation for frequencies below 1600 Hz. The common literature value of ~ 0.1 dB/m seems to be realistic. For frequencies higher than 1600 Hz, the 4 diameters ducts made of 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 duct made of 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 regeneration of 1.3 dB/m at 1600 Hz is rather unrealistic and unexplained. This difference is obtained although the numerous duct connections were tightened with care, using adhesive tape around the male connectors.

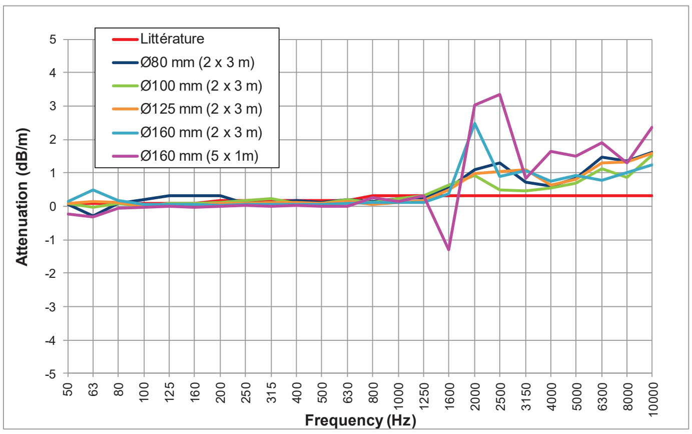



Figure 4 – Measured attenuation in dB/m in a straight duct.

The frequency from which the acoustic behavior changes should be related to the cut-off frequency when the travelling waves are no longer plane waves (for Ø80 mm at 2530 Hz, Ø100 mm at 2030 Hz, Ø125 mm at 1624 Hz and Ø160 mm at 1270 Hz), but Figure 5 does not show such differences, as the frequency with an accident appears to be always around 1600 Hz. This remains unexplained; the use of a loudspeaker in axial position does not favor the generation of higher order mode in the duct.

## 4. BENDS

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 on the pressure loss coefficient (i.e. the shape of the obstacle). Most of bends used in ventilation have a soft radius of curvature, approximately 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 (see Figure 5) have been tested to determine their sound power level and sound attenuation.

The generated noise level for such rounded bends was found very similar to a straight duct reference configuration, meaning that 1D curvature radius bends can be considered not generating noise in ventilation ductworks. This is not the case for the sharp bend which is rather unrealistic and not used in practice, the generated noise was about 10 dB(A) higher than the standard rounded bend (see (4) for results).

(a) (b) (c)

Figure 5 – Tested bended elements: (a) rounded bend, (b) bend by straight parts, (c) sharp bend.

Since the bend represents an impedance change in the duct for the traveling waves, it generates sound reflections to upstream which reduces the sound energy transmitted to downstream, resulting in some attenuation. Measured insertion loss is presented 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. The insertion loss is then obtained as the difference of downstream sound level measured without bend (replaced by straight element) and then with the bend. Figure 6 shows the obtained insertion loss for the 2 bends, compared to results based on (5). Round bends have a lower pressure loss; their noise attenuation is low, equal to zero at low frequencies, and around 2 dB at higher frequencies. The measured attenuation is between the predicted results (close to zero) and the results based on (5). On the contrary, the sharp bend presents a greater pressure loss, thus a higher sound attenuation. Predicted results are in relative accordance to those based on (5), but the measured maximum attenuation appears at a different one-third octave band. Note that the bend by straight parts provides similar results to the rounded bend.

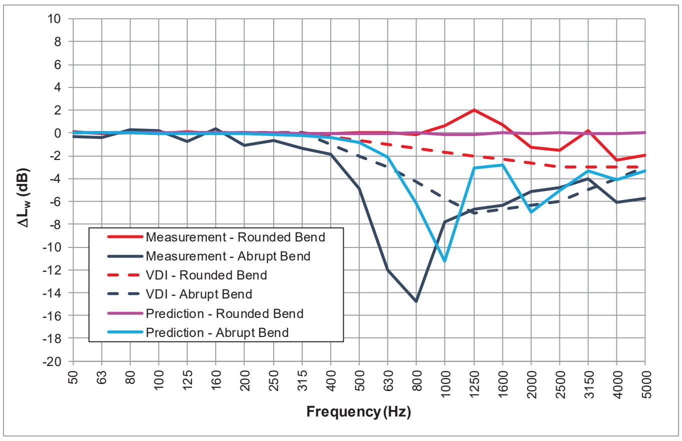



Figure 6 – Sound attenuation of sharp and rounded bends.

It must be noted that when the loudspeaker is located on a lateral position of the upstream duct, the attenuation is a little bit lower for the sharp bend.

## 5. DAMPERS

Dampers are used to adjust air flow rates in the various branches of a ductwork. Dampers create pressure loss, which in turns can create noise. Five types of dampers of Ø 160 mm have been investigated, as displayed in Figure 7: iris, blade (full or perforated), blast gate, flexible opening foam (made with agglomerated PU foam) with removable pieces to adapt pressure drop. Sound generation associated to these dampers was measured but is not presented in this paper. It was obtained that the overall sound power level dB(A) could be fairly well evaluated using expression given in VDI 2081 (5).

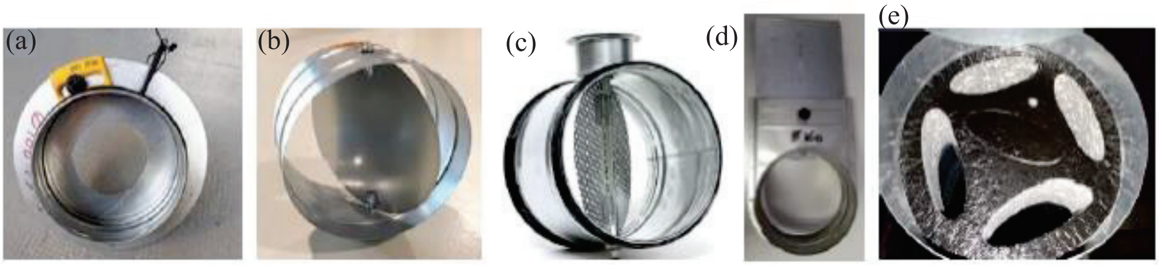



Figure 7 – Tested dampers: (a) iris damper, (b) damper with full blade or (c) perforated blade, (d) blast gate, (e) flexible opening foam.

As for bends, the damper is an obstacle in the duct that changes the impedance, creating reflection 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 levels in the reverberant room with and without the damper. Figure 8 compares several types of damper in a configuration close to a pressure loss coefficient $\xi = 10$. Metal dampers have no effect between one-third octave bands 125 and 630 Hz, before to reach 2 dB (full blade damper) or about 4 dB (iris damper). The perforated blade damper has no effect, although it is in closed position (90°!). On the contrary, the foam damper, made of absorbing agglomerated PU foam, is associated to sound attenuation, above 10 dB above 6.3 kHz.

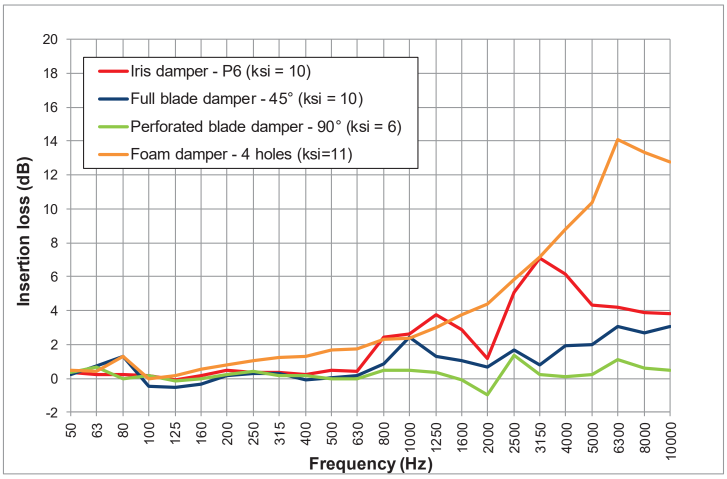



Figure 8 – Insertion loss of different dampers for $\xi \cong 10$.

## 6. ACOUSTIC BEHAVIOR OF MANIFOLD

Manifolds are used in ventilation systems to distribute the air coming from the ventilation unit to the downstream branches bringing air to the different rooms. Their size and shape vary, from rectangular to circular, with spigots on lateral sides or on all sides as well. As proposed in the Ashrae handbook (1), 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 (2D prediction model). For practical reasons, the tested manifold as seen in Figure 9 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.

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.

(a)

(b)

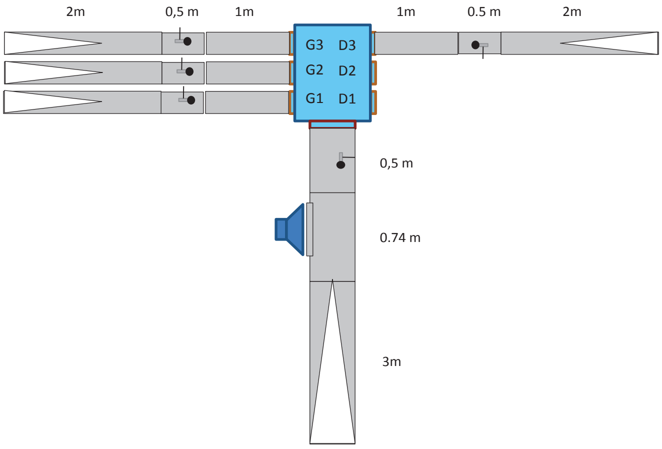


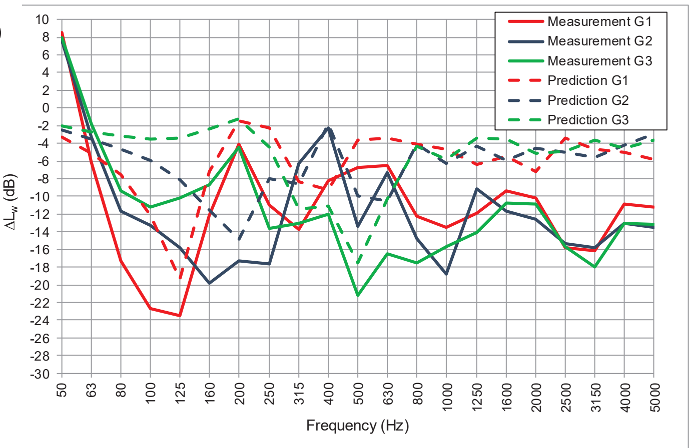

Figure 9 – (a) manifold used for tests; (b) set-up for the acoustic characterization of manifold.

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

Figure 10(a) shows the experimental and calculated transmission variations from inlet to outlet. The prediction and measurement results are rather consistent, at least for the frequencies where maxima and minima appear. These maxima and minima are related to the modal situation in the manifold volume and the branches position. For example, as shown in Figure 10(b), it is observed that at 125 Hz, the first outlet G1 (close to inlet) is positioned on a node of the manifold pressure field distribution leading to a low transmission. On the contrary, the 3rd outlet G3 is positioned on an anti-node of the manifold pressure field distribution, and the sound energy is then easily transferred from inlet to outlet. At high frequencies, prediction results are higher probably because no damping was considered. The general level of attenuation appears to be between 2 and 20 dB for the measurements, with high differences depending on frequency regions. The first outlet (G1) exhibits a high attenuation at 125 Hz whereas the second G2 has a maximum attenuation at 200 Hz.

The analysis of all predictions (Figure 16) results for one outlet shows that all the possible combinations of all other outlets is less important than the position of the considered outlet. Increasing the number of connected outlets reduces the amplitude, acting like a damping.

(a)




(b)

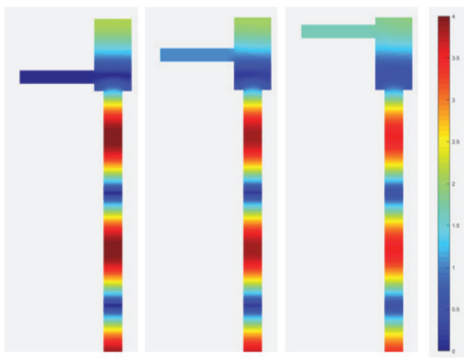


Figure 10 – (a) Comparison of measured and predicted sound level variation for only one connected outlet and (b) Pressure distribution in the system with one connected outlet at 125 Hz.

(a) (b)

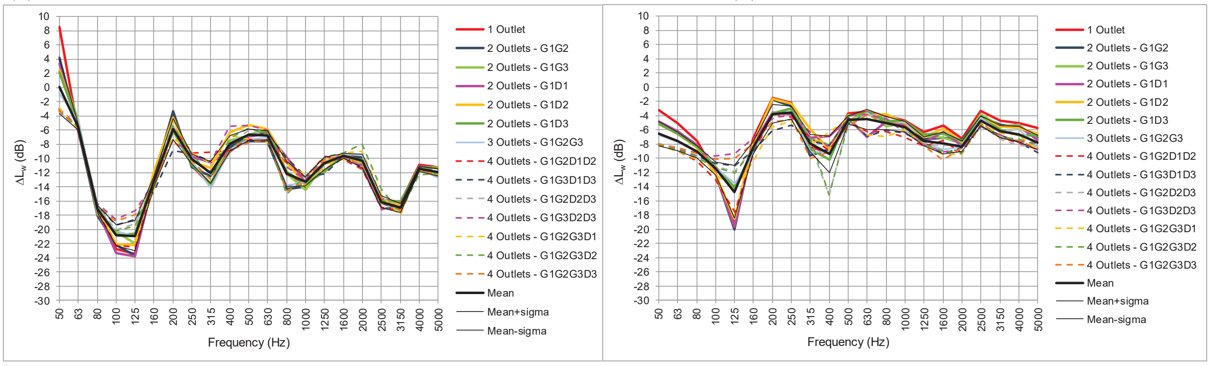


Figure 11 – Sound level variation for G1 outlet, for 1 to 4 outlets connected;
(a) Measurement and (b) Prediction.

## 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. The AcouReVe project has experimentally investigated several components and this paper has focused on sound attenuation associated to some of these elements. The main conclusions are as follow:

- The literature about junctions is correct for symmetrical configurations. For T90 or T45° junction branches, the calculation requires a different treatment for each branch, the lateral one receiving less sound energy than the straight one.
- 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 at high frequencies, with more than 1 dB/m for the considered galvanized spiral steel duct.
- Bends provide limited sound attenuation, as given in the literature. Furthermore, if the curvature radius is at least 1D, there is generally no noise generation compared to a similar straight duct.
- Dampers have to be considered as an important source of noise rather than an attenuating element. However, the tested foam damper was associated to a large insertion loss in the high frequency range.
- Manifolds provide an attenuation at frequencies that are related to their geometry (pressure field distribution in manifold). The number of connected outlets slightly influences the attenuation levels.

## ACKNOWLEDGEMENTS

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