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1 Introduction

1.1 Objectives

WP5 deals with the exploration of mitigation approaches to reduce the noise produced and propagated in the Environmental Control System (ECS). A strong constraint that is taken into account concerns the overall efficiency and reliability of the control. Passive strategies, which do not inject additional energy into the system, are adopted for the sake of safety and energy consumption. The mitigation operates on two fronts: flow and acoustic control, using turbulence grids to dampen out the inflow inhomogeneity and turbulence, and porous liners/micro-perforates for acoustic absorption. Both experimental and numerical work has been carried out in WP5, though not systematically for each Task. The original plan has been carried with only one small modification which is the omission of the study of so called laminar flow devices (Task 5.1). This type of device, which in principle is a duct section with tubes so narrow that the flow inside is laminar, can be used as a pressure regulating device that is silent as compared to a valve that produce turbulence. However, low frequency models for the sound transmission of such devices are already developed and has also recently independent of this project been extended to frequencies beyond the plane wave range [1]. It was therefore decided to skip this part of the work and to put more effort on developing the knowledge around the use of so called micro-perforated plates (MPP’s) for noise control in ducts. MPP’s was originally proposed by Professor Maa [2] in the 1970’s and is essentially a perforated plate, with holes that are so small so that the viscous losses dominate over the inertia of the air moving at each hole. The classical application for MPP’s have been in room acoustics as so called panel absorbers. More recently it has been suggested that MPP’s also have a large potential for other applications for instance involving noise in ducts [3]. This was the motivation behind including MPP’s in the study of new and innovative passive noise control measures within the Idealvent project. In particular for such applications it is important to find the effect on the MPP plate of flow both one and two-sided.

1.2 References


2 Innovative noise control in ducts - KTH and KUL

2.1 Introduction

Tonal noise emitted from duct and pipe systems has a significant impact on the sound-emission of, e.g. air-conditioners [1] or turbochargers of car engine [2]. Among many others, a particular silencer designed to filter out tonal compressor-noise utilizing an optimized wall impedance, the so called “Cremer-impedance” [3], shows encouraging results with high transmission loss for the optimized frequency even for very short filters [4]. A drawback of such filters is, however, the specific wall impedance which may be highly reflective for other modes than the one for which is optimized. This applies, especially for higher order acoustic modes if the filter was designed for the plane-wave mode. Systems of large dimensions as for instance air-conditioners can reach large Helmholtz-numbers within the frequency range of interest and, naturally, numerous modes then propagate at the same time. Strongly reflective surfaces for those modes, however, may in combination with flow introduce instabilities in the acoustic field, e.g., whistling [5]. As a solution of this problem, the existing Cremer-silencer proposed in [4] is extended by prefixing an additional “modal-filter” designed to deafen higher order acoustic modes. The so called “modal-filter” consists of micro-perforated plates that are perpendicular to the duct cross-section and placed in the pressure nodes of the acoustic modes. The idea was first proposed in Ref.[6]. As shown in the full report D5.1, such filters are invisible for the plane-wave mode and can, hence, be used in combination with Cremer-silencers without restrictions.

2.2 Compact Dissipative Silencers with a Cremer Wall-impedance

The principle of using an expansion chamber muffler with a micro-perforated sheet tube as a dissipative silencer was proposed by Allam and Åbom in [11], see Figure 1. The damping behaviour of such a filter is constituted by both, the impedance of the cavity and the impedance of the micro-perforated plate which covers the cavity. Kabral and Åbom used this idea and computed an optimum impedance to deafen tonal noise of compressors in turbochargers [4]. As proposed by Kabral and Åbom the optimum wall impedance for plane wave ((0,0) mode) damping is chosen as the Cremer impedance [3] adapted to circular ducts and flow as proposed by Tester [13]. The computed impedance could easily be realized by designing cavity structures fitting to an existing micro-perforated layer[2]. A problem in practical applications is, however, that the optimal impedance changes with frequency and modes. A silencer is therefore only maximal effective for a small frequency band and a certain acoustic mode.
2.3 Modal Filters

Åbom and Allam first mentioned the idea to use micro-perforated plates perpendicular to the cross-sections of a duct to filter single acoustic modes from the acoustic field [6]. If the plates are placed in velocity maxima (or pressure minima) of the mode shapes, their resistance can effectively damp the acoustic wave. Theoretically, for the (m,0)-mode, a number of \( m \) micro-perforated plates is needed for such a filter; however, one has to consider that combinations of the same (m,0)-mode (but spinning in contrary directions) equilibrate to take a propagation path of smallest resistance. This is half of an azimuthal wave-length phase shifted (to have velocity minima at the filter surface), which makes the filter ineffective. Such a phase shift can be avoided using more plates, preferably a number of \( 2m \) plates. Due to the symmetry in the azimuthal modes, a filter designed for a higher mode will always also effect all lower modes but not vice versa.

2.4 Computation of filter properties

In the report D5.1 the damping of the Cremer and the Modal filter are computed from their multi-port representation. Advanced post-processing techniques are applied to the extracted pressure fields to map the (analytical) mode shapes in positive and negative direction of propagation on the computed or measured pressure field. As a result, the reflection and transmission-coefficients of the modes through the filter can be extracted which is used to compute the damping of the filter. An advantage of this approach is, that three different kind of damping-properties are accessible. The transmission loss \( L_T \) is the ratio of incident acoustic energy on the inlet and the outgoing acoustic energy on the outlet of the filter. This value, however, can be inadequate for some cases, as strongly reflective (but not absorbing) surfaces give high transmission losses. Therefore, the reflection loss \( L_R \) may be a useful quantity which shows the ratio between incident and outgoing acoustic energy at the inlet of the filter. From both those quantities, a total (dissipative) filter loss \( L_{tot} \) can be computed. If the transmission from the inlet to the outlet and the reflection-coefficient at the inlet for the filter is determined for the (m,n)-mode in \( T_{mn} \) and \( R_{mn} \), respectively, the three losses can be computed in dB with the following equations.
\[ L_T = 10 \log_{10} \frac{1}{r^2_{mn}} \]  
\[ L_R = 10 \log_{10} \frac{1}{r^2_{mn}} \]  
\[ L_{tot} = 10 \log_{10} \frac{1}{r^2_{mn} + r^2_{mn}} \]

One may also be interested in the overall loss \( L \) in an acoustic field, consisting of a sum of propagating modes incident on the filter. Following [5], the power scattering matrix \( S \) can be utilized, which gives

\[ L = 10 \log_{10} \frac{a^c a}{a^c (S^c S)a} \]

where \( a \) is a vector of modal amplitudes defined so that their squares give the acoustic modal power and \((…)^c\) is the operator for a complex conjugated matrix.

### 2.5 Experimental characterization of modal filters

Within WP5.1, KUL has investigated experimentally the properties of a set of modal filters with one MPP panel mounted in the mid-plane over the full length of the filter (Figure 2) [15].

![Figure 2 – modal filters of length L=2D, 4D and 6D (left) and close-up of an Acustimet micro-perforated panel indicating the flow direction for the different panels (right).](image)

Three modal filter elements with length L=2D, 4D and 6D, where D=84 mm is the diameter of the duct, have been manufactured which are also used in WP5.2 and are discussed in section 6. The MPP panel is mounted in a slit in the wall and can be replaced to study the effect of the MPP impedance on the behavior of the filter. The MPP used in this study are Acustimet panels, consisting of slit-shaped perforations. Five different panels have been provided by SNT, with a thickness of approximately 1 – 1.5 mm and a porosity of approximately 0.7% (MPP A and B), 1.5% (MPP C), 2.5% (MPP D) and 3.5% (MPP E). Due to the manufacturing process, a significant variation of the properties over the panel may occur. After characterization of the panel impedance, only panels A, C and D were considered for detailed study because of the similarities between panels A and B, and D and E respectively. The orientation of the panels in the modal filters is chosen such that the slits are perpendicular to the flow direction for MPP A and C as shown in Figure 2. Due to the smaller size of the available panels, MPP D could only be mounted with the slit shaped perforations parallel to the flow.
2.5.1 Experimental characterization of modal filters in a quiescent medium

Figure 3 shows the effect of the MPP on the reflection coefficients for the (1,0) mode in a quiescent medium. Due to the acoustic damping of the panel, the resonance dominated behavior of the rigid filter is smoothed. However, the properties of the MPP itself have a negligible impact on the reflection coefficients and the reflection coefficients for all three MPP collapse for all filter lengths. The conversion coefficients are also identical to the reflection coefficients for the azimuthal modes, as is the case for the modal filter when the MPP is replaced by a rigid plate.

![Figure 3](image)

On the other hand, the transmission coefficients of a modal filter are strongly dependent on the properties of the perforated splitter plate. These coefficients describe how sound waves are propagating through the coupled channels of the modal filter, which is determined by the duct modes within the element. These modes strongly depend on the properties of the MPP. The effect of the MPP is therefore not limited to the introduction of damping in the system, but completely changes the transmission properties of the modal filter with respect to the rigid reference case, as shown in Figure 4.
Figure 4 – Transmission and conversion coefficients (T+) for the (1,0) mode for a modal filter with length L=2D (left) and L=4D (right) with a rigid splitter plate (gray dashed), MPP A (black), MPP C (red) and MPP D (blue) for a quiescent medium.

For the filter with length L=2D, the similarities between the different panels are most likely due to the end effects at the inlet and outlet of the filter, which can have a significant impact on the filter behavior for short filter lengths. For the longer filter (L=4D), the differences between the panels are more pronounced and the frequency dependency no longer follows the same trend for all panels. This illustrates that the properties of the modal filter not only depend on the type of panel and the associated duct modes, but are also strongly influenced by the length of the element.

2.5.2 Experimental characterization of modal filters in a moving medium

When a flow is present through the modal filter, the micro-perforated panel is subject to a two-sided grazing flow. It is well-known that the presence of a flow alters the acoustic impedance of a MPP. For micro-perforated panels with circular and slit-shaped perforations, the presence of a grazing flow on one side of the panel increases the resistance and slightly decreases the reactance [11]. Similar effects can be expected for two-sided flow over a MPP with slit-shaped perforations.

Figure 5 shows the influence of a grazing flow with a centerline Mach number of M = 0.1 on the transmission and conversion coefficients of the downstream propagating (1,0) mode for the different panels and filter lengths. For all filter lengths, the effect of the flow on the more porous panels C and D is clearly more pronounced than on panel A. Due to their larger perforations and more rough surface, these more important flow effects were expected based on earlier studies [16], [17] for perforated plates.
Figure 5 – Transmission and conversion coefficients (T+) of the scatter matrix of a modal filter with length L=2D (left), L=4D (center) and L=6D (right) and equipped with MPP A (black), MPP C (red) and MPP D (blue) in quiescent conditions (dashed line) and with an axial flow of M=0.1 (full line).

All trends can be explained by the expected increase of the MPP resistance due to the grazing flow. For all transmission coefficients, the flow results in a shift of the coefficients in the direction of higher panel resistance: away from the coefficients for MPP D in the direction of MPP A.

2.5.3 Acoustic power dissipation

Because of the multi-modal nature and the strong interaction between the azimuthal modes, it is not straightforward to draw conclusions on the attenuation based on the scatter matrix coefficients alone. Therefore, based on the measured multi-port characteristics, the power loss of the filters has been computed using equation (4). Since the plane mode propagates unattenuated through a modal filter, a single downstream propagating (1,0) mode is chosen as incident sound field. Figure 6 shows the power loss for this incident field in quiescent conditions.
The modal filters yield a broadband power loss between 2dB and 5dB, with a slight decay as a function of frequency. The higher values observed for frequencies below 2500 Hz are not physical and caused by the accumulation of errors and uncertainties originating from the singularity of the modal matrix at the cut-off frequency. Although the effect is small, the power loss consistently increases with the MPP resistance for all filter lengths.

Figure 7 shows the increase of the power loss due to the presence of a mean flow, with respect to quiescent conditions. For all filter lengths, MPP A only experiences a minor influence of the grazing flow. For the other panels, an additional 1dB of power loss can be obtained as a result of the flow through the modal filter. This gain can compensate for the inferior damping of these panels with respect to MPP A, observed in quiescent conditions. Whether or not this trend holds for higher flow velocities, is currently still under investigation.

2.6 Numerical Analysis and Results

2.6.1 Frequency-domain simulations
A Cremer – Modal – Filter (CM - Filter), i.e., a combination of a MPP modal filter & a Cremer silencer is investigated numerically by solving the Helmholtz equation for a circular duct for the modes of interest. This is a simplified method adapted from [12] for higher order modes but without flow. The modes are injected at a source cross-section and the field is captured at 100
positions before and after the filter. For the all the details please refer to the report D5.1 here only the main results will be given. A filter is designed that sufficiently (target 10 dB) damp the tonal noise at the blade passing frequency (BPF) of an axial compressor in a circular duct. The modal data of this compressor is adapted from ref. [1] and is presented in Figure 8.

Figure 8: Modal source power of an axial compressor, from [1]. Note the BPF at 2700 Hz.

For this compressor, the (0,0)-mode and the (0,2)-mode contribute most to the acoustic field, especially at the BPF at 2700 Hz and the CM-filter is designed to mainly damp both modes.

The filter design contains three steps; First, the Cremer-silencer is designed to have a maximum damping for the (0,0)-mode at the BPF. Second, the influence of the impedance on the modal filter is tested generally and also for plates which are available for possible applications. Third, different combinations of both filters are tested, namely both filters are series-connected and a CM-Filter with wall impedance and micro-perforated plates in the same section. As a geometrical requirement, the size of the final CM-filter is constraint to be less than 300 mm in axial direction with an inner diameter of 150 mm to fit the outlet of the compressor.

The Cremer-Impedance was implemented into the FE-model as a normal impedance boundary at the channel wall. Two different approaches were tested. First, a Cremer-impedance cladding the modal-filter from section 2.4, over the complete length of 2 duct diameters. Second, a series-connected combination of both filters with a length of 1 duct diameter each. For the plates, the real plate with most encouraging damping was used, i.e. $r=1.5$ and $f_0=1350$ Hz. The results are shown in Figure 9. The serial CM-filter had a higher total loss for the (0,0)-mode, which is reasonable due to a longer effective Cremer-impedance. The serial-connected CM-filter became more effective to the (1,0)-mode above 2700 Hz and for the (2,0)mode for all frequencies. Figure 10 shows the transmission loss for the same filters. The cladded filter shows much higher damping for all modes, which is due to the double length of the affective surface of the Cremer impedance and the micro-perforated plates.
2.6.2 Time-domain simulations
The modal filters, characterized experimentally in section 2.5, were also simulated by KUL using their in-house time-domain Runge-Kutta discontinuous Galerkin solver for the linearized Euler Equations [18], [19]. Within this framework, the micro-perforated panels are modelled using the time-domain transfer admittance formulation developed within WP 3.2. The approach
of WP 3.5 is applied to obtain the multi-port characteristics from the time-domain simulation results.

Figure 11 shows a comparison between the measured and simulated transmission coefficients for panels A and C and for different lengths of the modal filter. It can be observed that the correct trends are obtained for all different configurations. However, most transmission coefficients are overestimated due to the use of the LEE as governing equations. Because of the absence of the viscous term, only the effect of viscosity inside the MPP is implicitly taken into account by the transfer admittance while the viscous scrubbing losses over the surface of the MPP and against the duct walls are neglected.

Overall, considering the approximations made in the choice of governing equations and the rational impedance fit, a reasonable agreement is obtained. However, further research is needed to investigate the origin of these discrepancies.

2.7 Impedance eduction

2.7.1 Liner impedance eduction using a analytical multi-port model

Impedance eduction techniques are measurement techniques, where the liner impedance is obtained by comparison of a model to indirect, non-intrusive measurements. KUL developed such a technique based on two-port measurements and an analytical model [20]. Within WP 5.1, this existing impedance eduction technique has been extended to higher frequencies [21]. This is achieved by replacing the underlying two-port model by a multi-port model.
The setup consists of a rectangular duct where, within the test section, one wall is replaced by a locally-reacting liner. The rigid inlet and outlet sections are equipped with microphone and loudspeaker arrays. The first step of the impedance-eduction procedure consists of a multi-port characterization, according to the procedures described in detail in previous deliverables. For convenience, the transfer matrix formulation – relating the modal pressure and axial velocities – is used instead of the more common scatter matrix formulation.

In a second step, the measured scatter matrix is compared to an analytical model for a lined duct. This model considers the N least attenuated modes in the lined section, where N is the number of cut-on modes in the rigid inlet and outlet duct at that frequency. As shown in Figure 12, this matrix is complemented with a model for the hard wall-soft wall transition to account for the scattering occurring at the liner leading and trailing edge. Two different approaches for the modelling of the transition have been investigated: a fully populated transition matrix $T_{tr}$, assuming no a-priori knowledge on the transition, and a sparse formulation incorporating some physics of the scattering behavior.

![Figure 12 – Schematic overview of the impedance eduction approach](image)

After some algebra, the transition matrix coefficients can be eliminated from the mathematical formulation. Considering only the fundamental mode (0,0) and the first transversal mode (1,0), the impedance eduction algorithm comes down to solving a system of 9 equations in 9 unknowns: the axial and transversal wavenumbers in the upstream and the downstream direction for each mode and the liner impedance. When some physics are introduced in the transition model, the equations for the different modes are uncoupled and the procedure reduces to solving two systems of only 5 equations in 5 unknowns.

The procedure has first been validated on a numerical test case, where a known impedance is imposed in a FEM simulation (using LMS Virtual.Lab Rev. 13.2) and retrieved using the impedance eduction algorithm. The results of this validation showed excellent agreement between the imposed and the educed impedance values. Additionally, a first experimental application of the methodology has been carried out on a previously characterized SDOF liner [22]. Figure 13 shows the results of these measurements. The educed impedance, obtained with both formulations for the transition matrix, is in reasonable agreement with the reference data. Both the resistance and the reactance are slightly lower than the previous results. However, given the possibly different measurement conditions (temperature, humidity, …) and excitation levels, and the different acoustic incidence between the Kundt’s tube measurements and the impedance eduction measurements, the agreement is deemed satisfactory.
2.7.2 Impedance eduction for MPP under two-sided flow conditions

One objective of the project was to investigate the effect of one and two-sided flows on the impedance of a MPP. The case with a one sided flow has been studied earlier at KTH and has also been addressed again in this project by KUL, see report D5.1. But the case with flow on two sides which is of interest in, e.g., applications where the MPP is used as a modal filter or as guide vane, has not been studied before. In order to set up the measurements a special new test rig, see Figure 14, was constructed with the support of SNT and ASU labs at Ain Shams University, Cairo. The connection to Ain Shams is via the previous Marie-Curie project FLOWAIRS, where both KTH and Ain Shams were partners, resulting in a joint PhD student who did the main part of the work on this particular task.

Figure 14: The new test rig designed for measuring the effect of a two-sided flow on the impedance of MPP plates (from Ref. 14). The test plate is mounted between section 2 and 5. Incident plane waves can be generated at each of the 4 (1, 3, 4 & 6) connected pipes. The reflected and transmitted fields are decomposed using 3 x 4 microphones.

A full description of the procedure and some results from the first measurements have been presented as a paper at the recent AIAA/CEAS aeroacoustics conference [14]. The rig has recently been used to test a number of both standard MPP plates (with circular holes) and plates made by the Idealvent partner SNT. Based on these tests an effort will be made to come up with a modification of the formula for the one-sided case proposed e.g.[11].
2.8 Conclusions

A new filter design was presented to reduce tonal noise of an axial compressor. An optimal wall impedance (Cremer impedance) was used to reduce the (0,0)-mode significantly. Noise from azimuthal modes were reduced using a modal filter that consists of micro-perforated plates perpendicular to the cross sections. Both filters were investigated numerically using the finite element method. The results were encouraging. The combination (Figure 9) to a serial Cremer-Modal-Filter (1D long Cremer + 1D long Modal) was able to damp the (0,0), (1,0), and (2,0) by 15 dB, 14 dB, 9 dB compared to 25 dB, 14 dB, and 8 dB for the cladded Cremer-filter (2D Cremer) at the BPF (Figure 8). The transmission loss (Figure 10), however, was much higher for the cladded filter due to its longer affective surface.

A set of “modal filters” consisting of a micro-perforated panel in a duct have been characterized using the multi-port framework. This technique, accounting for all cut-on modes in the inlet and outlet duct, describes the properties of the element independent of the upstream and downstream impedance and therefore allows for a direct comparison between experimental and numerical results.

Modal filters with three different lengths and three different micro-perforated panels, varying the porosity and the orientation of the slit-shaped perforations, have been considered. These duct elements have been characterized experimentally in a quiescent medium and in the presence of a low Mach number flow. Comparison with the reference case containing a rigid splitter plate shows the importance of the filter length and allows evaluating the influence of a MPP on the transmission and reflection properties of the element. It was shown that the properties of the panel strongly influence the transmission and conversion coefficients of the element, while the reflection coefficients are nearly independent of the properties of the splitter plate. In terms of dissipated power, it was shown that a higher panel resistance yields a higher acoustic power loss. This effect is more pronounced for longer modal filters, where the end effects play a less significant role.

The influence of a flow through the filter element strongly depends on the properties of the MPP. More porous Acustimet panels, having larger perforations and a rougher surface, are significantly more affected by the presence of a grazing flow. Because of this effect, such panels can outperform panels with higher resistance, that yielded a higher power loss in quiescent conditions.

This experimental work has been complemented with numerical simulations using the KUL in-house Runge-Kutta discontinuous Galerkin discretization of the time-domain linearized Euler equations. The MPP is represented by a recursive time-domain transfer admittance formulation, using a rational approximation of the measured impedance. Only filters in a quiescent medium have been considered numerically at this stage due to the lack of reliable corrections for the MPP impedance under two-sided grazing flow. The simulations predict correctly the trend for all scatter matrix coefficients, but yield an offset in amplitude. This is most likely due to the
absence of viscous effects in the simulations. Further research will investigate these discrepancies in more detail and consider the presence of a mean flow.

Finally, the impedance eduction technique based on an analytical two-port model has been extended to higher frequencies using the multi-port framework. The analytical model accounts for a limited set of modes in the lined section and accounts for the hard wall – soft wall transition using an unknown transition matrix. The multi-modal nature allows for different assumptions in the model for the transition matrix, resulting in two slightly different mathematical formulations. Both techniques have been validated using a numerical model. A first experimental measurement on a reference liner sample has been carried out and revealed some convergence issues around the cut-off frequency. These are most likely related to inaccuracies in the multi-port characterization and are currently still under investigation. Overall, the measurements in a quiescent medium show good agreement with the reference data. In the presence of a grazing flow, the method captures the expected trends: an increase of the resistance and a slight decrease of the reactance.

2.9 References


[3]. Cremer L. Theory regarding the attenuation of sound transmitted by air in a rectangular duct with an absorbing wall, and the maximum attenuation constant produced during this process. Acoustica 3. 1953.


3 Acoustic simulations of perforate plates

3.1 Introduction

Within this task, SISW has performed some research on computational modeling aspects of perforate plates. The research work can be divided into two distinct contributions.

The first task consisted in extending the potential flow acoustics propagation solver to account for transfer impedance boundary conditions in the presence of grazing flow. This improvement allows an efficient modelling of the sound propagation in flow ducts with micro-perforated plates. This work is presented in section 3.2.

The second task involved the use of an in-house frequency domain high-order Linearized Navier Stokes Equations (LNSE) solver prototype to characterize the acoustic impedance of the slit resonator used in WP3.5. The results have been compared with the ones obtained in parallel by the KUL with their own time domain LNSE solver and reported in the deliverable D3.5 [1]. This comparison allows determining better the differences between the solvers and also the sources of discrepancies with respect to the experimental results. This work is presented in section 3.3.

3.2 Transfer admittance modeling

3.2.1 Formulation

The purpose of this task is to implement a transfer admittance model in the Linearized Potential Equation (LPE) code of Sysnoise [3]. It represents an extension of the transfer admittance functionality for cases with a non-zero mean flow. A transfer admittance relation describes the relationship between acoustic pressure and velocity of two surfaces. If surface S1 is linked to surface S2 through a transfer admittance relation, then the following equation is satisfied

\[
\begin{pmatrix}
  u_1 \\
  u_2 
\end{pmatrix} = \begin{pmatrix}
  \alpha_1 & \alpha_2 \\
  \alpha_4 & \alpha_5 
\end{pmatrix} \begin{pmatrix}
  p_1 \\
  p_2 
\end{pmatrix} + \begin{pmatrix}
  \alpha_3 \\
  \alpha_6
\end{pmatrix},
\]

where \( \alpha_i \) are the transfer admittance coefficients, \( u_1 \) and \( u_2 \) represent the local (structural) velocity of surfaces S1 and S2 respectively, and \( p_1 \) and \( p_2 \) represent the acoustic pressure on S1 and S2. The velocity can be related to the structural displacement \( \xi \) as \( u = i\omega\xi \). The acoustic pressure can be expressed in terms of acoustic velocity potential \( \phi \), which is the variable on which the LPE formulation is based, as

\[
p = -\rho_0 (i\omega\phi + \bar{u}_0 \cdot \nabla \phi),
\]

where \( \bar{u}_0 \) is the mean flow velocity vector. Therefore, the transfer admittance relation can be re-written as

\[
\begin{pmatrix}
  i\omega \xi_1 \\
  i\omega \xi_2 
\end{pmatrix} = \begin{pmatrix}
  \alpha_1 & \alpha_2 \\
  \alpha_4 & \alpha_5 
\end{pmatrix} \begin{pmatrix}
  i\omega \phi_1 + \bar{u}_0 \cdot \nabla \phi_1 \\
  i\omega \phi_2 + \bar{u}_0 \cdot \nabla \phi_2
\end{pmatrix} + \begin{pmatrix}
  \alpha_3 \\
  \alpha_6
\end{pmatrix}.
\]
In the presence of flow, Myers boundary condition can be used to prescribe an acoustic impedance on a surface. Myers boundary condition can be expressed as [4]

$$\nabla \phi \cdot \mathbf{n} = i\omega \xi + \mathbf{u}_0 \cdot \nabla \xi - \mathbf{n} \cdot (\mathbf{n} \cdot \nabla \mathbf{u}_0) \xi.$$

Applying this expression to the transfer admittance relation described above, we obtain:

$$\nabla \phi_1 \cdot \mathbf{n}_1 = [i\omega + \mathbf{u}_0 \cdot \nabla - \mathbf{n}_1 \cdot (\mathbf{n}_1 \cdot \nabla \mathbf{u}_0)] \left\{ \alpha_1 \phi_1 + \frac{\alpha_1}{i\omega} \nabla_0 \cdot \nabla \phi_1 + \alpha_2 \phi_2 + \frac{\alpha_2}{i\omega} \nabla_0 \cdot \nabla \phi_2 + \frac{\alpha_3}{i\omega} \right\}$$

$$\nabla \phi_2 \cdot \mathbf{n}_2 = [i\omega + \mathbf{u}_0 \cdot \nabla - \mathbf{n}_2 \cdot (\mathbf{n}_2 \cdot \nabla \mathbf{u}_0)] \left\{ \alpha_4 \phi_1 + \frac{\alpha_4}{i\omega} \nabla_0 \cdot \nabla \phi_1 + \alpha_5 \phi_2 + \frac{\alpha_5}{i\omega} \nabla_0 \cdot \nabla \phi_2 + \frac{\alpha_6}{i\omega} \right\}$$

These two conditions must be included in the variational formulation [3]. The resulting integrals to be added in the left-hand side of the integral formulation are:

$$\int_{S_1} \rho_0^2 \alpha_1 \left\{ i\omega \phi_1 + \phi_1 (\mathbf{u}_0 \cdot \nabla \phi_1) - \phi_1 (\mathbf{u}_0 \cdot \nabla \phi_1) - \frac{1}{i\omega} (\mathbf{u}_0 \cdot \nabla \phi_1) (\mathbf{u}_0 \cdot \nabla \phi_1) \right\} dS$$

$$+ \int_{S_1} \rho_0^2 \alpha_2 \left\{ i\omega \phi_2 + \phi_2 (\mathbf{u}_0 \cdot \nabla \phi_2) - \phi_2 (\mathbf{u}_0 \cdot \nabla \phi_2) - \frac{1}{i\omega} (\mathbf{u}_0 \cdot \nabla \phi_2) (\mathbf{u}_0 \cdot \nabla \phi_2) \right\} dS$$

$$+ \int_{S_2} \rho_0^2 \alpha_4 \left\{ i\omega \phi_2 + \phi_2 (\mathbf{u}_0 \cdot \nabla \phi_2) - \phi_2 (\mathbf{u}_0 \cdot \nabla \phi_2) - \frac{1}{i\omega} (\mathbf{u}_0 \cdot \nabla \phi_2) (\mathbf{u}_0 \cdot \nabla \phi_2) \right\} dS$$

As for the right-hand-side, the terms related to $\alpha_3$ and $\alpha_6$ should be added. However, in the case with flow, their gradients $\nabla \alpha_3$ and $\nabla \alpha_6$ need to be known to have the complete term. Since the user will not have easy access to this information, and since $\alpha_3$ and $\alpha_6$ are 0 in most applications, these terms will be excluded from the implementation of the transfer admittance with flow.

Note that the integrals related to $\alpha_2$ and $\alpha_4$ involve variables on both surfaces. If the two surfaces are not matching each other, a specific approach to handle incompatible meshes must be applied [5].

3.2.2 Validation

3.2.2.1 Muffler (Sullivan & Crocker 1978) with mean flow velocity $U=0$

In this first test case (Figure 15), the new implementation is validated in the absence of mean flow. The muffler investigated by Sullivan & Crocker has been used in the past to validate the transfer admittance implementation in the standard solver based on the Helmholtz equation. The transmission loss results given by this solver are compared to those obtained with the new implementation in the LPE formulation. The same boundary conditions are applied (acoustic velocity at the inlet, uniform impedance $Z=\rho_0 c_0$ at the outlet) for both computations.
Note that there is a small gap between the inner and outer cylindrical surfaces on which the transfer admittance is applied. This makes the areas of the surfaces different, and should be taken into account in the formulation. The solver incorporates a correction for this automatically, so the transfer admittance coefficients do not need to be corrected to take this into account. The plot Figure 16 shows results of transmission loss for both solvers. The agreement is perfect, since the LPE model should give the same results as the no-flow solver when the mean velocity is \( U_0 = 0 \).

Contiguous ducts with flow and wall impedance \( Z \), and no transfer between them \( (\alpha_1 = \alpha_5 = 1/Z, \alpha_2 = \alpha_4 = 0) \)

In order to validate the implementation of the terms related to the transfer admittance coefficients \( \alpha_1 \) and \( \alpha_5 \), a setup consisting of two contiguous square ducts is selected. There is no actual transfer of acoustic waves between them \( (\alpha_2 = \alpha_4 = 0) \); instead, the contiguous surfaces (Figure 17) have a certain wall impedance \( Z \). This case would not require a transfer admittance relation, since the ducts are in fact totally independent from each other, but it can be used to validate the terms related to \( \alpha_1 \) and \( \alpha_5 \), which should be equal to the wall admittance on the surfaces \( (\alpha_1 = \alpha_5 = 1/Z) \).
The duct sections have a side length $L$. The flow is uniform, with a Mach number $M=U_0/c_0=0.3$ on both ducts. As acoustic boundary conditions, an acoustic normal velocity is imposed in all inlets/outlets ($\pm u'$ on the inlet/outlet of one duct and $\pm 0.5u'$ on the inlet/outlet of the other). [Note that the use of a normal acoustic velocity for boundaries where there is normal mean flow is not supported, but it has been implemented for validation purposes]. The wall impedance at the contiguous walls of the ducts is $Z=(1.5+0.5i)\rho_0 c_0$.

Figure 18 shows pressure contour plots for several frequencies. The solutions of two different sets of computations are shown: on the left, no transfer admittance relation has been defined, only a wall impedance $Z$ on the contiguous walls; on the right, instead of a wall impedance, a transfer admittance relation with $\alpha_1=\alpha_5=1/Z$ and $\alpha_2=\alpha_4=0$ is applied on the contiguous surfaces. The agreement between them is perfect.

![Pressure (real part), frequency $f=0.2c_0/L$](image1.png)

![Pressure (real part), frequency $f=0.7c_0/L$](image2.png)

**Figure 18: Comparison between the impedance condition (left) and the transfer admittance condition (right) for the case of no transfer between the ducts ($\alpha_2=\alpha_4=0$) and conforming meshes**
In Figure 19, the results of additional computations are presented, where the meshes for each duct are chosen to be non-conformal. Once more, the agreement between the case with standard wall impedance and the case with transfer admittance is excellent.

Pressure (real part), frequency $f=0.2c_0/L$

Pressure (real part), frequency $f=0.7c_0/L$

![Figure 19: Comparison between the impedance condition (left) and the transfer admittance condition (right) for the case of no transfer between the ducts ($\alpha_2=\alpha_4=0$) and non-conforming meshes](image)

### 3.2.2.2.3 Contiguous ducts with flow and transfer admittance: $\alpha_1=2/Z$, $\alpha_2=-2/Z$, $\alpha_4=1/Z$, $\alpha_5=-1/Z$

The case presented in section 3.2.2.2 is selected to test now the transfer admittance relation when all the coefficients are non-zero. The set of coefficients are chosen such that they satisfy the boundary conditions of the global problem. In particular, in the previous problem the inlet/outlet acoustic boundary conditions in the second duct have half the amplitude of those in the first duct. The set of transfer admittance coefficients $\alpha_1=2/Z$, $\alpha_2=-2/Z$, $\alpha_4=1/Z$ and $\alpha_5=-1/Z$ satisfies the conditions of a problem where $u_1=2u_2$.

In Figure 20, some results are shown for this case and several meshes. The agreement with the results shown in section 3.2.2.2 is good, although the accuracy decreases when the surface meshes involved in the transfer admittance relation are nonmatching. The reason is that any small imbalance in the integration corresponding to the two surfaces produces an additional net flux of acoustic velocity from one duct to another and, because of the way the problem is defined, this has an important impact on the results.

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Conformal meshes: Pressure (real part), frequencies $f=0.2c_0/L$ (left) and $f=0.7c_0/L$ (right)

Non-conformal meshes: Pressure (real part), frequencies $f=0.2c_0/L$ (left) and $f=0.7c_0/L$ (right)

Figure 20: Results obtained for the case $u_1=2u_2$, $\alpha_1=2/Z$, $\alpha_2=-2/Z$, $\alpha_4=1/Z$ and $\alpha_5=-1/Z$

3.2.2.4 Contiguous symmetric ducts with flow and transfer admittance: $\alpha_1=\alpha_5=1/Z$ and $\alpha_2=\alpha_4=-1/Z$

The same ducts of previous sections are considered, but, in this case, acoustic boundary conditions are exactly the same for each duct, making the problem symmetric. The transfer admittance coefficients are also chosen to be symmetric: $\alpha_1=\alpha_5=1/Z$ and $\alpha_2=\alpha_4=-1/Z$. The Mach number is still $M=0.3$ in both ducts. As a result of the problem’s symmetry, the net acoustic velocity in the direction transversal to the ducts must be 0 (resulting in no actual acoustic transfer between the ducts), and only longitudinal wave propagation can be a solution of the problem. The results shown in Figure 21 confirm this. When the meshes are matching, the transversal acoustic velocity is practically 0. When the meshes are non-matching, some acoustic velocity appears, but still the acoustic results exhibit mostly plane propagation behavior. Different meshes, including linear hexahedral elements and linear and parabolic tetrahedral elements, were tested.
3.2.3 Conclusions

In this subtask, SISW has extended its potential flow acoustics propagation solver to account for transfer impedance boundary conditions in the presence of grazing flow. This model applies the Myers boundary condition to model an equivalent infinitely thin boundary layer at the liner interface. A series of verification scenarios have been presented which demonstrate the accuracy of the approach. Targeted application is the coupling of two acoustic cavities through perforated plates in the presence of strong background potential grazing flows.
3.3 Characterization of a slit resonator

3.3.1 Objectives of the task
In this subtask, an in-house high-order frequency domain Linearized Navier Stokes (LNSE) solver is used to characterize the acoustic impedance of a slit resonator. This study relies on the experimental campaign on a slit resonator under grazing flow [2] carried out within WP 3.5 (Network models for the prediction of components interactions) and reported in Deliverable D3.5 [1]. The test rig is displayed in Figure 22. In WP 3.5, the KUL has compared the experimental results with some numerical results obtained from their own LNSE solver. The same approach is repeated here, except that the SISW solver is used instead. The two approaches being radically different from a numerical point of view (KUL solver uses the Discontinuous Galerkin Solver in the time domain, whereas the SISW solver uses the high-order FEM in the frequency domain) the objective of this subtask is to provide a better understanding of the respective merits and drawbacks of each solver and to better identify the sources of discrepancies.

Figure 22: KUL slit resonator test rig (left) and close up of the slit-shaped opening (right)[2].

3.3.2 Presentation of the LNSE frequency domain solver
The linearized Navier-Stokes equations provide an interesting physical model to simulate the linear regime of an orifice. The LNSE are generally preferred to the classical Linearized Euler Equations (LEE) for this type of applications, where the viscous terms play a crucial role in the acoustic/flow interaction mechanisms in and around the orifice.

In compact form, the frequency domain version of the linearized Navier Stokes set of equations can be written

\[ i\omega q + \frac{\partial A_r q}{\partial x_r} + \frac{\partial f^v}{\partial x_r} + Cq = 0 \]

In this notation, \( \omega \) denotes the angular frequency, the terms \( A_r q \) and \( f^v \) are the flux Jacobian for respectively the convective and the viscous effects. The term \( Cq \) contains the contribution from the non-uniform flow field. For a two-dimensional case, the vector of unknowns is defined as
This set of equations is solved using a high-order Finite Element Method, based on a high-order hierarchical continuous polynomial approximation basis [6]. A Petrov-Galerkin stabilized formulation is applied, in order to remedy the poor convergence of native FE schemes for convection dominated problems [7]. Non-reflecting boundary conditions based on the method of characteristics are applied at the duct outlet. The system is excited by enforcing the incoming acoustic characteristics at the inlet of the computational domain. The LNSE frequency domain formulation takes the following form:

\[
i \omega \int_{\Omega} \vec{\bar{q}} \cdot \mathbf{q} d\Omega + \int_{\Omega} \vec{\bar{q}} \cdot \mathbf{Cq} d\Omega - \int_{\Omega} \frac{\partial \vec{\bar{q}}}{\partial x_r} \cdot \mathbf{A_rq} d\Omega - \int_{\Omega} \frac{\partial \vec{\bar{q}}}{\partial x_r} \cdot \mathbf{f_r^v} d\Omega \\
= - \int_{\Gamma} \vec{\bar{q}} \cdot [n_r \mathbf{A_r}] \mathbf{q} d\Gamma - \int_{\Gamma} \vec{\bar{q}} \cdot \mathbf{f_r^v} d\Gamma
\]

This formulation is evaluated at each frequency, using high-order Gauss quadrature and the resulting linear system is solved using a sparse linear solver. Acoustic quantities are then easily post-processed at points of interest.

3.3.3 Numerical set-up

The same two-dimensional computational grid is used than for the Runge Kutta Discontinuous Galerkin (RKDG) simulation of the KUL [2]. The mesh contains 1505 triangular elements, it is shown in Figure 23. Although this mesh is not fine enough to resolve the acoustic boundary layer, it is sufficient to model the phenomena governing the interaction between the grazing flow and the acoustic waves.

![Figure 23: close up of the mesh used for the LNSE simulations][1]

The same order (p = 4) is also defined, such that the spatial resolution of the two schemes is comparable and a similar level of accuracy is expected (provided the time discretization does not dominate). Note that the functional basis is however very different in comparison with the DGM, since the approximation is now continuous across the mesh. The frequency range is set from 20Hz to 1700Hz.
Finally, the same mean flow field obtained from a steady RANS simulation is used, visible in Figure 24 for \( M = 0.075 \). The mean flow is mapped onto the acoustic mesh using a least squares interpolation strategy.

![Velocity Magnitude](image)

**Figure 24: Mean flow velocity magnitude around the resonator for \( M = 0.075 \) [2]**

The pressure fluctuations are stored at a set of points suitable for the computation of the scattering matrix of a duct segment around the orifice. This duct segment has a length of 25mm, it corresponds to the part of the duct above the cavity. The scattering matrix is computed with the technique described in section 1.3 of [1], requiring the pressure fluctuations in at least two points in the inlet and outlet duct.

### 3.3.4 Discussion of the results

The pFEM LNSE simulations are ran from 20Hz to 1700Hz with a frequency step of 35Hz (50 frequencies). The coefficients of the scattering matrix are compared with the experiments and the KUL results (RKDG method) in Figure 25.

Some significant differences are observed between the pFEM and the RKDG simulations, particularly at low grazing flow magnitude. The KUL results exhibit a slightly lower resonance frequency and a slightly more damped solution than those obtained by SISW. Knowing that the mesh, the mean flow mapping, the spatial resolution are similar for both methods, these differences can be attributed to the difference in the treatment of the viscosity terms. In the RKDG approach, the viscous terms are implemented using a two-step approach, in which the velocity gradient are first computed as independent variables, while the pFEM solves the LNSE all at once. The fact that the differences between the two methods tend to mitigate for larger grazing flow magnitudes corroborates that assumption. At larger mean flow velocities, the losses related to the interaction between the shear layer and the acoustic field will tend to dominate over the standard viscous losses in the vicinity of the orifice.

As for the comparison with the experimental results, a few arguments can be put forward to explain the discrepancies with the simulations[1, 2]:

It is assumed that the flow is fully developed when it enters the acoustic domain. However, this is not verified experimentally. Also other factors influencing the mean flow profile, such as turbulence levels, have not been verified. It is therefore likely that the simulated mean flow does not exactly correspond exactly to the experimental situation, especially due to the fact that
the turbulent boundary layer thickness has a significant influence on the noise propagating mechanisms.

Figure 25: Scattering matrix coefficients (abs. value) for various mean flow configurations, M=0 (black), M = 0.025 (cyan), M=0.05 (magenta), M=0.075 (blue), M=0.1 (red). RKDG (dashed line), pFEM (crosses), and experimental results (straight line)[2].

The experiments are carried out on a 2,5D slit resonator geometry in a rectangular duct, while the simulations considered a 2D geometry. Although a high aspect ratio is used in the experiments, 3D effects and edge effects cannot be totally excluded.

Taking these factors into account, the agreement between the RKDG simulations and the experiments is deemed satisfactory.

3.4 References


[5]. LMS Virtual.Lab Acoustics R11, Coupling of structural-acoustic computations with incompatible meshes, 2011


4 Assessments of benefits of using innovative MPP sound absorbing panels in ventilation ducts

4.1 Background of application

Components in The fresh air ventilation system inside the aircraft have to meet demanding specifications on many fields as for instance:

- High noise absorbing properties on wide band frequencies
- Low risk of fire
- Low weight
- Give low addition to pressure drop in the system
- Able to withstand high air speed flow without tearing or giving dust
- Long life time without aging or degrading by corrosion
- Self-supporting panel stiffness also as guide vanes.

Traditionally the noise-absorbing materials are bulky foams or fibers and taking considerable space from the flow area in the ducts.

In the Idealvent project we have developed and tested new concepts of using the thin MPP panels to absorb noise by thin linings in the ducts. Thereby using new ways to tune the absorption to specific frequencies that is specifically annoying. The MPP also gave the opportunity to design a filter (see sections 2.4 & 3.3 and Figure 2, Figure 26) that kills the tricky higher order and rotating modes in the system. The formability and self-supporting strength of the thin metal sheets facilitated these 3-D geometrical designs without causing detrimental pressure drop.

4.2 Material costs comparison

Absorptive materials in aviation applications has to comply with the strict and demanding non-flammability regulations according to FAR25.853. This filters away many of the commonly used PU foams and polyesterfiber based absorbants. The allowed weight limits further sets limits to available materials to choose among. Bulky glassfibers need to be compressed to very dense batts in order to achieve the necessary sound absorbing properties has shown to be too heavy. In a duct system used in aeroplanes there is also a limit of total space available for silencers etc.

Here opens the thin film unburnable and light weight opportunity of MPP that has a tailored impedance that has to match the actual application geometry. By the advanced calculation of acoustically maximum possible attenuation of noise in the given duct with measured spectral composition from the fans and from obstacles upstream, the project has managed to reach a very high sound reduction in the prototype laboratory-duct. The cost benefit in favour of the
MPP solutions is estimated to be a reduction of 71% compared to complicated and space demanding solutions with special long fibrous materials behind perforated metal plates.

Even though the cost per square meter of the MPP is higher than conventional porous materials the versatility of the MPP give many more possibilities to actually tailor the silencer to the desired performance. This can be achieved and at the same time get a lower pressure drop, save total weight, better durability and lifetime more robust mechanical design.

4.3 Benefits for the SME entrepreneur

The successful development and testing using the MPP panels in components as silencer or filter can be brought to the market as top of the line products for a wider application than just aircraft ventilation. The aircraft industry will certainly have no problem to accept the new designs and hopefully the industry for rail-vehicles and passenger ships can be potential users of these inventions. Even ventilation for urban buildings can utilize these findings. The findings have of course to be developed into defined products before presented/ marketed to a wider audience.
5 Test of the novel passive noise control techniques involving micro-perforated plates

5.1 The test cases

The two novel noise reducing modules, the MPP (Micro-Perforated-Plate) modal and the Cremer filters, have been tested on the VKI test-rig in similar conditions than those used in WP2. Both modules have been designed by KTH and built by SNT. The modal filter, shown in Figure 26, is designed to damp the two lowest circumferential modes and has an inner diameter and length of 150 mm. The Cremer filter, shown in Figure 27 is optimized to damp the plane wave mode at the fan BPF, and has an inner/outer diameter of 150/175 mm and a length 150 mm. Both are described and analyzed in sections 2.3-2.6.

Figure 26: Pictures of the MPP modal filter

Figure 27: Picture of the Cremer filter
Figure 28: Filters (see Figures 26&27) installed on the VKI test-setup.

The modules have been installed on the VKI test-rig, as shown Figure 28, in order to test their efficiency on a fan, already characterized in WP2. Three different configurations were tested. Measurements are first performed on an empty configuration, with an empty duct upstream of the fan and the bellmouth used at the inlet. Using the same bellmouth inlet, the filters are then inserted upstream of the fan and measurements are performed for two configurations, with the Cremer+modal filters or modal+Cremer filters, in order to verify their relative efficiency depending on their relative position to the inlet. The three tested configurations have exactly the same duct length upstream of the fan in order to have similar duct end effects and then a direct comparison of the measurements. A microphone module, described in WP2, is located between the inlet bellmouth and the tested filter, including 16 B&K ¼ inch microphones, to perform the modal decomposition. Microphones are calibrated in phase and amplitude in a similar manner then in WP2, but using a chirp signal. An additional microphone is located between the tested filters and the fan, and 2 far-field microphones are located outside of the duct, at 1m from the duct axis and at a zero or 0.6m distance from the bellmouth inlet. The fan is operated at the same conditions as for all the test performed in WP2, with 32 m/s centerline velocity downstream of the fan, which is adjusted with an external additional fan downstream of the tested fan. The limited pressure drop of the additional filters is then compensated by the external fan to have similar operating conditions for all configurations. Only active measurements are performed, using a sampling frequency of 32768 Hz and a duration of 60 s. Measurements are repeated 90 times to obtain a good convergence of the results.

5.2 Results

In Figure 29 the results from the tests at VKI using the Liebherr fan are presented. As can be seen from the plots the serial Cremer-Modal filter unit is very efficient. It both reaches the target of more than 10 dB damping at the fan BPF but also delivers a broad band damping. This is due both to the action of the modal filter and the Cremer silencer. Because the Cremer silencer can although it is tuned to a certain frequency be quite effective in a band around this frequency. It can also be noted although the Cremer silencer is optimized for the plane wave, it is still very or often more efficient for the higher order mode components at the BPF. However, the drawback as shown by the simulations in section 2 is that is quite reflective for the non-optimum modes. This is compensated for by combining the Cremer silencer with the upstream modal filter which gives a broad-band damping to circumferential modes up to order 2.
5.3 Tests at Liebherr (LTS)

LTS work in WP5 to test the proposed new MPP noise mitigation solutions described in this report will be reported in WP6. This modification of the plan makes it possible to fully test all the proposed ideas described in this report including Micro-Perforated-Plate (MMP) guide vanes. These guide vanes have been designed and built by SNT and KTH and sent to LTS for the final testing, see Figure 30. LTS will compare the expected attenuation of the KTH filter modules (Figure 29) measured by VKI on the fan alone with the attenuation measured on the final ECS system in LTS anechoic chamber. Similar tests will be conducted for the MPP guide vanes. All the results with comparisons to the VKI data are be presented in WP6, D6.3.

Figure 30: Sketch of the MPP guide vane arrangement designed by KTH and SNT to be inserted into the inlet duct of the Liebherr fan.
6 Noise mitigation approaches applied to the butterfly valve

6.1 Introduction

With respect to the butterfly valve, studied at KUL, the work performed in WP5.2, and discussed in detail in D5.2, includes a more thorough analysis of the actual noise generating mechanisms, including a thorough comparison of scaling laws, investigated by Gordon for spoilers in flow duct systems [1], [2] as well as with the commonly used Nelson and Morfey theory [3]. Furthermore, since the detailed aeroacoustic measurement campaign, discussed in D2.4, D2.7 and 2.8 showed a significant influence of the presence of the 90° bend on the aerodynamic characteristics but not on the active acoustic noise generation, further investigation on possible installation effects, including a variation of the distance between the bend and the valve and a change in the valve’s orientation angle, are carried out. Finally, the influence of installing modal filters, discussed in D5.1 and turbulence screens on the final acoustic radiation is performed on the butterfly valve case. In this deliverable, a selective overview of these activities are reported. A more in-depth analysis of the results can be found in D5.2

6.2 Test configuration

Similar as for the preliminary acoustic survey, reported in D2.1, D2.6 and D2.7, in WP5.2 acoustic radiation measurements are performed using five Brüel & Kjær type 4942 1/2” microphones with built-in preamplifier. The five-microphone array is located at the height of the duct centreline, in a quarter circle of 1 m radius with a 18° spacing between each microphone. The array is placed on the left side of the outlet duct, so that the first microphone is aimed perpendicular to the air flow. The frequency spectrum from 0 to 8192 Hz is acquired with a frequency resolution of $\Delta f = 0.5$ Hz. For every measurement, 100 samples are taken and averaged with a 50% sample overlap. The obtained sound pressure levels are, subsequently averaged in third octave bands for a more straightforward analysis and discussion. The narrow-band spectra are, however, also analysed (for each measurement) for possibly tonal phenomena but for all measurement a broadband nature is observed. As such, the narrow-band spectra are not included in this deliverable. The baseline measurement configuration of the butterfly valve without bend and turbulence screens is shown in Figure 31 and Figure 32.

![Figure 31: Schematic top view of the reference configuration (valve only) with upstream duct length 0D (= no duct) or 6D.](image-url)
In order to identify this aeroacoustic behaviour in the ECS mockup, the experimentally obtained pressure data (in third octave bands) is scaled to the free-stream flow velocity $U_0$, resulting in a Velocity-Scaled Sound Pressure Level (VSSPL):

$$VSSPL = 20 \log_{10} \frac{p^2}{U_0 \pi}$$

In this equation, $n$ is a positive integer. For a free-field monopole, $n = 4$; for a free-field dipole, $n = 6$. In Figure 33, the VSSPL is given for the an opening angle $\alpha$ of 30° that is also investigated in the detailed measurement campaign, reported in D2.6 and D2.7, both for $U_0^4$- (left) and $U_0^6$-scaling (right). At low frequencies, the valve behaves (as predicted by Gordon [1],[2] and Doak [4]) as a free-field monopole acoustic source which can be noticed by the fact that the VSSPL is constant for different flow rates with a $U_0^4$-scaling. For higher frequencies, the valve acts as a free-field dipole source which can be noticed by the fact that the VSSPL is constant for different flow rates with a $U_0^6$-scaling. The frequency at which the valve switches from free-field monopole to free-field dipole behaviour is clearly identified as the calculated cut-on frequency of the first transverse duct mode ($f_0 = 2370$ Hz) for the 30° valve angle.
Figure 33: Velocity-Scaled SPL for different valve opening angle $\alpha$ equal to 30°, with $U_0^4$ (left) and $U_0^6$ (right) scaling

Subsequently the third octave band filtered radiated SPL are transformed in-duct Sound Power Level ($SWL_D$) using a simple radiation open end impedance model. The in-duct Sound Power Level are normalized according to scaling laws derived by Nelson and Morfey\cite{3}, which were modified for circular ducts by Oldham and Ukpholo\cite{5} and Karekul \cite{6}.

For frequencies below the cut-on frequency $f_0$:

$$20 \log_{10} K(St) = SWL_D - 10 \log_{10} \frac{\rho_0 A \sigma^4 C_f^2 U_c^4}{16 c_0} - 120$$

For frequencies above the cut-on frequency $f_0$

$$20 \log_{10} K(St) = SWL_D - 10 \log_{10} \frac{\rho_0 \pi A^2 St^2 \sigma^4 C_f^2 U_c^6}{24 c_0^3 d^2} - 10 \log_{10} \left(1 + \frac{3 c_0}{8 \rho_0 f_c}ight) - 120$$

With $C_f$, the pressure loss coefficient, $\sigma$, the open area ration, $U_c$, the incoming mean flow velocity, $St$ the Strouhal number and $d$, dimension of the spoiler.

In Figure 34, the source strength spectrum $K^2(St)$ is presented according to the Oldham and Ukpholo modification of the Nelson and Morfey scaling laws, for a valve opening angles $\alpha$ of 30°. For $St > 1$, the collapse of the data is excellent. Furthermore, it can be observed that above $St = 1$, there is a 20 dB per decade decay in the value of $20 \log_{10} K(St)$ with increasing Strouhal number, confirming the observations made by Nelson and Morfey. As such, for the higher frequencies, the behaviour of the source strength spectrum $K^2(St)$ follows a similar pattern for more complex geometries such as a butterfly valve as it does for the simple geometries used in previous research. It was stated by Nelson and Morfey that the scaling laws derived from the dipole source model were valid up to maximum local flow Mach numbers below 0.3. However, in the experiments performed in this research, the constriction flow velocity $U_c$ significantly exceeded this limit (up to $M \approx 0.75$), and the data still display a satisfactory collapse.
This leads to the conclusion that the modified Nelson and Morfey theory, which was initially derived for simple spoiler geometries, is indeed also valid for a butterfly valve configuration characterized by a large blockage area and maximum constrictions flow velocities in the transonic region. This means that the behavior of the source strength spectrum $K^2(St)$ is known, and therefore it can be used to relate the aerodynamic noise generation of a spoiler to steady pressure drop measurements.

### 6.4 Installation effects

Next to the identification of the aeroacoustic phenomena, an important part of this research is devoted to the installation effects of other ECS components (e.g. an upstream duct of a certain length, a 90° bend...). Although each ECS component can be characterized independently, the global aeroacoustic behaviour of the system will also depend on the aerodynamic and acoustic interaction between the different elements. As such, the second goal of this deliverable is to determine the influence of the installation of components upstream of the butterfly valve on the global aeroacoustic behaviour of the system, again using far-field acoustic radiation measurements.

The installation of an upstream bend does not have a significant influence on the SPL generated by the ECS mockup. This was also valid when a different duct lengths (0D, 2D, 4D and 6D) are installed between the valve and bend for which the radiated SPL are shown in Figure 35. Based on these observations, neither installing a bend nor the length between valve and bend influences the radiated SPL significantly. This can be explained by the fact that the pressure drop over the valve, which was continuously monitored is kept constant to 500 mbar, as discussed in D2.2 and D2.4. Furthermore, also the upstream pressure which is also continuously measured does not significantly change when the bend is installed or when the length between the valve and bend is altered. As such, it is concluded that the pressure loss coefficient $C_f$ remains identical for all cases under consideration. As indicated by Nelson and Morfey’s theory, discussed in the previous section, the pressure loss can be considered to be the
dominating aerodynamic noise generating mechanism for the butterfly valve. As a result, the conclusions, drawn from Figure 35 are perfectly aligned with the Nelson and Morfey theory.

![Figure 35: Influence of the presence of an upstream duct on the SPL, with an upstream bend installed, for valve installation angle 0°, at Q = 410 m³/h.](image)

The bend itself also acts as a separate source of aerodynamically generated noise with noise generating mechanisms which are also governed by the generating of fluctuating dipoles, generated by the flow impinging the downstream duct of the bend. This self-noise of the bend is not observed in the SPL. On the one hand this is, as already mentioned before, due to the fact that the pressure drop over the bend is negligible in comparison to the pressure drop over the valve. Furthermore, the passive (multi-port) scatter matrix results, discussed in D2.4, show a low transmission coefficient of the downstream propagating modes over the whole frequency region. As such, the valve itself offers an efficient shielding of upstream noise sources which radiate in the downstream direction.

Similar observation as for the bend are observed when analyzing other installation effects such as e.g. changing the valve orientation angle. These results indicate that neither the valve installation angle nor the presence of an upstream bend or duct have a significant influence on the acoustic noise generation of the ECS mockup. The installation effects of these components on the global sound generation of the mockup are negligible. It can be concluded that the noise production of the mockup is dominated by the pressure drop over the valve, which was kept constant for the different configurations at a given flow rate.

### 6.5 Turbulence screens

Once the installation effects (discussed in the previous section) are determined, an attempt is made to mitigate the sound production of the ECS mock-up using turbulence screens, which are placed perpendicular to the flow direction. The screens are tested in two positions: directly upstream and directly downstream of the valve, both with and without the presence of an
upstream bend. Since, as mentioned before, the bend does not significantly influence the final noise radiation only the results, obtained for the ‘valve only’ measurements are discussed in this section. For the experiments, four different screen types are used (as shown in Figure 36)

![Different turbulence screens.](image)

The third octave band averaged SPL of the measurements with the screens installed upstream of the valve are shown in the left of Figure 37. Similar as for the bend, where it is observed that adding large scale turbulent structures and a non-uniform mean flow does not alter the far-field radiation large scale, the generation of small scale turbulence upstream of the valve does not change the radiated SPL. Similar as for the analysis of the installation effects, this is due to the fact that the pressure drop over the valve is almost not influenced by the presence of the upstream turbulence screen and that possible induced self-noise is efficiently shielded by the downstream located valve.

![Influence of the presence of an upstream (left) and downstream (right) turbulence screen on the SPL, at Q = 440 m³/h.](image)

The third octave band averaged SPL of the measurements with the screens installed downstream of the valve are shown on the right of Figure 37. Similar as upstream screen neither the presence nor the type of turbulence screen mounted downstream of the valve significantly changes the radiated SPL. In contrast to the upstream screens slight difference in the radiated SPL can, however, be noticed. This is caused by aerodynamic self-noise generation by the turbulent screens itself. In this case the valve indeed does not act as a shielding of downstream radiated
noise but causes additional reflections of the aerodynamically generated noise by the turbulence screens towards the downstream end. This additional noise generating mechanisms can, however, be considered to be negligible and does not yield to any noise mitigation of the aerodynamically generated noise by the butterfly valve.

6.6 Modal filters

The last type of noise mitigation studied in WP5.2 is using so-called Micro-Perforated Panels (MPP’s), discussed in section 2. On the right of Figure 38 shows the radiated SPL for different lengths of modal filters. The difference between the radiated SPL of the various lengths of modal filters are caused by the fact that the total length of the downstream duct is changed by mounting different lengths of modal filters. The presence of the valve in combination with the impedance of the downstream radiation impedance causes the occurrence of geometric resonances of the ducts with a characteristic wavelength corresponding to an odd multiple of four times the length of the downstream duct. As shown in Figure 38, increasing the length of the modal filter and thus the length of the downstream ducting, the resonance peaks indeed shift to lower third octave bands.

![Figure 38: Influence of the MPP length on the SPL, for panel type A, without upstream duct or bend installed, for panel installation angle 0°, at Q = 440 m³/h (left). Active source vector, discussed in D2.4, obtained with the active multi-port characterization of the butterfly valve (right).](image)

The fact that the presence of the modal filter itself does not influence the radiated SPL (although as discussed in section 2 a potential of 4-6dB sound prediction was predicted) can be explained by looking at the active multi-port source vector, discussed in detail in D2.4 and shown in the right of Figure 38. Analyzing the different active source vector components, shows that most of the aerodynamically generated content excites the plane wave mode \( (0,0) \). In section 2 it is explained that the modal filter is only damping noise of the energy which is contained in the azimuthal modes which are only weakly excited. As such, for the butterfly valve application
the presence of a modal filter does not yield any significant noise mitigation. As such, changing the type of MPP, the installation angle of the modal filter or the length between the modal filter and the butterfly valve does not change the radiated SPL.

6.7 Conclusions

In WP5.2 an extensive measurement campaign on the butterfly valve mock-up is carried out where far-field acoustic radiation measured are used to gain further understanding of the aerodynamic noise generating mechanisms of the butterfly valve configuration as well as to assess installation effects and possible noise mitigation which can be achieved by installing turbulent screens and modal filters in the flow duct systems.

A detailed analysis of the noise radiation with varying valve openings and flow velocities, indicated that the butterfly valve behaves in a similar way as the simple geometry spoilers which are commonly reported in literature. The drag dipole behaves as a free-field monopole, scaling with $U_0^4$, below the cut-on frequency; and as a free-field dipole, scaling with $U_0^6$, above the cut-on frequency of the first transverse mode. The theory derived by Nelson and Morfey, and modified by Oldham and Ukpofo, is therefore also applicable for the butterfly valve configuration characterized by a large blockage area and maximum constriction flow velocities in the transonic region. The source strength spectrum determined in this paper shows a similar frequency dependent behaviour as reported in literature for single vane dampers. As such, the source strength spectrum can be used to estimate the aerodynamically generated sound power using steady pressure drop measurements or steady CFD.

The far-field acoustic measurements further show that the installation of other ECS components upstream of the valve (a bend with varying distance between the bend and valve) and the modification of the valve installation angle produces no noticeable effect on the global aeroacoustic behaviour of the mock-up. It is clear that the noise production of the mock-up is dominated by the pressure drop over the valve, which was kept constant for the different configurations at a given flow rate. Furthermore, self-noise introduced by flow duct components which are mounted upstream of the valve are efficiently shielded by the valve itself and do not contribute to the final noise radiation. As such, the most effective way of reducing the global noise generation of the ECS mock-up is to reduce the pressure drop over the valve. However, it should be mentioned that this is, for most applications, often not possible.

Sound mitigation by means of MPP’s and turbulence screens of various types also proved ineffective. Similar as for the installation effects, the breaking up of large scale turbulent structures into small scale turbulence does not change the major noise generating mechanism, which is related to the pressure loss coefficient and, as a results, does not change the final acoustic radiation. The potential of noise mitigation using modal filters cannot be exploited for the current butterfly valve configuration since the aerodynamic sound production is dominated by acoustic energy which is fed to the plane wave propagation mode, on which the presence of an MPP place in the direction of the flow does not has any influence. As a result, more effective mitigation could be achieved by developing acoustic dampers which effectively attenuate the plane wave propagation mode.
6.8 References


