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1 Introduction

Reaching low levels of cabin and ramp noise is crucial to ensure the satisfaction of aircraft passengers and a safe working environment for the crew and personnel servicing the grounded aircraft. From the passenger and personnel standpoint, good thermal and acoustic comfort is important to minimize both stress symptoms and tiredness. Required to provide satisfactory air quality and temperature, the Environmental Control Systems (ECS) currently used are key contributors to the acoustic nuisance within the cabin and around the grounded aircraft. Reducing the amount of noise produced by the ECS will therefore have a direct impact on the passenger satisfaction and personnel health and safety.

Unlike aircraft exterior noise, which has received considerable attention in past and currently running research projects, the noise emitted by confined flows in ECS assemblies involves complex mechanisms that haven’t been sufficiently investigated to permit the noise reduction wished by passengers and regulators. Acoustic and hydrodynamic interactions between subcomponents have so far been largely neglected despite being crucial. In this framework, the IDEALVENT project aims in filling this gap by gaining further insight and developing numerical prediction techniques for the analysis of noise generation and radiation mechanisms in ECS systems with specific focus on the influence multi-component interactions on the aeroacoustic behavior.

Thorough experimental studies (WP2) are conducted in order provide a deeper understanding of these mechanisms. A combination of accurate scale-resolved methods with low-CPU cost statistical/stochastic methods are proposed as an original modelling and design approach (WP3) and are validated on simplified ECS mock-ups (WP4).

Integrated passive flow and noise control strategies are explored both experimentally and numerically (WP5). The knowledge gained in the experimental and numerical investigations of the installation effects will permit devising and optimize strategies having the best potential for reducing ECS noise. The best noise reduction strategies will be finally tested on a full-scale ECS system (WP6), and their impact towards improved passenger comfort and airport personnel health will be assessed with respect to the objectives of the Work Programme and relevant regulations.

This deliverable summarizes the experimental research activities, carried out within the IDEALVENT project, related to WP2: “Experimental characterization of flow-acoustic and multi-component interactions”.

The major objectives of this Work Package are:

- The design of simplified but realistic ECS mock-ups, foreseeing the implementation of the flow and acoustic control strategies to be tested in WP5.

- Measure the mean and transient features of the flow and acoustic fields in components of simplified but realistic ECS mock-ups.
- Quantify the installation effects that occur when
  - an industrial butterfly valve is placed downstream of an elbow,
  - a blower unit is subjected to inflow distortion due to different inlet geometries
  - the same blower is placed upstream of a ducted diaphragm.
- Establish publicly available documented reference databases at each step, permitting the validation of the numerical methods developed in WP3 and applied in WP4.
- Establish a ranking of the sources affecting ramp noise, by means of acoustic measurements performed around a grounded aircraft.

This deliverable gathers the highlights of the different sub-work package deliverables which are available for all consortium partners:

- In deliverable 2.1: Task 2.1 report: Specifications of individual components and of integrated ventilation system the different test facilities are described and the simplified ECS mock-ups, i.e. a butterfly valve with inlet distortions, a blower unit with inlet distortions and a blower unit with outlet distortions are defined

- In deliverable 2.2: Task 2.2 report: Preliminary acoustic survey, simple acoustic and aerodynamic measurements are performed in order to obtain the final geometric and operational conditions for the simplified ECS mock-ups, which will be investigated in detail. The selection of the final configuration is defined in consultation with the different project partners and is based on:
  - Relevance of the measured sound spectra (e.g. tonal vs. broadband components, overall sound pressure levels,…) and representative flow regimes for engineering ECS-applications
  - Feasibility to apply the various numerical simulation approaches, developed in WP3, to be carried out in WP4 to the selected configurations.

- Deliverable 2.3: Task 2.6 report: Acoustic database including outside ramp noise, establishes a ranking of the sources affecting ramp noise, by means of acoustic measurements performed around a grounded aircraft.

- Deliverable 2.4: Task 2.3 report: Detailed flow and acoustic measurements of valve with bend, discusses the aerodynamic and acoustic measurements performed on the simplified butterfly valve with upstream bend mock up.

- Deliverable 2.5: Task 2.4 report: Detailed flow and acoustic measurements of fan with inlet distortion, discusses the aerodynamic and acoustic measurements performed on the simplified fan with inlet distortion mock up.
• Deliverable 2.6: Task 2.5 report: Detailed flow and acoustic measurements of fan with downstream obstacle, discusses the aerodynamic and acoustic measurements performed on the simplified fan with outlet distortion mock up.

The major amount of research, performed within this work package, is related to the latter three deliverables. The major objectives of these three deliverables can be summarized as follows:

• Providing accurate inlet boundary condition information for the (un-)steady CFD simulations, performed in WP4

• The development of an experimental database, which can be used to validate the different numerical aerodynamic and flow-acoustic simulation techniques, developed in WP3.

• A qualitative assessment of the influence of upstream and downstream obstacles, for both the fan and valve case on both the aerodynamic and aero-acoustic behavior of the system under consideration, including guidelines for the accurate numerical modelling of the aerodynamic noise generating mechanisms.

• Determination of the dominant noise generating mechanisms, in order to determine the efficiency of possible noise mitigation solutions, which will be developed and tested in WP5.

Following outline is used for this deliverable (for an in-depth analysis of the various sub-tasks, the reader is referred to the various sub-work package deliverables):

In section 2 the different test facilities of VKI and KUL, used within this project, are described. Subsequently, section 3 describes the various geometrical test configurations which are envisaged within the project. Section 4 discusses the definition of the final geometric and operational flow conditions for the detailed aerodynamic/acoustic measurement campaign for the valve with inflow distortion, fan with inflow distortion and fan with outflow distortion measurements. The theoretical background on the active multiport characterization methodology, as well as, on aero-acoustic guidelines for fan noise and prediction of installation effects are given in section 5. Section 6 discusses the different data acquisition approaches, used within this work package. The final results, obtained for the detailed measurement campaigns for the different test configurations, are described in section 7. Section 8 discusses the ranking of ramp noise sources based on experiments and the final conclusions, which can be drawn of this Work Package are summarized in section 9.
2 Description of the test facilities

2.1 VKI test facility

One of the major objectives of this work package is to perform a flow and acoustic survey on the ECS fan installed in a duct with upstream and downstream disturbances. This report mainly focuses on the effects of downstream obstacles including a diaphragm with different axial positions and different operating conditions of the ECS fan and the effect of the inlet distortions using T-junction and rectangle-circular inlets.

In order to investigate the aerodynamic sound generated by an ECS fan (provided by Liebherr Aerospace) subjected to installation effects, a modular test-rig has been built at VKI as seen in Figure 1 and Figure 2. The modular structure allows investigating the noise generated by the axial fan, including both upstream and downstream installation effects. The rig inside the VKI low-speed fan anechoic room is schematically represented in Figure 1.

![Schematic representation of the IDEALVENT generic ECS in VKI low-speed fan anechoic room](image)

The ECS fan, provided by LTS, has been mounted in a circular duct in the VKI anechoic room. The duct is manufactured from plexi-glass with an inner diameter of 150 mm. For the construction of the various ducts plexi-glass is selected due to optical visibility for possible PIV measurements for validation purposes. The loudspeaker modules are manufactured from aluminum by KTH in order to carry the loudspeakers which are used in the modal decomposition analysis. Figure 2 (left) shows a photo of the ECS fan rig installed in the VKI anechoic room.
The ECS fan, used in the experimental rig, has 15 rotor blades and 10 stator blades and is operated at 11200 rpm. A static frequency converter has been used in order to run the ECS fan in its design conditions. The power-supply converts the three phase input to 115 V and 400 Hz output according to aeronautical standards. There is neither voltage nor frequency control on the power supply, therefore the rotational speed is fixed for a given pressure rise. The Blade Passing Frequency (BPF) is equal to 2800 Hz.

Upstream and downstream ducted microphone modules have been designed and manufactured by KTH, located upstream and downstream of the fan and diaphragm assembly. In total, 32 flush-mounted ¼” Brüel&Kjær 4938-11 high-pressure field microphones and one ½” Brüel&Kjaer 4191 free-field microphones are used. The in-duct microphone sections are used to measure the reflection coefficient downstream of the duct and they will be later used for modal decomposition measurements on the same rig.

The modular structure of the test rig allows investigating installation effects through different configurations including empty duct, fan only, diaphragm only and fan-diaphragm configurations. The test rig also contains an auxiliary fan located outside of the anechoic room in order to generate flow for configurations in absence of the ECS fan.

The flow measurements can be performed using both hot-wire and Particle Image Velocimetry (PIV) methods. A hot-wire module is designed and manufactured which can scan the flow-field in 360 degrees from tip to the duct center as shown in Figure 3. Based on the modular structure of the test-rig, the same hot-wire module can be used at different positions. A more detailed description of hot-wire acquisition is given in section 6.1.1.

Two Prandtl tubes are located in the duct, one in the downstream end of the duct and a second located upstream of the ECS fan, in order to measure the centerline velocity. The difference between the total and static pressure is acquired in order to obtain the centerline flow velocity. The prandtl tube data are used to cross-validate the hot-wire measurements and for easy-scaling of different configurations in the preliminary survey of the test-rig.
Figure 3: Hot-wire module

The fan performance curve measurements have been performed in the anechoic chamber at VKI. The anechoic chamber contains two rooms separated by a thin wall. Two pressure sensors used to measure the pressure of upstream and downstream rooms. Two flush-mounted pressure sensor arrays are also located upstream and downstream of the fan to measure the static pressure rise of the fan. The arrays contain 4 azimuthally distributed sensors. The static pressure on the duct is then averaged over 4 sensors. Figure 4 shows the positions of the static pressure sensors in the generic ECS fan test-rig.

Figure 4: Location of the static pressure probes on the ECS fan test-rig
The flow rate in the duct can be controlled by using the auxiliary fan downstream of the anechoic facility. Using the auxiliary unit, the flow rate in the duct can be controlled; hence different operating conditions of the ECS fan can be investigated. In order to be able to change the resistance in the rig -if necessary in future- the operating point of the fan is selected as with presence of auxiliary fan operating in a low-speed. The auxiliary fan is not stable for the lowest speeds; therefore the operating point is selected after the auxiliary fan reaches a stable condition. The operating point is therefore selected as the centerline velocity equal to 32 m/s. The static pressure rise is measured as 1.66 kPa as seen in Figure 5 where the curves are provided by LTS. For the selected operating point, the volume flow-rate is first predicted based on fully-developed flow assumption of 8th power-law on the duct exit a flow-rate which corresponds to 473 l/s (blue circle). Later, the volume flow-rate is measured using integration of a complete hot-wire scan on the upstream duct cross-section more accurately. The measured flow rate is then 531 l/s (red circle).

2.2 KUL test facility

Besides the ECS fan mock-up of the fan with inflow and outlet distortion, investigated experimentally at VKI, also a mock-up of an ECS valve, provided by LTS, with inflow distortion is analyzed in the aeroacoustic test facility of KUL. The ECS valve, shown in Figure 6, is a butterfly valve which is installed in a duct with inner diameter \( D = 84.9 \text{mm} \). The valve opening is adjustable between 0° and 90° and typical operating positions are situated between 10° and 30°. Under normal operating conditions, the inlet valve temperature is situated between 80°C and 110°C and typical relative inlet pressure is between 1,25bar and 3,00bar, corresponding to typical mass flow rates of 0-1350g/s. The valve CAD is available in the FTP folder of the project web site.
The valve experiments have been carried out at the open-circuit aero-acoustic test facility of KUL. In this test rig, a uniform flow speed is generated using a parallel Roots’ blower configuration. Due to the working principle of the blower, the maximum relative pressure at the outlet is limited to 1,1bar with a maximum volumetric flow rate equal to, approximately, 13m$^3$/min. Both compressors have three lobes to minimize both noise generation and small time-pulsations of the flow. On one of the Roots’ blowers, a frequency regulator with PID controller is installed. The latter will be used to couple the Roots’ blower with a pressure sensor, located upstream of the valve, in order to ensure identical inlet conditions between the different measurement campaigns and test configurations. Furthermore, a Schiltknecht MiniAir20 Turbine flow meter is installed at the downstream end of the test rig to measure the downstream centerline velocity and temperature. During all measurements, the outgoing flow velocity at the centerline is kept constant to 32 m/s.

As the air compression is responsible for a temperature increase of more than 80°C, an aftercooler is installed immediately after the blowers. This generates an outlet temperature of the compressed air of approximately 7,5°C above the ambient temperature with temperature fluctuations of less than 2,5% between different measurement campaigns. After the heat exchanger, the flow is guided through a duct system containing four 90° inside a semi-anechoic room where the flow (HWA and PIV) and acoustic measurements (far-field pressure measurements and in duct modal acoustic measurements) are carried out.

For the PIV measurements, the duct located upstream and downstream of the valve has to be transparent in order to have optical access inside the duct. For this reason, plexi-glass is chosen for most of the modules that are used for the different geometrical configurations. Only the valve casing, 90° bend and loudspeaker modules are manufactured out of steel or aluminum. In order to reduce the manufacturing time, the inside duct diameter of the plexi-glass components is chosen equal to $D_{duct, in} = 84\, mm$, which is slightly smaller than the 84,9mm internal diameter of the valve casing. However, taking into account the material tolerance, which is of the order of 2%, and the specific mounting of the valve geometry, this will not yield significant influences on both the aerodynamic and acoustic measurements. In order to avoid the influence of structural vibrations, while maintaining a good optical access,
a duct with a thickness of 8mm is selected, resulting in an outer diameter of 100mm. For the given internal diameter and thickness, a standard plexi-glass duct is commercially available.

Besides investigating the valve only configurations, an additional test will be carried out with a 90° bend, positioned upstream of the bend. The curvature of the bend equals the internal diameter $D$ of the casing. The length $l$ between the bend and the diaphragm will be taken equal to 1.5D corresponding to the length of the shortest plexi-glass module in order to enable HWA and PIV measurements in between the bend and valve.

Since the operational flow conditions cannot be met with the test rig, a preliminary acoustic survey has been carried out for a number of different flow configurations varying the valve opening and the pressure drop over valve. The latter being carried out by pressure difference measurements on 2 measurement positions, located respectively 1,25D upstream and 2D downstream of the butterfly valve. The incoming mass flow rate and temperature are continuously monitored using a vortex flow meter.

From the far-field acoustic radiation measurements, carried out in the preliminary acoustic survey, discussed in D2.2, the final flow condition (i.e. valve position and pressure drop) is selected, in consultation with all members of the consortium. The selection is based on the fact that the far-field spectrum should be representative for the industrial application. On the other hand the chosen flow has to be of scientific interest and feasible to solve numerically with the computational prediction methodologies that are developed and validated in, respectively WP3 and WP4.

The aerodynamic measurements will be performed using both Hot-Wire Anemometry (HWA) and Particle Image Velocimetry (PIV) methods. The hot-wire measurements are performed downstream of the valve (for the valve alone and valve with upstream bend measurements) and in between the bend and valve (for the valve with upstream bend measurements). For all configurations, also the upstream velocity profiles are measured using HWA at a distance of 6D before the first object.

The second aerodynamic measurement technique that is used is PIV, which allows visualizing the flow field. Since the total measurement rig (except for the valve and bend) is manufactured using plexi-glass ducts, the PIV measurements can, in principle, be carried out at the same locations as the hot-wire measurements. During the actual measurements it was, however, due to the small duct diameter, impossible to obtain accurate PIV measurements with an indirect laser insertion, using 45° mirrors. As such, PIV measurements are only carried out downstream of the valve. For these measurements, a time-resolved 2D PIV is carried out on two perpendicular measurement planes due to the non-axisymmetric behavior of the flow field.

For the in-duct acoustic measurements, loudspeaker modules and wall mounted microphone modules are installed on the test rig.
3 Description of measurement configurations

3.1 Fan with inflow distortions

One of the points addressed in IDEALVENT project is the ECS fan operating in presence of upstream disturbances. A t-junction and a rectangular to circular transition modules have been manufactured representative of complex ECS duct components in realistic operating environments. Figure 7 shows the plexiglass inlet modules manufactured at VKI. The t-junction (left) consists of two symmetrical inlets where the areas of each inlet are equal to the one of outlet. Both inlet radii and the outlet radius are equal to 75 mm. The inlets are smoothly connected to the outlet via a tongue in the t-junction module. The rectangular to circular transition module (right) has a smooth transition between inlet and outlet. The inlet area is equal to the two times the outlet area. The length of the horizontal edge of the rectangular inlet is two times than the vertical one.

Figure 7: Photo of the inlet modules

Table 1: Configurations of interest for upstream disturbances

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<th>Dummy 45</th>
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The selected configurations are shown in Table 1 related to the upstream disturbances. The distance between the ECS fan and the inlet module is decided to be equal to 4 duct diameters. The lengths are given in units of centimeters. This distance is needed to be short enough to observe inflow distortion impinging to the fan due to different inlet modules and should be long enough to introduce pressure sensors and hot-wire module.
3.2 Fan with downstream obstacles

In the framework of IDEALVENT, two different fan-diaphragm configurations is selected as 2 duct diameter and 4 duct diameter separation between the fan and diaphragm. Two diaphragm diameters are considered, 85 mm and 116 mm in order to operate the fan in different conditions.

Since the system resistance is dramatically increased with the presence of the diaphragm with 85 mm aperture diameter, the operating point of the fan shifts to the left of its performance curve. The fan then operates in a critical condition close to surge. However in presence of the 116 mm diameter diaphragm, the fan operating condition is not changed critically. Therefore the diaphragm with 116 mm aperture diameter is then used in the measurement campaigns.

Table 2 shows the configurations of interest for the fan and diaphragm cases. An additional configuration, including tandem diaphragms is also introduced since it is also an attractive wake interaction problem.

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3.3 Valve with inflow distortions

In order to take into account the influence of upstream distortions, for the ECS valve mock-up onto the aerodynamic and aeroacoustic characteristics, a bend with a centerline curvature of 1D is mounted upstream of the valve. As such, the originally planned detailed measurement campaign, carried out on the open-circuit aeroacoustic test facility of KUL) contains three different test configurations:

- **Valve alone test case** for which following measurements were foreseen:
  
  - **HWA**: at a distance of, approximately, 5D upstream of the valve to determine appropriate inflow boundary conditions for the (un-)steady CFD simulations carried out in WP 4.1; as well as at a distance of 1.5D downstream of the valve to obtain high frequency (one-dimensional) velocity data (also in the vicinity of the walls) to validate the outcome of the unsteady aerodynamic simulations.
  
  - **PIV**: at a distance of, approximately, 1.5D downstream of the valve to obtain information of the low-frequency (vertical and horizontal) 2D distribution of
the turbulent flow structures which will be used to further validate the outcome of the unsteady aerodynamic simulations.

- **Active aeroacoustic multiport characterization**: to determine both the flow-acoustic scattering characteristics as well as the active noise generation vector. These data are, due to the fact that they are independent of the upstream and downstream impedance, ideally suited to validate the hybrid CAA prediction techniques developed in WP3.

**Bend alone test case** for which following measurements were foreseen:

- **HWA**: at a distance of, approximately, 5D upstream of the bend to determine appropriate inflow boundary conditions for the (un-)steady CFD simulations carried out in WP 4.1; as well as at a distance of 1.5D downstream of the bend to obtain high frequency (one-dimensional) velocity data (also in the vicinity of the walls) to validate the outcome of the unsteady aerodynamic simulations.

- **PIV**: at a distance of, approximately, 1.5D downstream of the bend to obtain information of the low-frequency (vertical and horizontal) 2D distribution of the turbulent flow structures which will be used to further validate the outcome of the unsteady aerodynamic simulations.

- **Active aeroacoustic multiport characterization**: to determine both the flow-acoustic scattering characteristics as well as the active noise generation vector.

**Bend + valve test case** for which following measurements were foreseen:

- **HWA**: at a distance of, approximately, 5D upstream of the bend to determine appropriate inflow boundary conditions for the (un-)steady CFD simulations carried out in WP 4.1; as well as at a distance of 1D downstream of the bend and 1.5D downstream of the valve to obtain high frequency (one-dimensional) velocity data (also in the vicinity of the walls) to validate the outcome of the unsteady aerodynamic simulations.

- **PIV**: at a distance of, approximately, 1D downstream of the bend and 1.5D downstream of the valve to obtain information of the low-frequency (vertical and horizontal) 2D distribution of the turbulent flow structure which will be used to further validate the outcome of the unsteady aerodynamic simulations.

- **Active aeroacoustic multiport characterization**: to determine both the flow-acoustic scattering characteristics as well as the active noise generation vector.

As mentioned before, due to diffraction effects of the incoming laser sheet, it was impossible (although various measure have been taken to suppress this phenomenon) to obtain accurate PIV-results upstream of the valve since the laser-sheet could, due to optical access, not be entered directly at the centerline of the duct. As such, it was decided not to perform PIV measurements upstream of the valve. The HWA offers a back-up for the validation the (un-)

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steady CFD simulations in between the bend and valve. For the HWA, a similar measurement set-up as VKI, discussed in section 2.1 was originally envisaged. However, due to the different duct diameters, the interchangeability of the rotating HWA device was impossible. As such, it was decided to perform the HWA for the valve with inlet distortions only on a horizontal and vertical line.

The test rig is made fully modular with three different lengths of the various test sections:

- An aerodynamic test duct with a length equal to 1.5D, for the HWA as well as the PIV
- An acoustic acquisition test duct with a length equal to 4D, needed to install the 9 microphones, which are required for the multi-port aeroacoustic characterization.
- An acoustic excitation test duct with a length equal to 6D, needed to install the 6 loudspeakers, which are required for the multi-port aeroacoustic characterization.

For the *valve alone test case*, shown in Figure 8, following components are used (from upstream to downstream end):

- A loudspeaker module (length: 6D), which is replaced for the PIV measurements with a dummy plexi-glass duct with equal length to avoid damaging the loudspeakers with the PIV seeding.
- An empty plexi-glass duct (length: 4D)
- A HWA-module (length: 1.5D) for the insertion of the HWA probes to determine a proper set of inflow B.C. As such, the inflow boundary conditions are measured at approximately 11D downstream of the last upstream flow obstacle, i.e. a bend needed to enter the semi-anechoic measurement room; and at a distance of, approximately, 6D upstream of the valve body.
- A microphone module (length: 4D), which is replaced for the PIV measurements with a dummy plexi-glass duct with equal length to avoid damaging the microphones with the PIV seeding.
- An empty plexi-glass duct (length: 1.5D)
- The butterfly valve
- A PIV-module (length: 1.5D) to determine the 2D spatial distribution of the unsteady aerodynamic field. This module is replaced by a HWA module with equal length for the hot-wire measurements.
- A microphone module (length: 4D), which is replaced for the PIV measurements with a dummy plexi-glass duct with equal length to avoid damaging the microphones with the PIV seeding.
- An empty plexi-glass duct (length: 4D)
- A loudspeaker module (length: $6D$)
- The anechoic termination

The latter two modules are not installed for the PIV measurements, due to spatial constraints for the direct insertion of the laser sheet at the centerline of the duct. As such, the PIV measurements are carried out with a reduced downstream end. It is, however, expected that the length of approximately $8D$ downstream of the measurement plane is sufficient to yield a similar flow behavior downstream of the valve in comparison with the acoustic multiport measurements and the hot-wire measurements.

![Figure 8: CAD representation of the valve alone case for the acoustic multiport characterization (left) and actual set-up for the PIV measurements (right).](image)

In the original description of the measurements also measurements on the bend alone test case are foreseen. In this case, the test rig, shown in Figure 9 is very similar to the valve alone test case, where the valve is replaced by the $90^\circ$ bend. In order to obtain a similar pressure field upstream of the bend as compared to the valve with upstream bend measurements, the valve is then installed in between the downstream loudspeaker module and the anechoic termination, and its angular position can be adjusted to obtain an identical upstream pressure as for the valve with bend measurements.

However, due to time and budget constraints, caused by unforeseen problems during the manufacturing and assembly-process, as well as with the PIV acquisition where no indirect access of the laser sheet was possible, it is decided, in consultation with the other project partners, to discard these measurements. This is motivated by the fact that:

- No partners will numerically simulate the bend alone aerodynamically and/or aero-acoustically in WP4.
- The aeroacoustic multi-port characteristics could, in principle, directly be deduced from the combination of the multi-port characteristics of the valve alone and the valve with bend measurements
- PIV data of the bend could not be obtained as, due to the presence of the valve as the most downstream element, no direct access of the laser sheet can be obtained.
Both the aerodynamic and aeroacoustic behavior of this type of, standardized bend (the radius of the 90° bend equals the duct diameter $D$) are extensively reported in literature in the past.

**Figure 9: CAD representation of the bend alone case for the acoustic multiport characterization.**

For the **valve and bend test case**, shown in Figure 10, following components are used (from upstream to downstream end):

- A loudspeaker module (length: $6D$), which is replaced for the PIV measurements with a dummy plexi-glass duct with equal length to avoid damaging the loudspeakers with the PIV seeding.
- An empty plexi-glass duct (length: $4D$)
- A HWA-module (length: $1.5D$) for the insertion of the HWA probes to determine a proper set of inflow B.C. As such, the inflow boundary conditions are measured at approximately $11D$ downstream of the last upstream flow obstacle, i.e. a bend needed to enter the semi-anechoic measurement room; and at a distance of, approximately, $6D$ upstream of the valve body.
- A microphone module (length: $4D$), which is replaced for the PIV measurements with a dummy plexi-glass duct with equal length to avoid damaging the microphones with the PIV seeding.
- An empty plexi-glass duct (length: $1.5D$)
- The 90° bend
- A HWA-module (length: $1.5D$) for the insertion of the HWA probes to determine the unsteady flow velocities between the valve and bend.
- The butterfly valve
- A PIV-module (length: $1.5D$) to determine the 2D spatial distribution of the unsteady aerodynamic field. This module is replaced by a HWA module with equal length for the hot-wire measurements.
- A microphone module (length: $4D$), which is replaced for the PIV measurements with a dummy plexi-glass duct with equal length to avoid damaging the microphones with the PIV seeding.
- An empty plexi-glass duct (length: $4D$)
- A loudspeaker module (length: $6D$)
- The anechoic termination

Similar as for the valve alone test case, the latter two modules are not installed for the PIV measurements, due to spatial constraints.

![Figure 10: CAD representation of the valve with upstream bend case for the acoustic multiport characterization (left) and actual set-up for the multiport characterization (right).](image)
4 Determination of the operational conditions, preliminary acoustic survey

The purpose of the preliminary acoustic survey for both the ECS fan and ECS valve mock-up is to determine the exact operational flow and geometric conditions, which will be used for the detailed aerodynamic and acoustic measurements. The choice of the operational point is based on the:

- Engineering relevance of the radiated sound pressure spectrum for ECS-systems, evaluated by LTS.
- Feasibility of the different numerical approaches, discussed in WP3, to predict the governing noise generating and propagation characteristics.

4.1 Fan with inflow distortions

In the far-field measurements, a ½” B&K microphone is used which is located at the suction side of the anechoic room. The inlet modules are located 4 duct diameters upstream of the fan module which includes one microphone module and two short dummy modules -to be replaced with hot-wire modules for later. Figure 11 shows the far-field SPL (left) measured at the inlet of the modules. The center-line flow velocity is equalized to 32 m/s for all three cases which is satisfied by using the auxiliary fan similar to the ones presented above. The black, red and blue represent results obtained with the bell-mouth, the t-junction and the rectangular-circular transition modules, respectively. It is seen that the lowest sound levels have been obtained in clean inlet conditions using the bell-mouth. Introducing the rectangular-circular transition and the t-junction modules, the sound pressure levels are increased around 5 dB and 15 dB at low frequencies, respectively. It should be noted that the sound generated by the fan is first propagated inside the duct and later reflected and scattered by the inlet modules before it reaches to the observer. In presence of the t-junction especially, the inlet geometry can introduce higher constructive and destructive interference fringes, and the observer can stay in shadow zone. Therefore additional in-duct measurements are performed for the fan operating in presence of inlet distortion. Figure 11 (right) shows the SPL of obtained through in-duct measurements performed downstream of the ECS fan. The black, red and blue represent results obtained with the bell-mouth, the t-junction and the rectangular-circular transition modules, respectively. Similarly, using the transition and t-junction modules increase the sound pressure levels at lower frequencies around 5 dB and 15 dB, respectively. The first and second axial cut-on modes can be seen clearly in the figure around 1300 Hz and 2200 Hz, respectively.
4.2 Fan with downstream obstacles

Two different diaphragm positions have been investigated. Diaphragm positions of 2 and 4 duct diameters downstream of the fan are selected representative of simplified but realistic configurations. The first comparison is performed between fan without diaphragm and fan with diaphragm for two positions, in absence of the auxiliary fan in the test rig. In this case the fan is operated by the power supply only. Since the system resistance is dramatically increased with the presence of the diaphragm with 1/3 duct area opening, the operating point of the fan shifts to the left of its performance curve. The fan then operates in a critical condition close to surge. Since there is no control of the rotational speed, the first comparison is made with the different rotational speed of the fan for different configurations. The rotational speeds are 11200, 11368 and 11376 RPM for the fan alone, 4D diaphragm and 2D diaphragm, respectively. Even the variation in the rotational speed is not more than 2%, since the system resistance has increased significantly; the flow speed provided by the fan has decreased from 30 m/s to 15 m/s. Figure 12 (left) shows the measured sound pressure levels for the fan alone, and 4D and 2D diaphragm positions in black, red and blue, respectively. The presence of a downstream diaphragm can generate additional sound due to the unsteady pressure fluctuations on its surface. However, it seems that the acoustic signature of the fan is preserved in presence of downstream diaphragm with a shift on the sound pressure levels. It can be said that the noise is emitted by the fan, rather than the diaphragm. The difference between with and without diaphragm cases can be seen on Figure 12 (right). The red and blue represents the difference in sound pressure levels between 4D and 2D diaphragm positions, respectively. The difference reaches up to 20 dB around 2 kHz. At the highest frequencies, the broadband noise is increased by 3-4 dB in presence of the diaphragm. The gray line shows the difference between the SPL of the 2D diaphragm case and the one of 4D. The difference is around 6 dB till 2.5 kHz. At higher frequencies, the difference is negligible.
Figure 12 Measured SPL (left) for the fan alone (black) and fan with 4d diaphragm (red) and 2d diaphragm (blue) for no auxiliary fan. (right) difference of SPL

One should note that comparing the acoustic spectra of different operating conditions of the fans can be misleading. In these configurations, the fan operating condition, rotational speed, flow-rate hence the pressure difference provided by the fan are not the same. In order to make a more accurate comparison, the flow speed in the duct has been adjusted using the auxiliary fan in the rig such that a common rotational speed can be obtained for the fan alone and diaphragm configurations. Two common rotational speeds of the fan are tested. Figure 13 shows the sound pressure levels and differences for 11328 and 11360 RPM, respectively. The correction on the RPM does not change the acoustic spectra significantly, the 2% shift on the rotational speed is found to be negligible for this comparison.
Another point which is worth to note is the modulation of the tonal fan noise in presence of diaphragm. Figure 14 shows a closer look on the first and second BPFs for the fan alone (black), 4D diaphragm (red) and 2D diaphragm (blue). The first two rows are for the 1st BPF whereas the third and fourth rows are for the 2nd BPF. First row of each group is from measurements using the power supply, which results in variable rotational speed for configurations. The second row of each group is using a common fixed rotational speed of 11328 RPM. It is seen that in absence of the diaphragm, the acoustic spectra has discrete tones at the first and second BPF. However, introducing the diaphragm downstream the fan, a modulation of BPF is observed. The optical measurements related to the rotational speed showed that the RPM is fixed during the acquisition time for the preliminary case. Therefore the modulation is assumed to be an acoustical phenomenon instead of a variation on the rotational speed of the fan. For the diaphragm configuration, the shaft rotational frequency is $11368/60 = 189.47$ Hz for 4D and $11376/60 = 189.60$ Hz for 2D. The modulation frequency step is 102 Hz for all configurations. It can be due to the pumping of the air between the fan and diaphragm resulting in a frequency modulation. Besides, since the fan is
operating in stall condition in presence of diaphragm, this modulation can be due to the rotating stall occurring between succeeding blades. In the literature, the rotating stall frequency has been reported between 30% and 80% of the shaft rotational frequency. In this configuration, the modulation frequency is 55% of the shaft rotational frequency and rotating stall can be resulting in modulation of the tonal fan noise.

Figure 14 Tonal fan noise modulation fan alone (black), 4d diaphragm (red) and 2d diaphragm (blue)

A comparison between the sound pressure levels (SPL) and overall sound pressure levels (OASPL) can be found in Table 3 and Table 4. The OASPL is obtained using the approximation:
Introducing the modulated tones to the BPF, the OASPL can be higher when the BPF is lower.

\[
\Delta SPL = \begin{cases} 
3 & \Delta SPL \leq 1 \\
2 & 1 < \Delta SPL \leq 3 \\
1 & 3 < \Delta SPL < 10 \\
0 & \Delta SPL \geq 10 
\end{cases}
\]

Table 3: SPL and OASPL for 1\textsuperscript{st} and 2\textsuperscript{nd} BPF for no aux fan

<table>
<thead>
<tr>
<th>DIAPHRAGM</th>
<th>RPM</th>
<th>1\textsuperscript{st} BPF (dB SPL)</th>
<th>1\textsuperscript{st} BPF (dB OASPL)</th>
<th>2\textsuperscript{nd} BPF (dB SPL)</th>
<th>2\textsuperscript{nd} BPF (dB OASPL)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-</td>
<td>11200</td>
<td>79</td>
<td>79</td>
<td>77.5</td>
<td>77.5</td>
</tr>
<tr>
<td>4D</td>
<td>11368</td>
<td>77</td>
<td>82</td>
<td>64</td>
<td>66</td>
</tr>
<tr>
<td>2D</td>
<td>11376</td>
<td>73.5</td>
<td>77.5</td>
<td>64.5</td>
<td>70.5</td>
</tr>
</tbody>
</table>

Table 4: SPL and OASPL for 1\textsuperscript{st} and 2\textsuperscript{nd} BPF at 11328 RPM

<table>
<thead>
<tr>
<th>DIAPHRAGM</th>
<th>RPM</th>
<th>1\textsuperscript{st} BPF (dB SPL)</th>
<th>1\textsuperscript{st} BPF (dB OASPL)</th>
<th>2\textsuperscript{nd} BPF (dB SPL)</th>
<th>2\textsuperscript{nd} BPF (dB OASPL)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-</td>
<td>11328</td>
<td>75</td>
<td>75</td>
<td>74</td>
<td>74</td>
</tr>
<tr>
<td>4D</td>
<td>11328</td>
<td>78</td>
<td>80</td>
<td>64.5</td>
<td>67.5</td>
</tr>
<tr>
<td>2D</td>
<td>11328</td>
<td>75</td>
<td>80</td>
<td>69.5</td>
<td>70.5</td>
</tr>
</tbody>
</table>

As stated above, introducing a diaphragm of 85 mm aperture diameter changes the operating point of the fan dramatically, hence the fan is operated in off-design (probably in rotating stall) condition. Additionally, the flow-rate provided by fan alone and fan with diaphragm is not similar, therefore a comparison between two configurations can mislead. For this sake, a new diaphragm diameter is also investigated. After discussions with LTS, the new diaphragm aperture diameter is selected as 116 mm hence the fan will not operate in critical conditions. Figure 15 (left) shows the sound pressure levels measured at the suction side of the anechoic room. The black, blue and red stand for fan alone, fan with 85 mm diaphragm and fan with 116 mm diaphragm, respectively. The distance between the fan and diaphragms is 4 duct diameters. Introducing the 85 mm diameter diaphragm again increases the sound levels up to 20 dB at lower frequencies, similar to the ones observed above. The center-line flow velocity is equal to 32 and 16 m/s for fan alone and with 85 mm diameter diaphragm, respectively. However, even using the auxiliary fan in the anechoic chamber, it is not possible to reach 32 m/s center-line flow velocity with this diaphragm. On the other hand, using the auxiliary fan, the center-line velocity is managed to be equal to 32 m/s in case of fan operating in presence of 116 mm diaphragm. Comparing same center-line velocities for fan alone and fan with 116 mm diaphragm, the sound pressure levels are seemed to be higher around 5 dB at lower frequencies. An additional set of measurements is also performed in the same condition using in-duct microphones located between fan and the diaphragm, shown in Figure 15 (left). Similarly, in presence of narrow aperture diaphragm, the sound levels are higher around 20 dB which is probably due to operating the fan close to rotating stall condition. However, in presence of wider diaphragm, the SPL is higher around 2 to 3 dB higher. The cut-on
frequencies of the duct can be seen clearly at 1300 Hz and 2200 Hz. Additionally, the tonal fan noise modulation has not been observed in presence of the wide aperture diaphragm.

![Figure 15: Measured SPL with different diaphragm diameters; fan alone (black) and fan with aperture of 85 mm diameter diaphragm (blue) and 116 mm diameter diaphragm (red), far-field (right) and in-duct (left) measurements.]

### 4.3 Valve with inflow distortions

The preliminary acoustic survey and aerodynamic characterization of the butterfly valve case is carried out in the semi-anechoic room of KUL. A uniform flow field is generated by a parallel Roots’ blower configuration, located outside of the semi-anechoic room. One of the Roots’ blowers has a frequency regulator and PID-controller to yield various incoming flow rates or pressure drops over the valve. A heat exchanger is positioned after the Roots’ blowers to minimize the temperature increase due to the air compression. After the heat exchanger the flow duct system enters the semi-anechoic room in which the valve is positioned between different plexi-glass ducts (Figure 16).

![Figure 16: Picture of the valve positioned inside the semi-anechoic room]

In the preliminary acoustic survey and the aerodynamic characterization of the butterfly valve 12 different valve positions, corresponding to, respectively, $90^\circ$ (open position), $75^\circ$, $60^\circ$, $45^\circ$, $30^\circ$, $22.5^\circ$, $15^\circ$, $10^\circ$, $5^\circ$, $2.5^\circ$, $1^\circ$, and $0.5^\circ$. 
40\(^\circ\), 35\(^\circ\), 30\(^\circ\), 25\(^\circ\), 20\(^\circ\), 15\(^\circ\), 10\(^\circ\) and 5\(^\circ\), are considered. To obtain a wide range of operational conditions, the frequency of the frequency regulated Root’s blower is being increased from 20Hz to 60Hz with a step of 10Hz and, afterwards the second Root’s blower is being add on. Given the fact that the total relative pressure increase is, due to the Root’s blower working principle, limited to 1000mbar, not all valve positions could be measured with all 11 different flow conditions. Table 5 gives an overview of the different measurements, shown in green, carried out on the butterfly valve configuration taking into account the test rig limitations, discussed above.

**Table 5: Measurement matrix for the butterfly valve (lines: valve position; columns: Root’s blowers (RB) operation condition)**

<table>
<thead>
<tr>
<th></th>
<th>RB1</th>
<th>RB2</th>
<th>RB3</th>
<th>RB4</th>
<th>RB5</th>
<th>RB6</th>
<th>RB7</th>
<th>RB8</th>
<th>RB9</th>
<th>RB10</th>
<th>RB11</th>
</tr>
</thead>
<tbody>
<tr>
<td>0Hz</td>
<td>20Hz</td>
<td>0Hz</td>
<td>0Hz</td>
<td>0Hz</td>
<td>0Hz</td>
<td>0Hz</td>
<td>0Hz</td>
<td>0Hz</td>
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<tr>
<td>30Hz</td>
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<td>60Hz</td>
<td>60Hz</td>
<td>60Hz</td>
</tr>
</tbody>
</table>

For the aerodynamic characterization of the butterfly valve, following measurements are carried out for the different flow speeds (Root’s blower configurations) and valve positions, shown in Table 5:

- The incoming flow rate and temperature is measured at a distance of 15D upstream of the butterfly valve. These measurements are carried out using a vortex flow meter.
- The pressure drop over the valve is measured using 4 pressure taps on both the upstream and downstream end of the butterfly valve. The pressure taps are located at a distance of 1.25D upstream and 2D downstream of the butterfly valve. They are equally positioned over the circumference of the upstream and downstream ducts, to determine the surface-averaged pressure on both sides.
- The total pressure downstream of the heat exchanger is measured, to determine the pressure drop of the total duct system.

Figure 17 shows the relation between the incoming flow rate (m\(^3\)/h) and the pressure drop (mbar) for the different valve positions. The measurements at small flow velocities (i.e. with only a flow generation of the frequency-regulated Roots’ blower) are carried out twice in order to check the repeatability of the flow conditions, the result from this analysis are incorporated in the graphs of Figure 17, and no significant discrepancies can be noticed.
As also shown in Table 5, for relatively open valve configurations (valve angle $\geq 35^\circ$), the maximum flow rate, delivered by both Roots blowers at maximum frequency, is insufficient to yield a further increase in the total pressure drop over the valve. At more closed valve positions, i.e. valve angles below $35^\circ$, the maximum pressure drop of the total duct system (1000 mbar) is reached before the maximum frequency of the Roots’ blower configuration. According to Bernoulli’s law ($\Delta p \sim v^2$), a quadratic relation between the pressure drop $\Delta p$ and flow rate $Q$ is observed. At large valve angles ($\geq 35^\circ$), a significant increase of flow rate with increased pressure drop is expected. At smaller valve positions ($< 35^\circ$), however, a saturation of the flow rate is noticed when the pressure drop would be increased. This indicates that the incoming flow velocity would not change significantly, if the pressure-drop over the valve is increased to the values which are of engineering relevance for ECS-applications, where a pressure drop of approximately 2000 mbar is reached. Furthermore, since the aerodynamically generated sound field scales, according to both Lighthill’s and Curle’s analogy, logarithmically with the incoming flow velocity, it can be expected that the sound field for valve positions smaller than $35^\circ$ is representative for ECS applications of engineering relevance.

In Figure 18 and Figure 19 the relation between, respectively, the pressure drop and flow rate for different valve positions is shown for constant values of the incoming flow rate or pressure drop. It is noticed that, when the incoming flow velocity is constant (Figure 18), the pressure drop is low and approximately constant for open valve positions (large valve angles) and suddenly increases significantly at smaller valve angles. For flow rates below 400 $m^3/h$, corresponding to the maximum incoming flow velocity which can be reached with the test rig for valve angles smaller than $35^\circ$, the sudden increase of the pressure drop for different valve positions occurs between 30$^\circ$ and 40$^\circ$ for all flow velocities. For a constant pressure drop over the valve (Figure 19) an almost linear relation is observed between valve position and flow rate, in the operational range of the test rig, especially for larger values of the pressure drop, i.e. above 300 mbar.
Figure 17: Relation between flow rate (m³/h) and pressure drop (mbar) for different valve positions.
Figure 18: Relation between pressure drop (mbar) and valve angle (°) for constant incoming flow rates.
The presence of the bend will cause an additional increase in the pressure drop of the total duct system, thus further decreasing the maximum reachable pressure drop over the butterfly valve. In order to ensure that both aerodynamic (hot-wire anemometry and PIV) and acoustic (multi-port characterization) measurements for the detailed experimental investigation of the various butterfly valve configurations can be carried out, the maximum pressure drop over the valve should not exceed 500 mbar for these measurements. Furthermore, for very small valve angles (≤ 10°), it was noticed that only very small incoming flow velocities could be reached.
and, for the high pressure drops which are induced for these closed positions, it was proven to be difficult to obtain repeatable and reliable measurements already for this simple measurement campaign. As such, the valve angles smaller than $10^\circ$ will not be considered further.

For the other valve angles, the incoming flow velocity ($v_0 = Q/A_0$, with $Q$ the incoming flow rate discussed above and $A_0 = 5398\, mm^2$ the cross-section of the different ducts) can be determined, as well as an estimate of the maximum velocity at the valve position ($v_{\text{max}} \approx Q/(A_0(1 - \cos \alpha))$, with $\alpha$, the valve angle) can be obtained together with the corresponding Mach numbers, for the maximum pressure drop of 500 mbar. These values are listed in Table 6 and the maximum velocity $v_{\text{max}}$ as a function of the valve position is shown in Figure 20. The maximum flow velocity at the valve increases asymptotically with decreased valve angles up to a Mach number of, approximately, 0.5 for a valve angle of $15^\circ$.

<table>
<thead>
<tr>
<th>Valve angle</th>
<th>$\Delta p$ (mbar)</th>
<th>$Q$ ($m^3/h$)</th>
<th>$v_0$ (m/s)</th>
<th>$M_0$ (–)</th>
<th>$v_{\text{max}}$ (m/s)</th>
<th>$M_{\text{max}}$ (–)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$90^\circ$</td>
<td>6</td>
<td>875</td>
<td>45,03</td>
<td>0,132</td>
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<tr>
<td>$75^\circ$</td>
<td>13</td>
<td>867</td>
<td>44,62</td>
<td>0,130</td>
<td>60,20</td>
<td>0,176</td>
</tr>
<tr>
<td>$60^\circ$</td>
<td>42</td>
<td>820</td>
<td>42,20</td>
<td>0,123</td>
<td>84,40</td>
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</tr>
<tr>
<td>$45^\circ$</td>
<td>261</td>
<td>763</td>
<td>39,27</td>
<td>0,115</td>
<td>134,06</td>
<td>0,392</td>
</tr>
<tr>
<td>$40^\circ$</td>
<td>288</td>
<td>667</td>
<td>34,33</td>
<td>0,100</td>
<td>146,72</td>
<td>0,429</td>
</tr>
<tr>
<td>$35^\circ$</td>
<td>428</td>
<td>550</td>
<td>28,30</td>
<td>0,083</td>
<td>156,51</td>
<td>0,458</td>
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<tr>
<td>$30^\circ$</td>
<td>500</td>
<td>424</td>
<td>21,82</td>
<td>0,064</td>
<td>162,87</td>
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<td>$20^\circ$</td>
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<td>10,14</td>
<td>0,030</td>
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<tr>
<td>$15^\circ$</td>
<td>500</td>
<td>112</td>
<td>5,76</td>
<td>0,017</td>
<td>169,16</td>
<td>0,495</td>
</tr>
</tbody>
</table>

Table 6: Incoming flow velocity and Mach number ($v_0, M_0$) and estimated maximum velocity and Mach number ($v_{\text{max}}, M_{\text{max}}$) for different valve positions and the maximum values of the pressure drop $\Delta p \leq 500\, mbar$ (calculated with a sound speed equal to $c_0 = 342\, m/s$).

**Figure 20:** Estimated maximum velocity (values from Table 6) for different valve positions.
For the preliminary acoustic survey of the butterfly valve, acoustic measurements are carried out for the different flow speeds (Root’s blower configurations) and valve positions. In total 6 different free-field microphones where installed in the semi-anechoic room at a distance of 1.5 m from the center of the downstream ducting. The microphones are positioned on two perpendicular planes with an angle with the duct centerline of 70° and 45° for the horizontal (xy-) plane and 90°, 70°, 45° and 30° in the vertical (xz-) plane. The data acquisition of acoustic pressure fluctuations is carried out using a sample frequency of 12.5 kHz and a frequency resolution of 1.5259 Hz. This corresponds to a resolved frequency range up to 6.25 kHz, corresponding, as described in D2.1 to the maximum frequency with which the acoustic multi-port characterization is carried out.

Six different microphones are used to investigate if a preferred radiation pattern occurs at the downstream end of the duct. The sound pressure spectrums for two different measurements are shown in Figure 21, for the other measurements, similar results are found. As can be seen, no preferred radiation direction is found. The small differences occurring between the sound pressure levels (SPL) at the different microphones are likely to be caused by ground reflections of the semi-anechoic room. This was however not further investigated since the sound pressure spectrum is very similar for all microphone positions. As a result, in the remaining part of the preliminary acoustic survey, only the results of 1 microphone position (i.e. the microphone located in the xz-plane at an angle of 45°) will be discussed. It is verified that the conclusions, drawn from this analysis also hold for the other microphone positions.

![Sound pressure level plots](image)

**Figure 21:** Sound pressure level (dB) for the six different microphones for 60° valve position and flow rate equal to 725 m$^3$/h (top) and 20° valve position and flow rate equal to 204 m$^3$/h (bottom).
Figure 22 and Figure 23 show the measured sound pressure spectra for different incoming flow velocities and valve positions. In these figures, the two most open valve positions (75° and 90°) are not included but the results at these angles show a similar trend as the 60° valve position. Furthermore, it can be expected that these positions will not be selected for the detailed measurements of the butterfly valve due to the small pressure drop over the valve, which can be achieved within the test rig limitations, in comparison with the engineering operational conditions of ECS-systems.

For all valve positions and flow conditions a very broadband noise radiation is observed. For large valve angles, and especially at the lowest flow velocities, an overall decay of the SPL with increasing frequency is observed. For valve angles smaller than 45° this decay is less apparent and the SPL is almost constant for all flow velocities over the whole frequency range, which is considered in this analysis. Besides this overall decay of the SPL, the signature (i.e. the shape of the acoustic spectrum) of the SPL does not change significantly with increasing flow velocities, for almost all measurements. This indicates that the aerodynamic noise generating mechanisms do not change significantly when the incoming flow velocity is changed.

At frequencies below the cut-off frequency of the high-order duct modes, corresponding for the butterfly valve duct system to $f_{\text{cut-off}} = \frac{1.84 \, c_0}{\pi D} \approx 2400 \, Hz$, tonal phenomena are observed, which are more pronounced at smaller valve angles. These tonal phenomena are generated by duct resonances. This is motivated by the fact that:

- At large (i.e. open) valve positions, the presence of the butterfly valve does not yield significant acoustic reflections. For this reason, no additional resonance phenomena are being generated by the butterfly valve. As such, flow duct resonances do still occur but at low frequencies (due to the large length of the duct system) and their harmonics. These phenomena are still observed for large valve angles, but less pronounced in comparison to the small angle valve positions.

- Tonal phenomena, caused by flow duct resonances, propagate preferably below to cut-off frequency of the high-order modes (i.e. below 2.4 kHz) as they are generated and propagated mainly by plane wave propagation. The fact, that these tonal phenomena are less apparent at higher frequencies, motivates these assumptions.

- In Figure 24, the SPL is shown for two different valve positions with constant inflow velocity but with different lengths of the duct downstream of the diaphragm. It can be observed that some of the low-frequency tonal peaks are shifted towards higher frequencies introduced by the smaller length of the downstream ducts, which decreases the acoustic wavelength. Other tonal phenomena, however, still occur at the same frequency for both the long and short downstream duct. This is due to the fact that acoustic resonances, induced by upstream duct resonances, are not influenced by the downstream duct length and still occur at the same frequencies. Since the length of the upstream duct system is less straightforward to adapt, without modifying the incoming flow conditions, the influence of the length of the upstream ducts is not...
investigated. However, due to the fact that the passive multi-port characteristics are independent of the upstream and downstream analysis, the detailed measurement campaign using an active multi-port characterization can further justify these assumptions. Above the cut-on frequency of the duct system, it can be noticed that the length of the downstream duct does not influence the aerodynamic noise generation.

![Diagram](chart.png)

**Figure 22**: SPL for different incoming flow rates with valve positions of (from top to bottom) 60°, 45°, 40° and 35°.
Figure 23: SPL for different incoming flow rates with valve positions of (from top to bottom) 30º, 25º, 20º and 15º.
Figure 24: SPL for 35°, \( Q=422\, \text{m}^3/\text{h} \) (top) and 25°, \( Q=296\, \text{m}^3/\text{h} \) (bottom) valve angle with a long (blue line) and short (green line) downstream duct.

Figure 25 shows a comparison of the SPL for different valve positions with approximately constant incoming flow velocities. It should be mentioned that (both for this section and the next section) not all SPL are measured at the same flow conditions, since the different measurements are carried out using identical frequencies of the frequency-regulated Root’s blower and (as opposed to the aerodynamic analysis) no interpolation can be carried out on the obtained data due to the dynamic character of the acoustic measurements. Both in Figure 25 and Figure 27, data are presented which correspond as close as possible to the mentioned flow conditions. Thus, discrepancies between the various measurements (and especially the intermediate distance between various SPL) can occur, especially at small valve angles where a small change of e.g. valve angle yields a significant change in both pressure drop and flow rate.

As mentioned above, the signature of the sound field radiation is dependent on both the incoming flow rate and valve position. For identical incoming flow velocities, the SPL shows a steeper decay as a function of increasing frequency when the valve angle is increased. This effect is more pronounced at low incoming flow rates. At small valve angles (<35°), the signature of the radiated sound pressure field is not significantly changed. This is also shown in Figure 26, where the overall sound pressure level (OASPL) is shown for different incoming flow rates and different valve positions. The OASPL does not significantly differ for large valve angles and especially at low flow rates. Starting from approximately 30°, the radiated sound pressure field starts to show a significantly different scaling with the incoming flow velocity in comparison to the large valve angles (>45°), where (predominately at low flow velocities) the SPL does not significantly change with different valve angle. This indicates
that the aerodynamic noise generation is not being caused by the valve itself but rather by the jet leaving the downstream duct, or by only the flow detachment of the valve’s trailing end. As shown in Figure 26 the relation between incoming flow velocity and radiated SPL, starts to be significantly different starting from valve angle smaller than 30°. Although the slopes of the smaller valve positions are slightly different, it can be concluded (due to the similarity between the different curves) that, starting from this angle, the aerodynamic noise generating mechanisms are very similar.

![Figure 25: SPL (dB) for different valve positions with incoming flow rate equal to (from top to bottom) 150m³/h, 200m³/h, 300m³/h, and 400m³/h.](image-url)
Figure 26: OASPL (dB) for different flow rates (logarithmic scaling) and valve positions.

Figure 27 shows the SPL for different valve angles when the pressure drop over the butterfly valve is kept (approximately) constant. Similar as before, discrepancies between the different curves are caused by the fact that the real pressure drop slightly differs from the values mentioned in Figure 27. It can be noticed that the shape of the radiated sound pressure spectrum is almost identical for the different valve positions when the pressure drop stays the same. Only a global shift downwards of the SPL is caused for smaller valve angles due to the smaller velocity of the incoming flow field.

This indicates that the actual aerodynamic noise generating mechanisms are generated by the same phenomena which cause the aerodynamic pressure drop over the butterfly valve, i.e. the impact and detachment of the turbulent flow over the butterfly valve’s surface, which is typical for a dipole-type of aerodynamic noise generation. This was verified by applying a scaling of the obtained SPL with the incoming Mach number. This analysis showed that, for valve angle smaller than 35° a relation between the SPL and the incoming Mach number, proportional to the 4th power of the velocity below the cut-off frequency of the high-order mode and to the 6th power at higher frequencies, indicating indeed a dipole character of the aerodynamic noise generating mechanisms. For large valve angles, going from 90° up to 45° the SPL scales between the 6th (low frequencies) and 8th power of the incoming Mach number, indicating quadrupole-type of noise generation caused by both the turbulent flow detachment in the wake of the valve and the noise generated by the free-jet expansion at the downstream end. For valve angles of 35° and 40° degrees, this scaling largely depends on the actual value of the incoming flow velocity, and is, at low velocities, similar to the open valve positions, while for large incoming flow speeds the scaling tends more towards a dipole-type of radiation.
Figure 27: SPL (dB) for different valve positions with pressure drop equal to (from top to bottom) 200mbar, 300mbar, 400mbar and 500mbar.

Based on both the (steady) aerodynamic characterization of the butterfly valve and the preliminary acoustic survey it can be concluded, that:

- The parallel Root’s blower test rig limits the maximum pressure drop over the butterfly valve to approximately 500mbar, corresponding to an incoming Mach number between approximately 0.05 and 0.025 for valve angles between 30° and 20° degrees.

- In this range of valve positions, the aerodynamics characteristics of the butterfly valve significantly differ from those obtained for open valve positions (i.e large valve angle). Furthermore it is shown that the incoming flow rate does not change significantly when the pressure drop is being increased further.
The aerodynamic noise generating mechanisms of the butterfly valve are caused by the impact of turbulent structures on the valve’s body, thus causing a dipole-type of noise generation. These phenomena are found to correspond with the total pressure drop over the valve and become dominant for valve angles smaller than 35°.

As such, it is chosen to perform the detailed measurements on the butterfly valve case with a valve angle of 30° and a pressure drop of 500mbar, corresponding to an incoming Mach number equal to 0.064 and estimated maximum Mach number in the cross section of the valve equal to 0.476. It is shown that the aerodynamic characteristics of this flow configuration are still (within the test rig’s limitation) representative for high-pressure ECS-system.
5 Theory

5.1 Active multiport characterization

Considering an arbitrary flow duct element, connected upstream and downstream to a straight duct, as shown in Figure 28, the acoustics characteristics, i.e. both the scatter properties and the active noise generation may be described as a linear time-invariant system by following equation:

\[ p_+ (\omega) = S^s p_- (\omega) + p_+^a (\omega) \]

Figure 28: Sketch of a duct with acoustic load and multi-port source. The source vector \( p_+^a \) contains the acoustic modes evoked by the source. The sound waves created by the acoustic load are reflected and transmitted at the multi-port. The indexes + and - denote the direction of the propagation.

where \( S^s \) denotes the scattering matrix related to the multi-port source, \( \omega \) is the angular frequency and \( p_- \) and \( p_+ \) are vectors of the directed wave modes. If an external, uncorrelated sound source is dominating the acoustic field, the source vector \( p_+^a \) vanishes in system of equations and the scattering matrix can be determined experimentally or numerically (Lavrentiev et al 1996). The approach to which is used to decompose the sound field from measurement data is developed by KTH and has been extensively validated (Åbom et al 1995). As \( p_- \) and \( p_+ \) include the spectra of all acoustic modes of the \( N \)-port in both propagating directions, the spatial sampling of the sound field must be performed by measuring the acoustic pressure for at least \( 4N \) spatial positions. Furthermore, to compute the scattering matrix, a set of \( 2N \) independent sound fields has to be ascertained.

The acoustic pressure at an arbitrary spatial position \( a_n \) in a duct with uniformly distributed flow can be written as

\[ p^{a_n} (\omega) = \Psi^{a_n} p_+ (\omega) + \Psi^{a_n} p_- (\omega) \]

where \( \Psi^{a_n} \) is a row vector containing the mode shapes at a certain spatial position. These functions are well known, e.g. for circumferential and rectangular ducts (Morse et al 1986).

Knowing the wave numbers for each mode that is propagating down and up-stream, a transfer matrix between two axial positions \( z_1 \) and \( z_2 \) can be written as:

\[ \left( T^{a_1, a_2}_{\pm} \right)_{mn} = \exp \left( \mp ik_{mn} (z_2 - z_1) \right) \delta_{mn} \]
where $\delta$ is the Kronecker's delta and $k_{mn\pm}$ is the wave number for the mode $mn$ (Abom 1989). This equation can be rearranged to
\[ T^{a_1,a_2} = T^{z_2-z_1} \]
with the exponent referring to the matrix elements. This leads to
\[ p^{a_n}(\omega) = \Psi^{a_1} T^{z_n-z_1} p_+ (\omega) + \Psi^{a_1} T^{z_n-z_1} p_- (\omega) \]
or written in matrix-vector notation
\[ p^{a_n}(\omega) = [\Psi^{a_1} T^{z_n-z_1} \quad \Psi^{a_1} T^{z_n-z_1}] [p_+ (\omega) \quad p_- (\omega)] \]
For a sound field measured at an arbitrary number of different spatial positions, a set of equations can be rewritten in matrix notation
\[ p(\omega) = M(\omega) p_\pm (\omega) \]
where $p(\omega)$ is a row vector containing the spectra of all spatial measurement points and, for a set of $n$ vector positions, $M(\omega)$ is the $[n \times 2N]$ transformed modal matrix.

Since this equation only contains the $2N \times 2N$ scattering matrix component as unknowns, while each measurement provides 1 relation between $p_+ (\omega)$ and $p_- (\omega)$, a minimum of $2N$ independent measurements are required. An over-determination of this system of equations, however, improves the accuracy of the final results, since the influence of measurements with a poor signal-to-noise ratio is less pronounced or can be discarded; and non-acoustic (aerodynamic) pressure fluctuations not obeying this vector identity are filtered out of the solution due to a least-squares procedure.

The goal of the active multi-port characterization investigation is not only determining the (passive) flow-acoustic scattering properties but also to carry out a full multi-modal source characterization. In this framework, only measurable values and constants shall be used, in particular the transformed modal matrix $M(\omega)$, the scattering matrix $S^s$, the termination reflection matrix $R$, and the vector of measured pressures $p(\omega)$.

Consider an active multi-port with two outlets $a$ and $b$ and the number of propagating modes equal to $N$, in this case the active source vector $p_a^s(\omega)$ is formulated as:
\[ p_a^s(\omega) = p_+ (\omega) - S^s p_- (\omega) \]
or in matrix formulation:
\[ \begin{bmatrix} p^s_a \\ p^s_b \end{bmatrix} = \begin{bmatrix} p^a_+ \\ p^a_- \end{bmatrix} - S^s \begin{bmatrix} p^b_+ \\ p^b_- \end{bmatrix} \]
With the reflection coefficient

\[ R_i = p_i - p_i^{-1} \]

\[ R = \begin{bmatrix} R_a & 0 \\ 0 & R_b \end{bmatrix} \]

This equation can be rearranged to

\[ p_\pm^s(\omega) = \left[ E - S^s R \right] p_\pm(\omega) \]

where \( E \) denotes the unit matrix. In order to determine \( p_\pm(\omega) \), similar as for the scatter matrix determination, a modal decomposition is applied:

\[
\begin{bmatrix} p_a \\ p_b \end{bmatrix} = \begin{bmatrix} M_{+a} & 0 \\ 0 & M_{+b} \end{bmatrix} \begin{bmatrix} p_{a+} \\ p_{b+} \end{bmatrix} + \begin{bmatrix} M_{-a} & 0 \\ 0 & M_{-b} \end{bmatrix} \begin{bmatrix} p_{a-} \\ p_{b-} \end{bmatrix}
\]

Or:

\[ p(\omega) = M(\omega) p_\pm(\omega) \]

In combination with the definition of the reflection matrix \( R \), this can also be formulated as:

\[ p = \left( M_+ + M_- R \right) p_+ = \mathcal{C} p_+ \]

This equation constitutes a reduction in the order of equations since only \( p_+ \) modes are calculated. Hence, an over-determination of microphones is archived, which can be used to suppress (non-acoustic) aerodynamic pressure fluctuations which are present in the microphone signals. Therefore, the microphones are paired in two groups and the source pressure is calculated, respectively,

\[ p_{+1}^s = \left[ E - S^s R \right] c_1^{-1} p_1 = \bar{\mathcal{C}}_1 p_1 \]

\[ p_{+2}^s = \left[ E - S^s R \right] c_2^{-1} p_2 = \bar{\mathcal{C}}_2 p_2 \]

The cross-spectral source matrix can then be defined as:

\[ G_s = \bar{\mathcal{C}}_1 \left( p_1 p_2^c \right) \bar{\mathcal{C}}_2 = \bar{\mathcal{C}}_1 G_{12} \bar{\mathcal{C}}_2 \]

where \( G_s \) contains the cross-spectra of the modes emitted by the source and \( G_{12} \) contains the cross spectra of all microphones.
5.2 Aero-acoustic guidelines

5.2.1 Context – State of the Art
Fan noise prediction is made challenging because of both the complicated incriminated flows and a true three-dimensional geometry. Facing cost and feasibility issues of numerical approaches, analytical techniques can be thought of as an alternative, at least for engineering purpose of preliminary design. But analytical formulations can only provide useful closed-form solutions at the price of drastic simplifications. Therefore they have to be validated, for instance by comparison with dedicated experiments or reference numerical solutions, before being applied to problems of industrial interest. The simplifications are a key step when proposing an analytical formulation. They can be done on the flow features but need being accepted first on the geometry.

In the case of a ducted turbo-machine, typically a rotor-stator stage, the blade and vane design is the major difficulty to deal with. A possible methodology is guided by the acoustic analogy, which states that the loadings on the blades and the vanes are acting as distributions of equivalent dipole sources. A model of sound generation and propagation requires two steps with this view. First the (essentially unsteady) loads must be described or modelled from some knowledge of the relative velocity disturbances they originate from; this can be achieved by some simplified theory of unsteady aerodynamics and is often the most challenging issue. Secondly the sound must be calculated from the sources, based on the general background of linear acoustics. This second step can be declined in open space or in a duct depending on the chosen Green’s function.

Solving the first step analytically requires simplifying the blade and vane geometry and leads to more or less academic generic problems. Generalized Sears’ theory of the response of a rectangular flat-plate airfoil to incident sinusoidal gusts, typically addressed by the so-called Amiet-Schwarzschild’s technique, is an example of such a generic problem. In its original form it does not account for 3D design parameters such as lean, sweep, twist and camber. Yet it can be applied to annular cuts of a blade or vane row of different radii within the scope of a strip-theory approach, which allows including step-by-step varying conditions in the span-wise direction. Furthermore the theory only addresses an isolated-airfoil response, quite questionable when applied to a cascade of airfoils. This means that the presence of adjacent blades is ignored when deriving the unsteady aerodynamic response of a rotor blade to oncoming disturbances, in the first step of the method. Because the equivalent sources of the analogy radiate without mutual influence the same simplification is implicitly accepted for the second step: sound radiation does not account for multiple diffractions by other blades. This is why the method is only reliable for rotors (stators) with non-overlapping blades (vanes) or of low solidity. Amiet’s isolated-airfoil theory is versatile: it describes the sources of the sound and therefore is useable for both free-field and in-duct radiation contexts. Because it artificially considers each vane of a stator individually it has also the advantage of easily accounting for stator non-homogeneity, for instance when some of the vanes are differently designed for structural reasons. A single-airfoil response associated with a strip-theory approach also simply accounts for twist and stagger angle effects, by considering different
inclination angles at different radii. Recent extensions of Amiet-Schwarzschild’s technique made solutions available also for a flat-plate airfoil of arbitrary trapezoidal shape, which allows including sweep and lean in the analysis. This has been shown to provide physically consistent estimates of real blade design effects in the case of modern counter-rotating open-rotor architectures in aircraft propulsion (Carazo et al 2011).

In the case of ducted turbomachines the definition of a best-suited analytical model must be reconsidered with regards to the high solidity of rotors and stators associated with the relatively large numbers of blades and vanes. In particular in a single-stage rotor-stator architecture the outlet guide vanes of the stator have to redistribute the exit flow along the axis; as such their chord is not far from parallel to the axis, with quite a significant camber at leading edge. The extended overlap makes cascade effects presumably very important. The need for cascade response functions to replace simple Amiet’s theory has been recognized and motivated the development of *ad hoc* mathematical formulations in the literature (see for instance Glegg’s model readdressed by Posson *et al* 2010). These formulations are again based on a strip-theory approach in which each annulus is unwrapped and described as a linear cascade. The blades or vanes are assumed staggered flat plates. The price to pay is very important in terms of computational time with regards to general performances of other analytical solutions. For broadband-noise calculations this may become an issue in the same way as for numerical strategies. Furthermore the blades are assumed parallel to each other in any strip, which generates spurious resonances that pollute the solution, and the overlap must be non-zero. Dealing with that point, Amiet’s method *de facto* releases the necessity of parallel blades or vanes.

### 5.2.2 Dimensional Analysis of Rotor-Stator Stages

Aforementioned points illustrate that different assumptions can be made, more or less justified, leading to different generic problems and different closed-form solutions, starting from the same turbomachine. The isolated-airfoil response function does not need assuming parallel blades or vanes but it ignores cascade effects. Conversely the linear cascade model suffers from spurious resonances and supposes relatively high solidity. It is understood that the best choice could result from a trade-off between realism and mathematical tractability, depending on the overall design of the investigated system. Therefore a preliminary classification of turbomachine architectures in terms of non-dimensional parameters appears as a necessary step.

A rotor-stator stage is first described by the inner and outer (hub and tip) radii, say $R_1$ and $R_2$, or equivalently the outer radius $R_2$ and the hub-to-tip ratio $\sigma=R_1/R_2$. Large values of $\sigma$, close to 1, define the narrow annulus for which a class of simplifications is acceptable. Then the numbers of blades and vanes, $B$ and $V$ respectively, define the angle between adjacent blades/vanes, say $2\pi/V$ or $2\pi/B$. The unwrapped linear-cascade approximation makes sense only if $B$, $V$ and $\sigma$ are large enough and becomes exact in the limit of infinite $B$ or $V$. The next parameter is the chord length $c$, from which the aspect ratio is defined as $(R_2-R_1)/c$ since $(R_2-R_1)$ is the span.
For a stator of small or moderate aspect ratio the vanes are generally untwisted; if furthermore both $\sigma$ and $V$ are large the behaviour of the stator can be described by a single annulus with an unwrapped linear cascade model. In contrast to stator vanes the blades of a rotor are significantly twisted (except in the case of the ECS fan, see Fig.1), therefore the parameters of the equivalent linear cascade are very different from the hub to the tip; this requires multiple annuli in a strip-theory approach.

Another important parameter is the rotor-stator gap $d$, defined as the distance between the rotor trailing edge and the stator leading edge. It has to be quite small in order that the aerodynamic benefit of the stator is maximized, but this is at the detriment of the acoustic signature, which expectedly decreases with increasing gap. With regards to wake-interaction noise the relevant parameter is $dl/(c_R \cos \gamma_R)$, where $\gamma_R$ is an averaged blade angle with respect to the duct cross-section and $c_R$ the characteristic blade chord. The smaller the parameter the higher the interaction noise, for both its tonal and broadband components.

Apart from geometrical considerations, the system is characterized by velocity and frequency parameters. The rotational speed $\Omega$ produces the rotational frequency $\Omega/(2\pi)$ and the frequencies of the tonal noise at harmonics of the blade-passing frequency (BPF), $\omega_m = mB \Omega/(2\pi)$. The main features of the sound propagation will therefore be imposed by the ability of complex oblique waves generated by the interaction mechanisms to be transmitted or not. The leading parameter is the Helmholtz number $k_R^2 = \omega R^2/c_0$. Below $k_R^2 = 1.84$ in a simple hard-walled duct only the plane-wave mode propagates, whereas multiple spinning modes are transmitted for substantially higher frequencies. The number of possibly transmitted modes at one of the BPF tones increases with the order $m$ of $\omega_m$ but they will be excited or not depending on the blade and vane counts. In contrast all possible modes are excited by random noise sources. The mode of azimuthal order $n$ has the azimuthal phase speed $M_n$ at the angular frequency $\omega$. It is transmitted only if $kR_2 > \beta_a K_{nj}$, where $K_{nj}$ is a tabulated value function of the hub-to-tip ratio $\sigma$ and $\beta_a$ the parameter defined below.

The tangential Mach number at blade tip, $M_t = \Omega R 2/c_0$ is another key parameter. It must be considered together with the axial mean-flow speed Mach number $M_a = W_0/c_0$. As long as $M_t < \beta_a = (1-M_a^2)^{1/2}$ the flow remains essentially subsonic. A sufficient (but not necessary) condition for a spinning mode (non-zero $n$) to be cut-off is that its tangential phase-speed Mach number at tip is subsonic in the sense that $\omega R 2/(n c_0) < \beta_a$.

The ECS fan corresponds to the parameters listed below (see Figure 29-left):

- chord lengths $c_R = 34.4$ mm, $c_S = 46$ mm
- $B=15$, $V=10$
- rotor-stator gap $d = 3.5$ mm
- $R_1 = 52$ mm
- $R_2 = 75$ mm (duct), 84 mm (fan)
- \( \sigma = 0.64 \)
- \( \Omega/(2\pi) = 190 \text{ Hz} \)
- \( \text{BPF} = 2850 \text{ Hz} \)
- \( M_t \sim 0.26, M_a \sim 0.1, \beta_a \sim 0.99 \)

The rotor and the stator have similar, quite high solidities, with negligible overlap for the rotor and nearly total overlap for the stator. Amiet’s (or another isolated-airfoil) approach is relevant for the rotor but questionable for the stator because of the cascade effect. The moderate number of vanes makes them far from parallel plates. The short rotor-stator distance reinforces periodic interaction noise mechanisms. The blades and vanes have negligible twist, lean and sweep. They are good candidates for simplified analytical models.

![Diagram](image)

Figure 29: ECS test fan side-view (left) and velocity triangles (right).

In Figure 29 the stagger angle of the stator is defined by the inclination of the mean camber line at leading-edge, which is the most suited convention for analytical tractability. Indeed linearized methods such as Amiet’s theory assume zero loading conditions on the flat plates mimicking the vanes, then zero angle of attack. Assuming that the components of the velocity triangle are aligned with the chord lengths (ignoring deviation angles for simplicity), simple estimates of the velocities can be found, at least corresponding expectedly to the best design operating point, according to

\[
\frac{W_R}{\Omega r} = \frac{\sin \gamma_S}{\sin (\gamma_S + \gamma_R)} \quad \text{with} \quad W_0 = W_R \sin \gamma_R
\]

Close to the tip radius, say 80 mm, this leads to \( W_R \approx 84 \text{ m/s} \) and \( W_0 \approx 54 \text{ m/s} \).
5.2.3 Rotor Wake Modelling

Wake-interaction noise modelling requires the definition of the periodic (averaged) part and of the random part of the velocity field in the wakes, understood as variations around the mean-flow velocity triangle (Figure 29-right). For convenience wake turbulence can be interpreted as homogeneous and isotropic turbulence modulated by a periodic envelope. This possibly introduces two periodic functions, one for each noise contribution. The same approach can be used to describe the mean velocity deficit in the wakes and the periodic turbulence envelope; only the former is detailed here. In absence of accurate flow simulations or dedicated experiments, an empirical wake model is acceptable, especially for rough estimates. Such a model describes the expansion of the wake and its attenuation with the distance from the trailing edge along the wake axis. It has to be superimposed on a previously determined rotor-exit velocity triangle. A Gaussian model is often accepted, according to which the time-periodic velocity deficit as seen from a stator vane reads

\[ w(t) = w_0 \sum_{n=-\infty}^{\infty} e^{-\xi(t-nT)^2/\tau^2} = \sum_{m=-\infty}^{\infty} W_{mb} e^{-imB\Omega t}, \]

and its Fourier series (to be cleaned of the non-zero average) is derived as

\[ W_{mb} = \frac{B\Omega w_0 \tau}{2\pi} \sqrt{\frac{\pi}{\xi}} e^{-\frac{\pi(4B\Omega)^2}{4\xi}}, \]  

(5.1)

with \( \xi=0.693 \), \( \tau \) being the half-wake passage time and \( T=2\pi/(B\Omega) \) the wake passing period. \( \tau \) corresponds to some wake half-width noted \( b \). Empirical correlations have been proposed in the literature for the parameters \( w_0 \) and \( \tau \) (or \( b \)) as functions of the streamwise distance \( s \) to the trailing edge along the wake centerline, with the concern that they address specific classes of cascades or true rotors and could be questioned when transposed to other classes. Furthermore the wake width is considered normal to the centerline. Anyway two wake regimes are recognized. In the near wake the velocity deficit \( w_0 \) has the order of magnitude of the external mean-flow speed and decreases quite fast with downstream distance; the wake width is relatively small. In the far wake, \( w_0 \) becomes much smaller than the external flow speed but only decreases slowly; the wake also goes thicker. The near wake behavior extends often up to a quarter chord length from the trailing edge. In view of Figure 29 the leading edges of the stator vanes precisely enter this range, which makes the interaction take place in the near-wake region. This corresponds to a high level of interaction noise, for both tonal and broadband components.

The Gaussian wake profile is certainly abusive in the near wake for a loaded blade; indeed the mean loading makes the velocity deficit asymmetric with respect to the wake centerline. Furthermore the wake is cut obliquely, which accentuates the asymmetry. Yet if the interaction takes place in the far wake the most determinant parameters are the local wake depth and width, at least for the first 4 BPF harmonics; as a result the model remains reasonably accurate for a first assessment. In the present case the asymmetry can be approximated as follows. Assume that the wake half-width is \( b_0 \) at the point cut by the stator leading-edges but normal to the wake axis, and introduce the local angle \( \psi \) such that
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\[ \tan \psi = \frac{ds}{ds} \]. Assume also that the wake is aligned with the blade chord at angle \( \alpha_2 \). The values of the half widths \( b_1 \) and \( b_2 \) for the steepest and the widest side, respectively, are derived as

\[
\frac{b_1}{b_0} = \left[ \cos \alpha_2 + \tan \psi \sin \alpha_2 \right]^{-1}, \quad \frac{b_2}{b_0} = \left[ \cos \alpha_2 - \tan \psi \sin \alpha_2 \right]^{-1}
\]

Based on two different parameters \( \tau_j = b_j / (\Omega r) \) for the two sides of a Gaussian wake profile, a more general formulation has been derived analytically by Roger (1994). The same description can be applied to the envelope of the turbulence in the wakes, provided that the parameters of the envelope are properly tuned.

5.2.4 Airfoil Broadband Noise Models

5.2.4.1 Preliminary Remark

In principle, accuracy motivations require that broadband noise sources in a duct are addressed with the general formalism of guided waves, accounting for cut-on or cut-off properties of the acoustic modes. However random source mechanisms are known to excite all possible propagation modes in the duct, in contrast to tonal noise sources. At high frequencies, the modal density is such that the acoustic energy delivered by the sources is distributed on a large number of cut-on modes; ignoring cut-off properties in that sense becomes justified. For a rough estimate, it can be reasonably assumed that the total acoustic power in the duct, summing upstream and downstream contributions, is the same as what the sources would produce in free field. This allows applying simple free-field models of airfoil noise.

Furthermore interference properties between the contributions of different blades or vanes of a rotor or a stator are determinant for the structuration of specific modes of tonal noise, but in contrast they are less crucial for random sources. This leads to the second simplification that the response of blades or vanes is described by isolated-airfoil theories, ignoring cascade effects.

The present analysis is only dedicated to order of magnitudes that can be expected from various sound generating mechanisms, in order to rank them and to focus the subsequent efforts of accurate in-duct modelling to the dominant one. This is why the isolated-airfoil, free-field approach is considered relevant at that stage. More advanced models of broadband noise including in-duct propagation and cascade response effects, are developed by the contributor in other work packages of the project.

5.2.4.2 Turbulence-Impingement Noise Modelling

The power spectral density (PSD) of the far-field sound pressure \( S_{pp}(x, \omega) \) is related to the statistics of the upstream turbulent velocity field via spectral turbulent models used as input data. The general approach has been described by Amiet (1975). A review of more recent works is proposed in Roger (2013). For large-span airfoils such that the aspect ratio \( L/c \to \infty \) an approximate expression is obtained from the general formulation, as

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\[
S_{pp}(x, \omega) = \left( \frac{\rho_0 k c x_3}{2 S_0^2} \right)^2 \pi U_0 \frac{L}{2} \Phi_{ww} \left( k, \frac{k x_2}{S_0} \right) \left| S \left( x, k, \frac{k x_2}{S_0} \right) \right|^2
\]  

(5.2)

where \( \Phi_{ww} \) is the two-dimensional wave-number spectrum of the turbulent velocity component normal to the airfoil. Here \( k_1 = \omega / U_0 \), and \( L, c \) stand for the span and chord lengths, respectively. This simplified formulation is well suited for sound estimates at the early design stage of a fan or when detailed information about the design or the flow is not available.

\( x_3, x_1 \) and \( x_2 \) are observer coordinates with origin at the centre of the airfoil, normal to the airfoil plane and along the airfoil chord and span, respectively. \( U_0 \) is the relative/oncoming flow speed, assumed along the chord. \( S_0 = [x_1^2 + \beta^2 (x_2^2 + x_3^2)]^{1/2} \) is a corrected distance accounting for sound convection by the surrounding flow. It is close to the geometrical distance \( r \) if \( M_0 \) is small. \( L \) is the span, \( k = \omega / c_0 \) is the acoustic wavenumber, and \( \rho_0 \) the air density.

The simplified expression of the aeroacoustic transfer function is the sum of the main scattering term and the trailing-edge back-scattering term which read, respectively,

\[
\mathcal{Z}_1(x, \omega U_0, k_2) = -\frac{1}{\pi} \sqrt{2 \left( k_1^2 + \beta^2 \mu \right) \Theta_4} \left[ 2 E(2\Theta_4) e^{-i\Theta_2} \right]
\]

(5.3)

\[
\mathcal{Z}_2(x, \omega U_0, k_2) \approx \frac{e^{-i\Theta_2}}{\pi \Theta_4 \sqrt{2\pi \left( k_1^2 + \beta^2 \mu \right)}} \left[ (1 - e^{2i\Theta_2}) - (1 + i) \left[ E(4\mu) \sqrt{2 \left( k_1^2 + \beta^2 \mu \right)} e^{2i\Theta_2} E\left(2 \left[ \mu (1 + x_1 / S_0) \right]\right) \right] \right]
\]

(5.4)

with \( \Theta_4 = \mu \left( 1 - \frac{x_1}{S_0} \right), \Theta_2 = \mu (M_0 - \frac{x_1}{S_0}) - \frac{\pi}{4}, \mu = \frac{k_1^2 M_0}{\beta^2} \) and \( E(\xi) = \int_{0}^{\xi} \frac{e^u}{\sqrt{2\pi u}} \, du \).

For low-frequency applications, Amiet’s theory may fail and original Sears’ theory corrected for compressibility effects can be used instead. The corresponding expression of the transfer function for parallel gusts, more expectedly involved because three-dimensional effects are less pronounced, is

\[
\mathcal{Z}(x, k, 0) \approx \left| S \left( k, \beta^2 \right) \right| \beta
\]

where \( S \) stands for Sears’ function. The threshold between both regimes is estimated around

\[
\mu = \frac{k_1^2 M_0}{\beta^2} = \frac{k c}{2 \beta^2} \approx \frac{1}{4}.
\]

So Sears’ theory must be used below the (relative) frequency 800Hz for the rotor blades and below the (absolute) frequency 600Hz for the stator vanes. The latter are source frequencies, not essentially sound frequencies.

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The minimum/simplest information for $\Phi_{ww}$ is provided by Liepmann’s or von Kármán’s models of homogeneous and isotropic turbulence, for which analytical expressions are available (Hinze 1975). Such turbulence is described by two parameters, the root mean square value of the velocity fluctuations $u_{rms}$ and some integral length scale $\Lambda$. For an assessment in the mid-span plane, the frequency spectrum $S_{ww}$ and the correlation length $\ell_y$ are related to $\Phi_{ww}$ by

$$S_{ww}(\omega) = \frac{u_{rms}^2 \Lambda}{6\pi U_0} \frac{3 + 8(k_1 / k_2)^2}{[1 + (k_1 / k_2)^2]^{1/6}}$$

$$\ell_y(\omega) = \frac{8\Lambda}{3} \left( \frac{\Gamma(1/3)}{\Gamma(5/6)} \right)^2 \frac{(k_1 / k_2)^2 [1 + (k_1 / k_2)^2]}{[3 + 8(k_1 / k_2)^2]}$$

with $k_x = \frac{\sqrt[5]{\pi}}{\Lambda} \Gamma(5/6) / \Gamma(1/3)$, and

$$S_{ww}(\omega) = \frac{u_{rms}^2 \Lambda}{2\pi U_0} \left( 1 + 3k_1^2 \Lambda^2 \right)$$

$$\ell_y(\omega) = \frac{3\pi \Lambda}{2\left[1 + k_1^2 \Lambda^2\right]^{1/2}} \frac{k_1^2 \Lambda^2}{\left[1 + 3k_1^2 \Lambda^2\right]}$$

Two functions $F$ and $G$ can be introduced such that

$$S_{ww}(\omega) = \frac{u_{rms}^2 \Lambda}{2\pi U_0} F(k_1 \Lambda)$$

$$\ell_y(\omega) = \alpha \Lambda G(k_1 \Lambda)$$

where $\alpha$ is some constant. The function $F$ goes to 1 at vanishing frequencies and asymptotically decreases like $(k_1 \Lambda)^{-n}$ at high frequencies, with $n=2$ for Liepmann’s spectrum and $n=5/3$ for von Karman’s spectrum. $G$ increases like $(k_1 \Lambda)^2$ at very low frequencies and decreases asymptotically like $(k_1 \Lambda)^{-1}$ at very high frequencies. This provides indicative low-frequency and high-frequency scaling laws as, respectively

$$\frac{c_0 S_0^2 S_{pp}}{\tau^2 c \rho_0 U_0^2} \propto \left( \frac{x_1}{S_0} \right)^2 \frac{U_0}{c_0} (k_1 \Lambda)^4$$

$$\frac{c_0 S_0^2 S_{pp}}{\tau^2 c \rho_0 U_0^2} \propto \left( \frac{x_3}{S_0} \right)^2 \frac{\Lambda}{c} (k_1 \Lambda)^{-3}$$

in which $\tau = u_{rms} / U_0$ is the turbulence rate.

5.2.4.3 Thickness Effect on TI Noise

A series of experiments discussed by Roger (2013) show that the response of an airfoil to incident turbulence is reduced by the thickness of the, assumed rounded, leading-edge area, with respect to what the thin-airfoil or the zero-thickness assumption would produce. Based on a reasonable collapse of some dimensionless data, the amount of reduction in decibels was found proportional to both thickness and frequency when going back to the dimensional data, according to the simple formula

$$\Delta_{dB} \approx 0.137 \frac{e}{\Lambda} \frac{f}{U_0}$$

(5.5-a)
in which the frequency is in Hz and the flow speed in m/s. Such a reduction is expected for the vanes of the ECS fan because of the leading-edge thickness. Its quantification cannot be accurate because of scatter in the initial data, and probably because main parameters such as turbulence homogeneity, integral length scale and so on play a determinant role and differ in various configurations. Thickness-induced reduction has been quantified for turbulence-impingement noise but it can be guessed that it holds also for deterministic excitations. In the study it will be applied systematically to all analytical noise predictions, both broadband and tonal, using Amiet’s and similar models. Equation (5.5-a) only holds for moderate frequencies and still remained to be confirmed. This is why an indication of the thickness effect has been simply provided by making new tests with the same NACA-0012 airfoil in the right and reversed (trailing edge in place of the leading edge and *vice versa*) positions, in the same grid-turbulence field. The test is reported in Figure 30. It appears that the proportionality of Eq.(5.5-a) is fairly well confirmed but with a substantially lower constant, 0.08 instead of 0.137.

It is worth noting that the reversed airfoil cannot behave in a normal way; separation at the rounded back-edge probably generates additional sound sources, which can be responsible for the multiple small humps on the top of the blue curves in Figure 30-a,b,c and for the discrepancies between the modelled and predicted differences at low frequencies in Figure 30-d.

The response reduction exceeds 10 dB around 3kHz in the present case, values at higher frequencies being questionable or inaccurate. This is why another model assumption has been tested on the measured reductions, based on a saturation at $\Delta_0=15$dB. The associated empirical expression reads

$$\Delta_{dB} \approx \Delta_0 \left[ 1 - \exp \left( -0.008 \frac{e}{\Lambda} \frac{f}{U_0} \right) \right]$$

(5.5-b)

and is plotted as continuous lines in Figure 30-d. It is as convincing as the aforementioned linear regression. This makes a reliable thickness correction in more general cases somewhat challenging for the moment. Yet a correction is needed. When applying Amiet’s theory to predict TI noise, the sound is clearly overestimated for a thick airfoil at leading edge, but it is very accurate for an airfoil with thin or sharp leading-edge.
5.2.4.4 Trailing-Edge Noise Modelling

The power spectral density of the far-field pressure due to trailing-edge noise, in the same limit of the long-span approximation, reads

\[
S_{pp}(x, \omega) = \left( \frac{k c x_1}{2 \pi S_0} \right)^2 L \Phi_{pp}(\omega) \left( k x_2 \right) \left( \frac{k x_2}{U S_0} \right)^2 \left( x_1, \frac{\omega}{U}, \frac{k x_2}{S_0} \right)
\]  (5.6)

It is written here in non-dimensional variables, introducing the wavenumber spectrum of the wall-pressure fluctuations

\[
\Pi_{0}(k_1^*, k_2^*) = \frac{1}{\pi} \Phi_{pp}(\omega) \ell_{k_2}(k_2^*, \omega)
\]
where $\Phi_{pp}$ is the local wall-pressure PSD induced close to the trailing edge by the incident turbulence only and $\ell_{\gamma}(k_z^*,\omega)$ is the correlation length defined from the coherence function between two spanwise locations $\eta$ apart from each other as

$$\ell_{\gamma}(k_z^*,\omega) = \int_0^{\infty} \sqrt{\gamma^2(\eta,\omega)} \cos(k_z \eta) \, d\eta$$

$N$ is another aeroacoustic transfer function, for which an expression was initially proposed by Amiet (1976-b) (main term and parallel gusts only) and recently extended by Roger & Moreau (2005) to account for leading-edge back-scattering and three-dimensionality. The simplest reduction of the model is enough for a preliminary estimate of compared contributions from various broadband-noise mechanisms, facing the lack of precise input data. $N \approx N_1$ with

$$N_1 = \frac{e^{-2ic}}{iC} \left\{ (1-i)e^{2ic} \sqrt{\frac{B}{B-C}} E[2(B-C)] - (1-i)E[2B]+1 \right\}$$

(5.7)

$$B = \kappa^* + k_z^* M_0 \ell(1-M_0) \quad C = \kappa^* \left( \frac{x_i}{S_0} - M_0 \right) \frac{k_z^* M_0}{\beta^2} \quad \kappa^* = \frac{\omega c}{2U_c}$$

$U_c$ is an averaged convection speed of the boundary layer disturbances.

The key point in the applications is to get relevant information on $\Phi_{pp}$ and $\ell_{\gamma}$. Trailing-edge noise is very sensitive to blade or vane design details and to loading conditions associated with the operating point of a fan, so that very different acoustic signatures are encountered. This makes analytical modelling a challenging task. According to Amiet’s formulation the needed parameters are the wall-pressure spectrum, as taken closely upstream of the trailing edge, and the associated spanwise correlation length and streamwise convection speed; furthermore the wall-pressure statistics must be close enough to homogeneity in the aft part of the considered blade or vane section. Typically model wall-pressure spectra can be built up based on the external variables of the boundary layers, or on the inner variables, keeping in mind that the former are of easier access than the latter. Most reported statistical models based on the outer boundary-layer variables for the scaling of the wall-pressure spectrum are in the form

$$\Psi_{pp}(\omega) = \frac{\Phi_{pp}(\omega)}{\rho_0^2 \delta^3 U_0^3} \quad \Psi_{pp}(f) = \frac{\Phi_{pp}(f)}{\pi \rho_0^2 \delta_1^3 U_0^3}, \quad f > 0$$

introducing the displacement thickness of the boundary layer $\delta_1$. For comparison with experimental data the second expression in terms of positive frequencies is more relevant than the theoretical one in terms of bilateral angular-frequency transform.

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Writing the analytical expression of the far-field pressure PSD in the reduced form

\[ \frac{S_{pp}^2 \phi_{pp} \ell_y}{L} = 2 \left( \frac{k_c}{4\pi} \right)^2 \left( \frac{x_3}{S_0} \right)^2 \left| \mathbb{F} \right|^2 \]

Few models are available for the span-wise correlation length \( \ell_y(\omega, k_z) \). According to Corcos’ assumption,

\[ \ell_y(\omega, k_z) = \frac{\omega \ell(bU_c)}{k_y^2 + [\omega \ell(bU_c)]^2} \]

Here \( b \) is a constant initially found for boundary layers over flat plates with no pressure gradients and taking different values depending on the pressure gradient, and \( U_c \), the convection speed beneath the boundary layer.

Measurements of low-Reynolds number and low-Mach number airfoils or fans reported by various authors have been readdressed by Roger (2013), leading to the assumption of a log-normal law for the span-wise coherence as a function of frequency. The associated expression for the correlation length is

\[ \frac{\ell_y(\omega)}{\delta_{i1}} = \frac{a}{\sqrt{\sigma} \sqrt{2\pi St}} e^{-\left[ \ln(\text{St}) - \mu \right]^2 / (2\sigma^2)} \quad (5.8) \]

with \( a=1.9 \), \( St = 2\pi f \delta_{i1}/U_0 \), \( \mu=\ln 0.44 \) and \( \sigma=0.55 \). This empirical fit predicts strongly decreasing values at higher frequencies for which measurements are not feasible in practice; therefore the high-frequency limit could be equivalently modelled by Corcos’ expression, which provides a much slower decrease. This could be considered as a strong uncertainty in the high-frequency limit.

For a turbulent boundary layer over an airfoil the relative displacement thickness as a function of the chord-based Reynolds number can be inferred from Schlichting (1968), as

\[ \frac{\delta_{i1}}{c} = 0.0477 \text{ Re}_{c}^{-1/5} \]

This makes very small values expected, typically 0.6%, 0.5% and 0.4% for couples of parameters \( (c=2 \text{ cm, } U_0=30 \text{ m/s}) \), \( (c=2 \text{ cm, } U_0=90 \text{ m/s}) \) and \( (c=5 \text{ cm, } U_0=90 \text{ m/s}) \), respectively. The actual values for loaded airfoils are certainly much larger.

Values of the displacement boundary-layer thickness \( \delta_4 \) of about 2-3nm depending on the operating point for a blade chord of 100mm and an external flow speed of 30 m/s have been reported as typical of low-speed fans (Guédel et al 2011). The values strongly depend on the flow conditions and can be significantly larger at high loads. Without any precise information this can be used with some transposition to infer orders of magnitude for any subsonic fan
blade. Since $\delta_1$ is proportional to the chord-based Reynolds number to the power (-1/5), two configurations labelled (1) and (2) having the same flow regime obey the relationship

$$\frac{\delta_1^{(1)}}{\delta_1^{(2)}} = \frac{c_1}{c_2} \left( \frac{Re^{(1)}}{Re^{(2)}} \right)^{-1/5}$$

The transposition makes boundary-layer thicknesses of 0.7 to 1 mm expected in the case of the ECS fan rotor blades, and of 0.95 to 1.4 mm for the stator vanes. These very indicative values can be used to produce orders of magnitude of the trailing-edge noise (see section 5.2.4.6).

5.2.4.5 Asymptotic Behaviour and Competition between Mechanisms

Except at off-design conditions, broadband noise mainly arises from turbulence impingement (TI) and/or trailing-edge (TE) scattering. Relevant predictions have to cope with both or at least to make sure that one of them is negligible if only the other one is retained. Providing rough estimates of their compared levels is of major interest for low-noise fan design.

The results of Figure 30 are repeated in Figure 31. The airfoil noise is measured both in a clean flow with boundary-layer tripping, and in grid-generated turbulence.

Both spectra coincide beyond 3kHz when cleaned of their respective background-noise contaminations. This suggests that trailing-edge scattering is responsible for the sound radiated in this high-frequency range. Indeed pure turbulence-impingement noise cannot be isolated because boundary layers develop over the airfoil surface also in the presence of upstream turbulence. In contrast trailing-edge noise is the only contribution when the airfoil is tested in clean flow. This is why a subtraction procedure is applied in Figure 31 to estimate the true TI noise at high frequencies. Though not accurate because of statistical uncertainties, this estimate is very close to the corrected prediction based on Amiet’s theory and on the empirical correction, Eq.(5.5).

The turbulence rate is of 4.5% in the experiment, somewhat similar or slightly lower than typical wake turbulence. The integral length scale is $\frac{3}{4}$ of the vane thickness, which is also a relevant value. This is why the isolated-airfoil example is believed representative of what happens with the stator vanes of the ECS fan. For a first insight, the results can be made dimensionless and transposed to the stator vanes. So trailing-edge noise is expected to dominate the stator broadband noise above a Strouhal number of about 10, based on the vane

![Figure 31: Measured versus predicted TI noise levels. Measured TE noise in blue, for comparison. NACA-0012 airfoil, 28 m/s, chord length 10cm, background-noise corrected.](image)
chord and the local flow speed. Below this threshold turbulent wake-interaction noise dominates. It must be noted that for a thinner airfoil with the same inflow turbulence, the measured TI noise would be closer to Amiet’s thin-airfoil prediction without thickness correction; therefore TE noise would not be discernible because hidden by TI noise over the full frequency range.

In the present case the relative flow speed along the rotor blade chords close to the tip is around 95 m/s; the relative speed along the stator vane chord is about 54 m/s (estimates based on the simplified velocity triangle of Figure 29). The aforementioned Strouhal number of 10 roughly corresponds to 11 kHz for the stator vanes and 24 kHz for the rotor blades. This means that trailing-edge broadband noise needs being considered only for the stator and in the very high-frequency range. It can be neglected to a first instance. The evaluation holds for attached flows corresponding to operation close to optimal design conditions. Flow separation on the vanes or on the blades could regenerate additional self-noise below the limit Strouhal number of 10. But separation on the rotor also produces strong vortical patterns which generate sound when impinging on the stator vanes. Again this tends to make turbulence-impingement noise the dominant mechanism.

The asymptotic high-frequency trend for turbulence-impingement noise is imposed by the term $S_1$ in the limit $k c = 2 M_0 k_t^* \to \infty$, which leads to

$$
\frac{S_{pp}}{\Phi_{ww}} = \frac{\rho_0^2 U_0 LM_0}{\pi R^2_c} \frac{\cos^2(\theta_c/2)}{(1 + M_0 \cos \theta_c)^3}
$$

with $S_0 = R_c (1 + M_0 \cos \theta_c)$, expressing the results in emission coordinates with respect to the surrounding flow. In the limit of small Mach numbers $M_0 \to 0$, a cardioid directivity pattern is found. The sound goes to zero upstream and focuses downstream. For the low-frequency limit, Amiet’s theory must be replaced by Sears’ theory and the radiation integral approaches 1. The ratio becomes

$$
\frac{S_{pp}}{\Phi_{ww}} = \frac{\rho_0^2 \pi U_0 L}{8 \beta^2 R^2_c (kc)^2} \frac{\sin^2 \theta_c}{(1 + M_0 \cos \theta_c)^3}
$$

The radiation is that of a compact dipole in motion, zero in the plane of the airfoil and maximum in the normal direction.

The asymptotic analysis of trailing-edge noise only makes sense for the high-frequency limit and the main term $S_1$. Introducing the convection Mach number $M_c$ yields the result

$$
\frac{S_{pp}}{\Phi_{pp} \ell_y} = \frac{M_c L \sin^2(\theta_c/2)}{\pi^2 R^2_c (1 + M_0 \cos \theta_c)} \frac{1 - (M_0 - M_c)}{[1 + (M_0 - M_c) \cos \theta_c]^2}
$$

in which the last factor can be ignored for low-Mach number applications. Now no sound is radiated downstream and the maximum sound is radiated upstream. Trailing-edge noise radiates preferentially upstream and turbulence-interaction noise preferentially downstream.
Combining all arguments leads to the conclusion that the transmitted broadband noise at exhaust of the ECS fan is essentially wake-interaction noise and more generally turbulence-impingement noise.

5.2.4.6 **Blind Estimates of Broadband Noise Contributions**

A first blind estimate of the broadband noise contributions due to wake turbulence impingement on the stator vanes and to trailing-edge noise from both the rotor blades and the stator vanes has been attempted, based on the theoretical background of previous sections. This estimate is aimed at determining the dominant sources that will have to be modelled more accurately with advanced models in the next steps of the project. This is achieved as follows.

An isolated-airfoil model is first used to predict the far-field pressure of an isolated blade or vane radiated on a sphere surrounding the airfoil. In a second step integration of the acoustic intensity over the sphere is performed to infer the associated acoustic power. This power is finally multiplied by the number of vanes or blades in order to get the total acoustic signature of the stator or the rotor. In a second step the power level is transposed to get a sound pressure level that can be compared to an external microphone measurement. The results are just an indication of relative orders of magnitude. They are reported in Figure 32. Both trailing-edge noise contributions are estimated from the expressions of section 5.2.4.4. Two empirical models for the correlation length are compared (Figure 33). Because of a higher relative flow speed and a smaller chord length, rotor TE noise is higher than stator TE noise with a higher energy in the high-frequency range. All estimates remain at least 10dB below the measured spectrum. This suggests that trailing-edge noise in the usual sense is not the dominant broadband noise mechanism. Yet flow separation could be responsible for the generation of additional self-noise, not addressed here.

**Figure 32:** Blind estimates of blade and vane trailing-edge noise. Inputs from sections 5.2.4.2 and 5.2.4.4. Corcos’ model as dashed lines and Eq.(5.8) as plain lines for the correlation length.

**Figure 33:** Empirical models of the correlation length compared to selected measured data (symbols). From Roger (2013).
Upper and lower curves stand for the higher and lower values of the boundary-layer thickness. They represent indicative uncertainties in the input data. In the next part of the project more realistic values could be deduced from computations performed by other partners. Note that the present rotor-vs-stator comparison is not contradictory with previous competition criteria between TI noise and TE noise of an airfoil. Furthermore it is worth noting that the present estimates are based on the assumption of optimum design blades and vanes, free of separation. Because the empirical fit of Eq.(5.8) is based on the assumption of a log-normal law for the spanwise coherence and because reliable measurements cannot be performed at higher frequencies, Corcos’ assumption is also tested against the data, according to the expression

\[ \frac{\ell_s(\omega)}{\delta_l} = 0.8 \frac{St}{\text{St}} \]

Because that model is invalid at lower frequencies (obviously in Figure 33), it is only considered for the middle-and-high frequency range. A significant scatter is observed between the various estimates but the same orders of magnitude are found. Quite logically, larger values of the displacement thickness lead to more sound at low frequencies and less sound at high frequencies. Corcos’ model has the effect of increasing the predicted sound levels at higher frequencies.

Turbulence-impingement noise generated by the stator and due to wake turbulence has also been estimated from the expressions of section 5.2.4.2. In absence of accurate wake data the relative flow speed has been taken as 54 m/s, the turbulence rate as \((84/54)\) times 6%, 6% being an indicative wake turbulence rate, and the integral length scale has been taken as 1.5mm. Two predictions are reported in the figure, namely without thickness correction (plain black) and with the correction according to Eq.(5.5-b) (dashed black line). The former is very close to the measured spectrum; the latter is about 15 dB lower, which seems quite exaggerate. Because the thickness effect has been characterized for very different scales and for homogeneous and isotropic turbulence to produce the empirical expression, it could be questionable in the present case. The true broadband wake-interaction noise is believed somewhere in between both estimates plotted in black. All possible sources of broadband noise are not taken into account in the analysis, therefore any prediction should remain below the measured spectrum. Finally turbulence-impingement noise appears as a good candidate for the dominant contribution.

**Preliminary Conclusion**

The present test makes wake interaction the expected dominant source of broadband noise in the ECS fan, with a secondary contribution of rotor trailing-edge noise. In contrast stator trailing-edge noise is probably negligible. Ignored sources still being to quantify are:

- Rotor noise due to the interaction of the blade tips with the casing boundary-layer turbulence. This noise will be predicted using a dedicated version of the turbulence-impingement noise model taking as input an extrapolation of the data measured on the test bench at various upstream cross-sections.
Separation noise either on rotor blades or stator vanes. No existing model has been identified for this mechanism. A feasibility analysis will be attempted in the following tasks of the project is separation is predicted by the numerical simulations or suggested by some additional measurements.

5.2.5 In-Duct Turbulence Modelling
The present section is dealing with advanced aspects of turbulence modelling for future implementation in a realistic fan environment, in cylindrical coordinates. The basic models are again von Kármán’s and Liepmann’s models introduced in section 3.2.

5.2.5.1 Model Turbulence in Cylindrical Coordinates
For broadband noise estimates the incident disturbances on a blade or vane row can be described by a statistical approach based on the assumption of frozen homogeneous and isotropic turbulence. At some radius \( r \) any component \( w \) of the two-dimensional velocity vector in cylindrical coordinates \( \mathbf{u} = (u_\theta, u_z) \) is considered in the frequency domain, with the expansion on azimuthal modes imposed by the periodicity condition:

\[
w(\theta, z, t) = \int_{-\infty}^{\infty} \tilde{w}(\theta, k_z) e^{-i k_z z} e^{i k_z t} \, dk_z, \quad k_z = \omega / W_0,
\]

\[
\tilde{w}(\theta, k_z) = \sum_{n=-\infty}^{\infty} w_n(k_z) \, e^{i n \theta}, \quad w_n(k_z) = \frac{1}{2\pi} \int_{0}^{2\pi} \tilde{w}(\theta, k_z) \, e^{-i n \theta} \, d\theta.
\]

For a true random process only statistical quantities make sense; the following identity expressing the orthogonality of azimuthal wavenumbers is used to introduce the two-wavenumber spectrum \( \Phi_{WW} \) of the convected turbulence (Roger 2010):

\[
\lim_{T_\infty \to \infty} \frac{\pi}{T_\infty} \left\langle w_n(k_z) w_m^*(k_z) \right\rangle = 2\pi \frac{W_0}{r} \delta(n-m) \Phi_{WW}(k_z, n/r)
\]

in which the symbol \( \langle \rangle \) stands for the statistical average and \( T_\infty \) for some arbitrary large integration time. The same statistical operator is found in the definition of the power spectral density (PSD) of the acoustic pressure or of cross-power spectral density of the unsteady loads acting as sources (see also Amiet 1975).

In the applications a model turbulence spectrum can be used for \( \Phi_{WW} \), tuning the turbulence intensity \( u_{rms}^2 \) and the integral length scale \( \Lambda \) on values of the turbulent kinetic energy and dissipation as produced by a steady RANS simulation, for instance. In Liepmann’s and/or von Kármán’s models of isotropic and homogeneous turbulence, \( k_1 = \omega / U_0 \) is the streamwise aerodynamic wavenumber and \( k_2 \) the wavenumber in the normal direction. Therefore \( U_0 = W_0 \) stands for the turbulence carried by the axial mean flow in a duct but the same model can be used locally to described the turbulence in a reference frame attached to a rotating blade. In this case \( U_0 \) refers to the chordwise convection speed along the blade chord.
Such a description is not very accurate but it is useful for engineering applications. Turbulence velocity spectra are also known to exhibit a faster high-frequency decay to the Kolmogorov scale due to viscosity. This is not detailed here and is simply accounted for by an additional factor. It must also be noted that large turbulence scales compared to the duct cross-section diameter cannot be modelled this way. The possibility of using an alternative model or a model of anisotropic turbulence is not considered in the present study.

5.2.6 Modulated Turbulence

Non-homogeneous turbulence needs being modelled explicitly in two cases of interest:

1 - In the presence of a stationary inflow distortion the turbulent intensity and length scale a priori depend on the azimuthal coordinate.

2 - When considering the passage of rotor wakes on stator vanes the wake turbulence is seen as intermittent in a reference frame attached to a vane.

Both situations can be reproduced assuming homogeneous turbulence modulated by a periodic envelop, with either time periodicity (case 2) or azimuthal periodicity (case 1).

In case 1 the quantity $\tilde{w}(\theta,k_i)$ must be multiplied by a function of $\theta$, say

$$\hat{\tilde{w}}(\theta,k_i) = F(\theta) \tilde{w}(\theta,k_i) ,$$

so that the modal expansion now becomes

$$\hat{\tilde{w}}(\theta,k_i) = \sum_{m=-\infty}^{\infty} \hat{w}_m(k_i) e^{im\theta} ,$$

with

$$\hat{w}_m(k_i) = \sum_{m=-\infty}^{\infty} F_m w_{m-m}(k_i) , \quad F_m(k_i) = \frac{1}{2\pi} \int_{0}^{2\pi} F(\theta) e^{-im\theta} d\theta$$

The function $F(\theta)$ is of period $2\pi$ for an arbitrary simple distortion and presumably of period $\pi$ in the case of the T-junction inlet. This leads to the corrected turbulence spectrum

$$\Phi_{ww}(k_1,\frac{n}{r}) = \sum_{m=-\infty}^{\infty} |F_m|^2 \Phi_{ww}(k_1,\frac{n-m}{r})$$

In the case of a rectangular inlet or of a T-junction, the azimuthal envelope is ideally of period $2\pi$. As an example consider the function

$$F(\theta) = 1 + \frac{A_0 b}{\sqrt{\pi}} - A_0 \sum_{n=-\infty}^{\infty} e^{-(\theta-n\pi^2/\lambda^2)}$$

that ensures an azimuthal average of 1. This shape features two Gaussian dips of the modulating envelope, $A_0$ being the amplitude of the corresponding distortion and $b$ the characteristic width of the dips. The Fourier coefficients read

$$F_0 = 1; \quad F_{2k+1} = -\frac{b A_0}{\sqrt{\pi}} e^{-k^2b^2}; \quad F_{2k} = 0 .$$
First tests not reported here suggest that the effect of the modulation in the present case is weak.

In case 2 standard derivations based on Fourier transforms and generalised functions are needed. \( w(\theta, z, t) \) must be multiplied by a function \( G(t) \) of period \( T = 2\pi/B\Omega \) and of Fourier transform

\[
\tilde{G}(\omega) = \frac{1}{2\pi} \int_{-\infty}^{\infty} G(t) e^{i\omega t} dt = \sum_{s=-\infty}^{\infty} G_s \delta(\omega - sB\Omega),
\]

introducing the coefficients of the Fourier series of \( G \):

\[
G_s = \frac{1}{T} \int_{0}^{T} G(t) e^{i2\pi st/T} dt. 
\]

The aforementioned analysis performed on \( \tilde{w}(\theta, z, t) = w(\theta, z, t) G(t) \) leads to

\[
\tilde{\tilde{w}}(\theta, k_1) = \sum_{n=-\infty}^{\infty} \tilde{w}_n(k_1) e^{i\phi} = \sum_{s=-\infty}^{\infty} G_s \tilde{w}(\theta, k_1 - sB\Omega/W_0),
\]

which yields the new values of the azimuthal coefficients as

\[
\tilde{w}_n(k_1) = \sum_{s=-\infty}^{\infty} G_s w_n(k_1 - sB\Omega/W_0),
\]

and of the velocity spectrum as

\[
\Phi_{ww}(k_1, n/r) = \sum_{s=-\infty}^{\infty} |G_s|^2 \Phi_{ww}(k_1 - sB\Omega/W_0, n/r).
\]

In practice the function \( G \) is any physically consistent periodic wake envelope, for instance an infinite series of identical Gaussian shapes. This enables expressing the factors \( G_s \) analytically, similarly to what is described in section 3.6 for tonal wake-interaction noise. For instance consider the weighting function

\[
G(t) = \sum_{n=-\infty}^{\infty} e^{-\xi(i\tau + n\tau)^2/\tau^2} = \sum_{s=-\infty}^{\infty} G_s e^{-i2\Omega t} \quad \text{with} \quad G_s = B\Omega \frac{\tau}{2\pi} \sqrt{\pi} e^{-2sB\Omega\tau/\xi}.
\]

\( \xi \) is a constant (here 0.693) and \( \tau \) is the characteristic passage time of the wake-turbulence envelope.

The wake turbulence itself is characterized by an integral length scale which cannot be larger than the half wake width. It is expectedly of a couple of millimeters at the location where the interaction takes place.
6  Data Acquisition

6.1  Hot Wire Anemometry

6.1.1  VKI test rig
The hot-wire carriage displacement and hot-wire signal acquisition is fully automated. Both are driven from a NI PXI desktop through a Labview program interface. Requested positions are sent to an EPOS controller box, connected through USB to the PXI, to rotate the motors also connected to the EPOS. The actual motor positions are sent back to the EPOS using encoders placed on the motor axis. The relation between motor quarter count (QC, unit used by EPOS) and the actual displacement in millimeter is performed through a calibration. The hot-wire signal is sent to the PXI for acquisition. As input to the Labview program, requested coordinates of hot-wire probe and acquisition parameters are only given. The automated measurement process is then fully controlled by the program.

A hot-wire constant temperature mode of operation is used and applied with a VKI home-made anemometer box, to which the probe is connected. The compensation inductance and the offset voltage of the anemometer electrical control circuit can be adjusted to achieve an optimal regulation and then a maximum frequency response. This is performed through a dynamic calibration while the correspondence between voltage and velocity is performed through a static calibration. Given that the electrical hot-wire cables are passing through the duct wall, the hot wire cannot be moved for calibration without disconnecting it from the anemometer. The calibration is then performed on the test rig by opening the duct and placing a calibration nozzle in front of the hot-wire probe. The calibration nozzle is designed to have sufficiently high surface ratio to be able to deduce outlet nozzle velocity from pressure in the stagnation chamber using Bernouilli’s equation.

![Figure 34](image.png)

**Figure 34** (a) Time signal response of the hot wire to a square test signal. (b) Phase-locked averaging of the time signal.

The dynamic calibration is performed by injecting a square wave at the wire terminal and the response is monitored using an oscilloscope. Given the smaller electrical wire sections compared to usual VKI hot-wire probes; the over-heating ratio (directly linked to the reference temperature and the wire temperature) has to be reduced to avoid the melting of the
wire sensor. This has a direct impact on the frequency response of the probe, a better frequency response being obtained with higher over-heating ratios. The optimization of the anemometer should be done for the mean flow velocity at which the hot wire is likely to operate. The hot-wire probe is then placed in a 30 m/s flow obtained in the potential core of the jet flow produced by the calibration nozzle using the pressurized air supply. The acquired time signal (shown in Figure 34 (a)) is phase-locked averaged over 100 periods (shown in Figure 34 (b)) and the frequency response is then determined as about 9.6 kHz (Arts, et. al, 2001). Based on that frequency, the low-pass filter frequency is fixed to 22.5 kHz and the acquisition frequency to 65536 Hz for all measurements.

The static calibration is used to determine the relationship between the acquired voltage and the corresponding velocity. Again, the calibration nozzle allows immersing the hot wire into a known flow velocity, determined from the pressure in the stagnation chamber. Because the compressed air supply temperature can vary in time, the calibration curves are measured twice as shown in Figure 35 (a) and (b). Together with the hot-wire voltage, the temperature is measured using a thermocouple fixed on the hot-wire probe, close enough to the wire sensor to measure the flow temperature to which it is subjected and far enough to not disturb the incoming flow. This repeated calibration allows applying temperature corrections in the post-processing if needed.

To account for temperature changes in the measured flow, temperature corrections can be applied. This correction is based on a priori knowledge of the wire temperature, that can be determined from:

\[ T_\omega = \frac{T_2E_1^2 - T_1E_2^2}{E_1^2 - E_2^2} \]

(a) \hspace{1cm} (b)

where E and T are, respectively, the voltage and temperature obtained for a constant flow velocity at two temperatures 1 and 2. This wire temperature is obtained from the calibration curves or from the first and last radial profiles (0 and 360 degrees) of the complete map for which the flows velocity remained similar but the temperature could have significantly

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The voltage is then corrected for a reference temperature $T_0$ (293.15 K in the present experiments) using the relation:

$$E_0 = E_{\text{meas}} \sqrt{\frac{T_{\omega} - T_0}{T_{\omega} - T_{\text{meas}}}}$$

Hot-wire measurements have been performed at different positions along the axial duct and for the different available inlet modules. The same measurement grid has been used for all tests. The grid, shown in Figure 36 (a), is constructed to obtain acceptable cell ratios once the grid is triangulated. It includes 385 points per plane with 16 radial positions and a minimum distance of 2mm from the wall, due to technical constraints of the hot-wire carriage. The path followed by the hot-wire probe is shown in Figure 36 (b) and is determined minimizing the probe displacement. At each location, a signal of 15 s length is acquired at an acquisition frequency of $2^{16}$ Hz. The total time per measurement plane is then about 2 h. Together with hot-wire signal, other flow diagnostics are measured as temperature or pressure.

Hot-wire mean velocity maps are shown in Table 7 for the different inlet modules. Planes are given at different axial positions along the duct, described in Figure 37. The distance from the step in the ECS fan is used in the shared data for sake of clarity. Only plane 1 (close to the inlet module) and plane 4 (just upstream the fan) are available for the rectangular and T inlet modules.
Figure 37 Hot-wire scan positions with respect to the step in the ECS fan duct casing

The velocity maps includes both radial and axial velocity components as the hot-wire sensor measures all components perpendicular to the wire. For all planes, the velocity is extrapolated to the wall by using a spline interpolation and imposing zero velocity at the wall. From the extrapolated planes, the velocity is integrated to obtain the volume flow rate. This allows to verify the coherence between measurements, the volume flow rate being conserved over the different axial positions, and to compare with estimated data in Task 2.2. Corresponding turbulence intensity maps are shown in Table 8, normalized with a 32m/s velocity. For bellmouth inlet, the increase of the boundary-layer thickness is clearly identified, together with an increase of the turbulence intensity close to the wall, while the flow remains uniform azimuthally from stations 1 to 4. In case of rectangular or T-junction, the flow velocity patterns present a butterfly shape for station 1 while the velocity maps are almost uniform for station 4, upstream the fan. With the axial distance, the strong non-uniformity of the turbulence intensity maps observed in station 1 are also smooth out.
Table 7: Hot wire mean velocity map [m/s] and corresponding integrated volume flow rate.

<table>
<thead>
<tr>
<th>Station 1 – 758 mm away</th>
<th>Bellmouth</th>
<th>Rectangular</th>
<th>T-Junction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Station 1</td>
<td>(\dot{m} = 0.5305 \text{ m}^3/\text{s})</td>
<td>(\dot{m} = 0.5236 \text{ m}^3/\text{s})</td>
<td>(\dot{m} = 0.5378 \text{ m}^3/\text{s})</td>
</tr>
<tr>
<td>Station 2 – 578 mm away</td>
<td>(\dot{m} = 0.5239 \text{ m}^3/\text{s})</td>
<td>(\dot{m} = 0.5295 \text{ m}^3/\text{s})</td>
<td>(\dot{m} = 0.5305 \text{ m}^3/\text{s})</td>
</tr>
<tr>
<td>Station 3 – 428 mm away</td>
<td>(\dot{m} = 0.5295 \text{ m}^3/\text{s})</td>
<td>(\dot{m} = 0.5366 \text{ m}^3/\text{s})</td>
<td>(\dot{m} = 0.5349 \text{ m}^3/\text{s})</td>
</tr>
<tr>
<td>Station 4 – 278 mm away</td>
<td>(\dot{m} = 0.5310 \text{ m}^3/\text{s})</td>
<td>(\dot{m} = 0.5366 \text{ m}^3/\text{s})</td>
<td>(\dot{m} = 0.5349 \text{ m}^3/\text{s})</td>
</tr>
<tr>
<td>Station</td>
<td>Distance</td>
<td>( \dot{m} ) 1</td>
<td>( \dot{m} ) 2</td>
</tr>
<tr>
<td>---------------</td>
<td>-----------</td>
<td>------------------</td>
<td>------------------</td>
</tr>
<tr>
<td>1 – 758 mm</td>
<td></td>
<td>0.5305 m/s</td>
<td>0.5236 m/s</td>
</tr>
<tr>
<td>2 – 578 mm</td>
<td></td>
<td>0.5239 m/s</td>
<td>0.5236 m/s</td>
</tr>
<tr>
<td>3 – 428 mm</td>
<td></td>
<td>0.5295 m/s</td>
<td>0.5366 m/s</td>
</tr>
<tr>
<td>4 – 278 mm</td>
<td></td>
<td>0.5310 m/s</td>
<td>0.5366 m/s</td>
</tr>
</tbody>
</table>
Figure 38 (a) Radial profiles of velocity for different azimuthal positions for station 1 and (b) Azimuthal averaging of velocity profiles for station 1 to 4.

Only the T-junction inlet module still presents high turbulence intensity upstream the fan compared to other inlet modules. Local high turbulence intensity spots are also present in some maps and correspond to possible acquisition problems in the automated procedure. The azimuthal uniformity of the flow is checked for the bellmouth case for station 1 in Figure 38 (a). Radial cuts are performed for 8 azimuthal positions and averaged. Corresponding azimuthally averaged velocity profiles are extracted for all stations in Figure 38 (b). The increase of the boundary-layer thickness is shown except for station 2 where the global velocity is too low. This is confirming that the volume flow rate of station 2 is slightly lower than for other stations, as shown in Table 8.

The measured planes are available to IDEALVENT partners in CGNS format (http://cgns.sourceforge.net/) and are labeled as:

X_section_Y.cgns

Where X is B, T, R for Bellmouth, T-junction or Rectangular inlet module respectively, and Y is the station number as defined in Figure 37. The data are written as unstructured grid with the coordinates and connectivity available in the CGNS file. The following variables are also available:

- V_mean_hw : Velocity measured by the hot-wire probe, including temperature correction and interpolation in the boundary layers
- V_std_hw : Standard deviation of the velocity measured by the hot-wire probe, including temperature correction
- V_prandtl_end : Centerline velocity measured by a Prandtl tube located at the end of the duct
- T : Flow temperature measured at the module inlet by a thermocouple
- Patm : Atmospheric pressure in the control room
- P_room_up : Pressure in the upstream anechoic room relative to the control room pressure
- $P_{\text{room\_down}}$ : Pressure in the downstream anechoic room relative to the control room pressure
- $V_{\text{prandtl\_tube}}$ : Centerline velocity measured by a Prandtl tube located inside the duct downstream the hot-wire probe
- Measured : 1 value if the $V_{\text{mean\_hw}}$ variable has been measured by the hot-wire probe or 0 if the value has been interpolated

Note that all the described variables are not available for all the planes. Only the hot-wire values are measured locally on the grid of Figure 36 (a). All the other variables are fixed probes recording a new value for each hot-wire acquisition point.

### 6.1.2 KUL test rig

Hot wire measurements are carried out to obtain high frequency information on the aerodynamic field including data in close vicinity of the walls. The measurements are carried out on both a horizontal $Y$ and vertical $Z$ line. The raw data pre- and post-processing is carried out using Dantec Streamware Pro V10. For all hot-wire measurement an L-shaped probe (type: Dantec 55P14) is used. The probe is inserted through insertion holes, shown in Figure 39, with a diameter of 10mm which are positioned at $0^\circ$ and $90^\circ$. In order not to cause too much leakage through the insertion holes, the hole is partially sealed, after entering the probe using a specific metal mounting. Leakage however cannot be avoided due to the continuous traversing of the probe.

The probes are calibrated using a Dantec 90H02 Flow Unit. The calibration is carried out for a logarithmic distribution of 35 velocities ranging from 2m/s up to 70 m/s with a continuous monitoring of the flow temperature. In order to take into account the varying temperature of the measured flow, a temperature correction is added during acquisition to take into account the variation in the voltage of the hot-wire probe with varying temperature. For the unsteady flow measurements the data acquisition is carried out using a Dantec Streamline acquisition system and the voltages of the hot-wire sensors are send to the processing software using a multifunctional I/O with correlated digital I/O for USB (NI USB-6229).

The probe is positioned linearly along the vertical and horizontal line with a spatial resolution of 0,2mm leading to a total number of, approximately, 400 measurement positions. The traversing is carried out using a Dantec light-weight traverse system with 3 translational degrees of freedom. For every position the time signal of the $x$-velocity (i.e. the stream-wise velocity) is measured with a sample frequency of 8,00kHz and with a total number of 16 384 samples for each measurement. The raw data are afterwards converted and processed with Matlab. The processed *.mat files are available for the consortium partners.
6.2 Particle image velocimetry

6.2.1 VKI test rig
PIV measurements were conducted by VKI for the diaphragm cases. However, due to optical access issues (partly due to the curvature of the duct and partly due to the poor optical quality of the Plexiglas used to manufacture it), the accuracy obtained from these measurements was not deemed sufficient to allow proper validation of the numerical works. Therefore, it was agreed within the consortium to base the CFD validation on the hot wire measurements only.

6.2.2 KUL test rig
The visualization of the flow field, measured using particle image velocimetry (PIV), is carried out using time-resolved 2D PIV. A laser sheet with a thickness of 0.5 mm is generated using a Dual Cavity Nd:YLF Pegasus-PIV laser from NewWave with a wavelength of 527 nm and a pulse energy of 10 mJ @ 1,000 Hz is inserted at the downstream centerline of the duct, shown in Figure 40. Due to the finite thickness of the laser sheet in combination with the, relatively, small radius of the ducts ($D=84mm$), diffraction effects made it impossible to yield accurate results when inserting the laser shield, using 45° mirrors, not directly into the flow field; even when the plexi-glass measuring duct was ‘straightened’ at the outside end to a wall thickness of approximately 2mm. As such, the only way of perform measurements upstream of the valve would be introducing a square measurement duct which would introduce increased turbulence levels upstream of the duct and, thus, would not produce reliable results for the validation of the (un-)steady CFD simulations. As such, it was decided to perform the PIV only at the downstream end of the butterfly valve.

The images of the movement of the particles, are recorded using 1 ‘HighSpeedStar 5’ CMOS camera with a resolution of 1,024 x 1,024 pixels. Due to hard- and software issues, related to frequency resolved PIV-system which is used within this project, the maximum sampling frequency is limited to $1024Hz$. When the sample frequency is further increased, this would lead to a decrease in spatial resolution. In order to validate the steady CFD-results, a lower sample velocity could be more beneficial, however, given the large local axial velocity in the close vicinity of the valve, a decrease of the sampling frequency, and, thus, an increase of the time step of the measurement leads to a poor capturing of the particle movement. As such, the sample frequency is set to $1.024 kHz$, which provides an optimal compromise, with respect to
accuracy, between spatial resolution (hard/software-defined) and temporal accuracy (i.e. tracking the particle movement).

The camera is positioned, as shown in Figure 40, perpendicular with the laser sheet plane, in close vicinity of the plexi-glass measurement section, for both the horizontal and vertical laser sheet alignment. A Scheimpflug adapter ensures that particles within the light sheet are focused throughout the entire measurement area. The calculation of the velocity vectors is done using the DaVis 7.1 software of LaVision GmbH. The calibration of the PIV system is done using a 2D calibration plate with dimensions equal to 58mm x 58mm. Due to the distortion of the camera view angle in the vicinity of the upper- and lower- vertical and horizontal walls the actual measurement plane is limited to, approximately, [-30mm, 30mm] in the flow X direction and [-35mm, 35mm] in the horizontal Y and vertical Z direction with a spatial resolution of, 1.6579 mm. In this framework, the HWA measurements provide more detailed information about the flow field in the vicinity of the wall as well as the ‘high’-frequency content of the turbulent velocity field. The final data are processed with Matlab and the *.mat files are available for the project partners.

![Figure 40: Picture of the direct insertion of the laser sheet and camera position during calibration](image)

### 6.3 Acoustic multi-port characterization

#### 6.3.1 VKI test rig

The passive part of the acoustic modal decomposition requires the combination of different source, generated by different loudspeakers, and observer locations. Each loudspeaker is used for the desired frequency at a time and the procedure is repeated for each loudspeakers and frequencies. In order to operate the test-rig, a control and acquisition set-up is created. A sketch of the control and operation cycle is shown in Figure 41.

The microphone signals are acquired by a National Instruments (NI) acquisition unit as shown in Figure 42 (left top). All microphone signals are acquired synchronously. A NI CompactDAQ case and board are used to generate the output signal from the PXI unit (Figure 42 right top).
The loudspeakers are fed by a switch-box designed and manufactured as shown in Figure 42 (bottom). The switch-box consists of three control boxes containing 8 loudspeakers each. The switch-box, also called as multiplexer, sends the signal received from CDAQ to the related loudspeakers. In order to control if the loudspeaker is damaged or not during the experiments, the output signal of the loudspeakers is also acquired. This is shown as the feedback in Figure 41.
In order to control the hardware, a NI Labview code is developed as shown in Figure 43. The code allows simultaneous excitation of four loudspeakers for different frequencies. “Step sine” signals are selected to drive the loudspeakers. The reason of using “step sine” signal is to obtain more averaging due to tonal excitation compared to chirp signal. In case of possible replacement or repetition of measurements, the acquired data are saved in different frequencies and different loudspeakers. Therefore the complete configuration is not needed to be repeated, but only the necessary part can be re-measured.

To avoid possible failure of loudspeakers the ‘step sine’ signal is guided by a ramp-up and followed by a ramp-down function. Hence a sudden voltage load to the loudspeaker membrane is avoided. The ramp-up and ramp-down time is 0.5 second each, and no acquisition is performed during ramps. Depending on the frequency of interest, the acquisition times are varying from 10 seconds to 60 seconds, for lowest and highest frequencies, respectively. Even that the highest frequency of interest is around 3.5 kHz for this configuration; the sampling frequency is selected as $2^{15} = 32768$ Hz.

During the measurements, the pressure at both of the anechoic rooms is acquired with temperature and centerline flow velocity. The acquired data is saved in ‘.tdms’ file in an external disk and shared with partners.

6.3.2 KUL test rig
In order to perform the active aeroacoustic multiport measurements, dedicated loudspeaker and microphone arrays are designed by KTH and built by KUL. According to the requirements of WP2, the frequency range for the measurements reaches from 600 Hz to 3600 Hz, the medium is air ($c_0 \approx 340m/s$) and the duct diameter $D$ equals 84 mm. Based on the geometry, the cut-on frequencies for the propagating modes can be calculated. For a quiescent
medium, the cut-on frequency of the first azimuthal mode \((\pm 1,0)\) occurs at 2372 Hz while the second azimuthal mode \((\pm 2,0)\) and first radial mode \((0,1)\) are cut-on at, respectively, 3935 Hz and 4937 Hz. As such, 3 modes propagate within the previously defined frequency range: the plane wave mode \((0,0)\) and the first circumferential mode \((1,0)\) together with its complex conjugate \((-1,0)\).

The design of the loudspeaker section is focused on the over-determination of the scatter matrix calculation. Since the actual position of each loudspeaker is of importance neither for the mode decomposition nor for the scatter-matrix calculation, a random distribution may be valuable in order to avoid singularities within the measurement range. To define the degree of over-determination, the sound fields created by the loudspeakers were calculated. To achieve low uncertainties with a, due to spatial constraints, reasonable number of sources, 6 loudspeakers (Morel EM428) are mounted on both the upstream and downstream loudspeaker array, shown in Figure 44. The loudspeakers are mounted on an aluminum duct with a length equal to 6D. The loudspeakers are mounted on top of a perforated section with an intermediate distance of 65 mm between the different speakers. Their relative angular position corresponds to \(0^\circ, 180^\circ, 50^\circ, 230^\circ, 90^\circ\) and \(270^\circ\).

![Figure 44: CAD representation (left) and picture (right) of the loudspeaker section](image)

To capture the sound field, a total of 18 \(\frac{3}{4}\)" microphones (12 PCB 378C10 and 6 BK 4938-A11 high pressure microphones) are used, equally split between the two sides. The microphone arrays, shown in Figure 45, have a length equal to \(4D\) and are made out of plexiglass, the microphones are mounted flush with the wall and are placed on dedicated placeholders which enable interchangeability between the two different types of microphones. They are separated in 3 cross-sections up- and downstream with an angular position of, respectively, \(0^\circ, 120^\circ\) and \(240^\circ\). The distance between the first cross section \(A\) and the second one \(B\) equals 40mm and the distance between the first cross section \(A\) and the last one \(C\) equals 110mm. The microphones are positioned as far as possible from the valve in order to reduce high-amplitude aerodynamic pressure fluctuations inside the microphone signals as much as possible. The microphone modules are mounted at a distance of \(4D\) from the loudspeaker modules in order not to measure in the near-field of the loudspeakers.
Before mounting the microphones on the different arrays, they are calibrated relative to each other using a dedicated calibration duct, shown in Figure 46. The calibration duct allows calibrating 8 microphones simultaneously. For the calibration procedure, one microphone is chosen as reference microphone and the transfer functions between all microphones are used to obtain proper values for both the phase and amplitude calibration. The calibration is carried out using a stepped sine excitation (similar as for the acoustic multi-port measurements) of a horn driver, located at one end of the calibration duct. The reference microphone is calibrated absolutely using a 1000Hz sine wave with an amplitude equal to 94dB.

In order to improve the measurement accuracy of the multiport measurements, an anechoic termination, shown in Figure 47 is designed by KTH. The anechoic termination is mounted at the downstream end of the downstream loudspeaker array. The anechoic termination is made of plywood and contains micro-perforated walls to avoid reflections at the downstream end.
For the determination of the multimodal scatter matrix, 12 different loudspeaker combinations are used. The loudspeakers are excited using a stepped sine excitation between 600 Hz and 3600 Hz with a frequency step of 20 Hz. The amplitude of the loudspeakers is varied over this frequency range in order to have an SPL of, approximately 130 dB (which is safely below the generally considered maximum amplitude for linear acoustic phenomena) at all frequencies. The data acquisition is carried out using two 16 channels LMS SCADAS III systems which are installed in a master-slave configuration. The raw data pre- and post-processing is carried out using LMS Test.Lab Rev.13A.

For the analysis of the microphone signals, at each excitation frequency, the data are directly narrow-band filtered over 99 cycles, yielding directly a single frequency pressure value for each step. This procedure is repeated for 50 frequency sweeps, thus, yielding a total number of, approximately, 5000 averages. Afterwards, the auto- and cross- correlations between the microphones and source are directly computed using Test.Lab and the data are stored in *.mat and *.txt files which are available for the consortium partners.

For the active multi-port characterization no loudspeaker excitation is present and the pressure fluctuations at the 18 microphone positions are directly transferred to the frequency domain using an FFT. The frequency step equals 1.25 Hz for these measurements and a sample frequency of 40,960 kHz was chosen. In total 1000 averages were taken for these measurements.

In order to validate the data acquisition approach using a stepped sine excitation with different types of microphones, as well as, to verify the post-processing routine to determine the scatter matrix coefficients, passive multi-port measurements are carried out on an empty plexi-glass duct both under quiescent conditions and with the presence of a flow with a centerline velocity of, approximately, 33 m/s, which is slightly above the operational flow velocities of the envisaged detailed measurements due to fact that not sufficient back pressure is provided by the empty duct system to achieve the required flow rate with the dual Root’s blower configuration. The different elements of the scattering $\mathbf{S}$, representing the transmission and reflection coefficients of each propagating mode in both the upstream and downstream direction are shown, respectively in Figure 48 and Figure 49.
Figure 48: Reflection and transmission scatter matrix coefficients of an empty duct without flow for the, from top to bottom, (0,0)-upstream mode, (1,0)-upstream mode, (-1,0)-upstream mode, (0,0)-downstream mode, (1,0)-downstream mode and the (-1,0)-downstream mode.

The results for both the quiescent case and the flow case are in good agreement with the expected solution: all acoustic waves are transmitted without reflection (i.e. the magnitude of the transmission coefficient equal to 1 and reflection coefficient equal to 0) starting from the cut-on frequency, which is, for the plane wave mode and the first circumferential equal to, respectively, 0 Hz and 2372 Hz. At the cut-on frequency (indicated with a dashed vertical line on Figure 48 and Figure 49), indeed a sudden raise of the transmission coefficient and decrease of the reflection coefficient is observed. The presence of spurious high-frequency oscillations for the empty duct with flow measurements is caused by the presence of aerodynamic pressure fluctuations, which are present in the different microphone signals, thus
reducing the signal-to-noise ratio of the different pressure spectra. The analysis of these measurements is carried out using both the post-processing routines of KTH and KUL and no differences are noticed. As such, these empty duct measurements do not only validate the actual test rig implementation and data acquisition approach but also verify the correct implementation of the modal decomposition post-processing routines.

**Figure 49:** Reflection and transmission scatter matrix coefficients of an empty duct with flow ($U_0 \approx 33 m/s$) for the, from top to bottom, (0,0)-upstream mode, (1,0)-upstream mode, (-1,0)-upstream mode, (0,0)-downstream mode, (1,0)-downstream mode and the (-1,0)-downstream mode.

For the determination of the active source vector $p^S_{\text{active}}(\omega)$ not only the passive scattering matrix $S^s$ is required but also the reflection matrix $R$, which is, for the upstream end defined as the
ratio of the downstream propagating wave to the upstream propagating wave and vice versa for the downstream end:

\[ R_t = \frac{p_i - p_{i+1}}{-1} \]

For the determination of the reflection coefficients, a similar test approach as for the passive scattering matrix determination can be used, since a similar multi-modal decomposition of the fluctuating acoustic pressure fluctuations into an upstream and downstream propagating field is needed. To determine e.g. the reflection coefficient of the upstream end (i.e. the reflection caused by the combination of the dual Root’s blowers, heat exchanger and the various duct system components), the empty duct configuration with flow is used. The acoustic field is excited with the downstream loudspeaker array and the upstream microphone array is used to acquire the acoustic pressure signals, which are finally decomposed in their upstream and downstream propagating components. For the determination of the downstream reflection coefficient (i.e. of the anechoic termination) a similar configuration is used with, respectively, the upstream loudspeaker array and the downstream microphone array.

The results of these measurements are shown in Figure 50. It can be noticed that the reflection coefficient of the upstream end is higher in comparison to the one of the downstream end which is caused by the fact that no acoustic dissipative materials are installed upstream of the test section and due to the presence of a number of 90° bends which cause back-reflections of acoustic waves in the downstream direction. For the anechoic termination (i.e. the downstream termination of the test rig), the reflection coefficient is smaller (around 0.15 for the plane wave mode) due to the presence of the micro-perforated materials. An anechoic termination \( R_t \approx 0 \), which was originally envisaged, is, however, not obtained. Since this is only required to improve the measurement accuracy of the scattering matrix determination but is not an absolute necessity, it is decided, due time-constraints, not to redesign the anechoic termination. Furthermore, the good results, obtained for the empty duct validation both with and without flow, show that the current termination provides enough accuracy to perform the active multi-port characterization.
Figure 50: Reflection coefficients of the upstream (upper 3 graphs) and downstream (lower 3 graphs) test rig termination with flow \((U_0 \approx 33 \text{ m/s})\) for the (0,0), (1,0) and (-1,0) mode.
7 Discussion of the results

7.1 Fan measurements

The data post-processing comprises 3 steps

- Computation of transfer-functions between microphones and sources using auto-correlation and cross-correlation data
- Decomposition of the each sound-field
- Computation of the scattering-matrix and the source-vector

To reduce measurement uncertainties and uncorrelated noise, cross-spectrum functions between the loudspeaker sources and the microphones are used. The transfer-function may be written as

\[ p^2 = \frac{C_{1m} - C_{1m}^*}{A_{11}} \]

where \( C \) is the cross-spectrum density, \( A \) is the auto-spectrum density and the indices \( l \) and \( m \) denote loudspeaker and microphone signals respectively. The spectra are calculated with Welch's method. The windows length was chosen to be 1 s to archive a resolution of 1 Hz. The sample frequency is 32768 Hz, which was mainly given by the cut-off frequency of the low-pass filter at 10000 Hz. The sample time was estimated based on signal-to-noise ratios of preliminary test-measurements.

The microphones are relatively calibrated with a calibration tube developed at KTH. Two reference microphones are used to calculate transfer functions between all microphones. The active calibration is performed on 1000 Hz with the reference tone of 94 dBd.

The post-processing codes are written in parallel python using the packages Numpy, Scipy and Matplotlib in there up-to-date versions (Summer 2014). The results are available as .txt and .npy and .mat files.

Measurements are taken on the different test elements. The empty duct under the presence of flow was a useful validation case, since the analytic solution is well known. The results will be presented for different flow cases to show the capability of the method. Furthermore, with the Liebherr fan and the wide orifice, the main-measurements of Task 2.5 are presented.

7.1.1 Measurement results: Empty duct with flow

The results for assemblies with different mean flow velocities were evaluated in means of their relative deviation from the analytical solution. Therefore, the scattering matrix was measured at 100 frequency points. In order to reduce the measurement time and to ensure constant conditions, the single frequencies were partly simultaneously exited with step-sinusoidal source combinations. Figure 51 a) shows the absolute value of the reflection and transmission coefficients for the plane wave mode which were calculated from the measurements. The values come close to the analytical solution, which is presented in Figure...
51 b). Until the cut-on frequency of the (0,1)-mode, the deviation is steadily less than 5%. The condition number (depicted in Figure 51 c)) increases with the number of propagating modes as it can be expected from the analytical investigations. However, when reaching the cut-on frequency of the (3,0)-mode, the condition number increases significantly. This is indicative of a deficient over-determination that goes along with ill-conditioned sound fields (compare Figure 5). The result is a higher amplification of input errors and thus larger deviations from the exact solution (Figure 51 b)). These input errors have various causes even for a straight duct without flow, e.g., defective microphone calibration, imperfectly manufactured microphone/loudspeaker arrays, or uncertainties in the wave-number determination and can hardly be removed.

The measurements were repeated under the presence of a mean-flow with a velocity of 32 m/s. Due to the higher noise level, the number of measured averages was increased. Figure 52 shows resulting elements of the scattering matrix in comparison to the measurements without flow. First, the scale of the relative error does not change under the presence of flow, which indicates a stable method. Secondly, the derivation of different modes from the analytical solution can be compared. The error increases with the mode order, which is due to the declining over-determination of both, microphones and loudspeakers. In the investigated case, the method seems to be more sensitive for the reflection determination. However, this is caused by the steadily small values, which are more sensitive even for tiny disturbances.

![Figure 51: Empty duct case: a) Reflection (dotted) and transmission (solid) of the plane wave mode. b) Deviation of the reflection from the analytical solution. c) Condition number. $S_{kl}$ denotes the scattering matrix with $k, l < 8$ representing the inlet-side modes and $k, l > 8$ representing the outlet side modes.](image-url)
Conspicuous are the magnitude bumps in the reflection coefficients, which are contributed equidistantly over the whole spectrum. Their positions change with the mode order and hence seem to be related to the wave number.

7.1.2 Measurement results: Diaphragm alone
A circumferential diaphragm of the diameter 116mm and a thickness of 8mm inserted in the duct (diameter 150mm) was investigated in the absence and presence of mean flow (M = 0:06 and M = 0:08). Figure 53 shows the resulting scattering matrix for the plane wave. The reflection and transmission coefficients are similar to the empty duct for low frequency, for which the diaphragm is acoustically invisible. The diaphragm becomes more reflective in the higher frequency ranges, starting at 2400 Hz. To investigate the behavior of the diaphragm under mean flow, the dissipation factor $\Gamma$ is calculated. The result is shown in Figure 54. For low frequencies, the dissipation increases in the presence of flow significantly. The flow separation at the sharp edges of the diaphragm dissipates energy from the acoustic field into the vorticity field. This effect is much stronger than the energy dissipation evoked by viscous effects at the duct wall and at the diaphragm.
Figure 52: Scattering of the empty duct with flow (M=0.08) for a) (0,0)-mode, b) (1,0)-mode, c) (2,0)-mode, d) (0,1)-mode and e) the (3,0)-mode.
Figure 53: Scattering of a diaphragm, M=0.08. The orifice is acoustically invisible for low frequencies and becomes more reflective for higher frequencies.
In frequency ranges, the dissipation factor increases and reaches zero at 1250 Hz. For those frequencies energy from the vorticity field dissipates to the acoustic field, amplifies the sound field and the diaphragm becomes susceptible for whistling effects. Figure 55 shows the minimal and maximal eigenvalue in absence and presence of flow. The power scattering matrix was calculated and the plane wave mode has been isolated to investigate its scattering separately.

![Figure 54: Dissipation factor for a diaphragm in absence and presence of flow](image)

This approach is valid up to the cut-on of the radial mode, since the plane wave and the circumferential modes will not interact due to their difference in spatial shape and propagation. Both $1 - \Gamma_{\text{min}}$ and $1 - \Gamma_{\text{max}}$ show values close to 0 in absence of flow. The slight descent for higher frequencies is due to the viscose effects within the duct. In the presence of flow, the values for $\Gamma_{\text{max}}$ change significantly. The potential dissipation increases for low frequencies which are related to increasing losses in the velocity field. The maximum at approximately 1200 Hz confirms the perception gained from the investigations on the averaged dissipation factor $\Gamma$. For higher frequencies, the eigen-values decrease much faster in presence of flow due to the acoustic energy dissipation into the vorticity field at the sharp edges of the diaphragm.

![Figure 55: Eigen-values of the power scattering matrix $\Gamma_{\text{min}}$ and $\Gamma_{\text{max}}$ for a diaphragm in absence and presence of flow for the isolated plane wave](image)
7.1.3 Measurement results: Fan

For the Liebherr fan, the source vector is of particular interest. However, in order to remove reflections from the test rig, the scattering matrix has to be determined in a first step. During the measurements, two problems appeared. First, the cut-on frequency of the (0,1)-mode is very close to the BPF of the fan, which will induce high uncertainties for those frequencies. Second, the signal-to-noise ratio becomes high close to the BPF, which induces uncertainties in the scattering calculation. Figure 56 shows the signal to noise ratio for the measurement based on the signal coherence for the frequencies around the BPF. At many frequencies, the expected measurement uncertainty is higher than 2 percent.

![Figure 56: Signal to noise ratio for the measurements and different microphones: (red) average of up-stream microphones, (blue) average of down-stream microphones, (black) average of all microphones, (grey) limit for weakest and strongest microphone.](image)

The scattering-matrix was calculated as it can be seen in Figure 57. It shows oscillating reflection and transmission behavior depending on the frequency.

In a second step, the reflection coefficients of the test rig terminations were measured on the empty-duct with flow (M=0.08). The result can be seen in Figure 58. It shows that the anechoic termination is not as silent as expected, especially between 1000 Hz and 2000 Hz. However, that will not influence the measurement results.

The active part was obtained in 1 Hz resolution under the consideration of 2000 averages. However, since the resolution of the scattering-matrix and the reflection-matrix is lower, the needed values were interpolated. A linear interpolation for real and imaginary part has been applied respectively. Figure 59 shows the spectral power density of the fan. The actual values are calculated

\[
SPL = 10 \log_{10} \left( \frac{\left(G_{mm}^s\right)^2}{p_{ref}^2} \right)
\]

where \(m\) denotes the mode order and \(p_{ref} = 10^{-5}\) Pa. The BPF-peak is located at 2700 Hz and has a maximum for the plane wave mode of 90 dB.
Figure 57: Scattering-matrix of the ECS fan on operating point (M=0.08)
Figure 58: Reflection for the anechoic termination measured for an empty duct with flow (M=0.08)
Figure 59: SPL of the fan
The dominating acoustic modes based on the measured data are seen in Figure 60. It is observed that the low-order modes can be dominating at higher frequencies.
7.2 Valve with inflow distortions

7.2.1 Discussion of the HWA-results
In this section the results obtained with the HWA-setup, are discussed in full detail for the valve alone and valve with upstream bend test case. Since both the HWA and PIV measurements are carried out on a horizontal XY-plane and a vertical XZ-plane, a positive direction needs to be defined. As shown in Figure 61, a right-handed Cartesian coordinate system (with the origin in the center of the duct) is chosen with a positive x-axis pointing in the flow direction and the positive z-axis pointing upwards.

![Positive direction for the different aerodynamic measurements.](image)

The data processing of the HWA presented in the deliverable is limited to a discussion of the time-averaged axial flow velocities $U_{x0}$ and the turbulent intensities ($TI$) defined as the ratio of the root-mean-square (rms) value of the stream-wise velocity fluctuation $u'_x$ with respect to the mean flow velocity $U_{x0}$:

$$TI(\%) = \frac{(u'_x)_{rms}}{U_{x0}}$$

This is mainly due to the fact that the acquired velocity data do not contain any tonal phenomena, neither a significant cut-off frequency of the turbulent velocity fluctuations is observed. More in-depth time- and frequency-domain information of the acquired data are, however, accessible for the consortium partners since the full set of 16 384 sampled data, obtained with a sampling frequency equal to 8.00kHz is recorded and available.

7.2.1.1 Valve alone
The left of Figure 62 shows the stream-wise velocity profile, obtained with HWA for both the horizontal and vertical axis. A similar profile is noticeable for both the y-axis and z-axis indicating that an axisymmetric velocity profile can be applied for the numerical simulations. A sudden increase in the x-velocity is observed at large (positive) values of the radial position. This is caused by leakage effects of the insertion of the hot-wire probe. Due to the, relatively, high pressures in the, with respect to the valve position, upstream duct, this causes an attraction of the fluid towards the insertion hole. As such the HWA measurements should be discarded starting from +20mm for both the horizontal and vertical axis. The turbulent intensity profiles, shown on the right of Figure 62, show the expected behavior. Inside the boundary layer, extending up to a distance of 10mm-15mm from the duct walls, the turbulent intensity gradually decreases from 12% towards 7%; when the axial flow velocity reaches a constant value of, approximately, 33.5 m/s, the turbulent intensity asymptotically goes
towards 6%. These medium turbulence levels, observed at the inlet, are caused by the fact that inside the duct system, guiding the flow from the Root’s blowers and heat exchanger towards the test section, via a duct system with four 90° bends, no turbulence decreasing measures, such as e.g. expansion chambers, turbulent screens or honeycomb structures, are present.

Figure 62: Inlet velocity profiles [m/s] (left) and turbulent intensity [%] (right) vs. radial position for the valve alone measurement on a horizontal (blue) and vertical (red) line.

The mean velocity profile at a distance of 1.5D downstream of the valve is shown in Figure 63. Although the inlet velocity profile can be assumed to be axisymmetric, the obtained HWA-measurements show large axial flow velocities in close vicinity of the –y and +z wall (i.e. the top-left side of the duct when looking upstream). Due to the fact that, in the neighborhood of the centerline, a minimum in the velocity profiles is present, it can be concluded that the hot-wire measurements are still carried out in the wake of the valve and that no reattachment of the streamlines has taken place.

Figure 63: Mean velocity profile [m/s] at a distance of 1.5D downstream of the valve. (left: horizontal axis; right: vertical axis).

When looking at the turbulence intensity levels, shown on the left and right of Figure 64 for, respectively, the horizontal and vertical axis, it is noticed that the turbulent intensity reaches maximum values of 50% to 60%, thus offering severe challenges for the numerical CFD-simulations of the valve case, envisaged in WP4. The maximum levels of the turbulent
intensity are encountered inside the boundary layer region and in the wake of the valve, i.e. close to the centerline. In the regions where a high mean flow is present, the turbulent intensity levels decrease up to, approximately, 45%. This indicates that the valve introduces large turbulent velocity fluctuations, which need to be properly taken into account by the (un-)steady CFD simulations, in order to accurately predict numerically the aerodynamic noise generating mechanisms.

![Figure 64: Turbulent intensity [%] at a distance of 1.5D downstream of the valve (left: horizontal axis; right: vertical axis).](image)

### 7.2.1.2 Valve and Bend

The inlet velocity profiles and the according turbulent intensity for the valve with upstream bend are shown, respectively, on the left and right of Figure 65. Besides the large positive values (i.e. above 20 mm) no significant differences are noticed in the mean velocity profile, as well as, the turbulence intensity levels. Furthermore, as shown in Figure 65, a very similar behavior is obtained in comparison to the valve alone measurements, discussed in section 7.2.1.1. The discrepancies at large (positive) radial distance are, again, caused by leakage effects due to the traverse insertion holes. The final conclusion of the inlet flow velocity measurement shows that, on the one hand, an axisymmetric velocity profile with relative uniform turbulent intensity level of 6%-7% can be used as inlet B.C. for the (un-)steady CFD-simulation of the valve with upstream bend. On the other hand, the distance of 6D upstream of the first obstacle, being either the valve (for the valve only measurements) or the 90° bend (for the valve with upstream bend measurements) is sufficient in order not to show any influence of the downstream obstacle. As such, a similar inflow B.C. profile can be used for both the valve alone and the valve with bend simulations.
Figure 65: Inlet velocity profiles [m/s] (left) and turbulent intensity [%] (right) vs. radial position for the valve alone measurements on a horizontal (blue) and vertical (red) line and the valve with upstream bend measurements on a horizontal (black) and vertical (green) line.

Figure 66 shows the mean velocity profile on the horizontal (left) and vertical (right) axis in between the valve and the bend. Similar as for the inflow velocity profiles, the values at small y-coordinates (< -25mm) and large z-coordinates (>25mm) should be discarded due to leakage occurring at the probe insertion holes, caused by the large pressures inside the duct, upstream of the valve. As can be expected, since the largest bend radius is located in the –y-direction, the flow velocity increases in the –y-direction, meaning the flow is pushed towards the outer side of the bend. As no variation is expected in a vertical plane, a symmetric behavior is noticeable around the z-axis. Close to the centerline, the blockage effect of the valve is noticed by a decrease of the axial velocity in both the horizontal and vertical plane.

Figure 66: Mean velocity profile [m/s] at a distance of 1.5D upstream of the valve. (left: horizontal axis; right: vertical axis).

The mean flow velocities downstream of the valve are shown on the left and right of Figure 67 for, respectively, the horizontal and vertical axis. The results for the valve with bend case (blue lines) are, in Figure 67, also compared with the valve alone (red lines) test case. A first remarkable observation is the fact that for the valve with upstream bend case, although the incoming velocity profile, shown in Figure 66, is not axisymmetric at all, the mean flow
velocity at a distance of 1.5D downstream of the valve shows a significant larger symmetric behavior (for both the y- and z-axis) in comparison to the valve alone test case, where the largest flow velocities were encountered at the top-left side of the duct. As shown is section 7.2.2, these observations are confirmed by the PIV measurements. In order to understand this phenomenon, further information about the 3-dimensional flow field in the close vicinity of the valve, both in the upstream and downstream direction, is needed and can possibly be provided by the (un-)steady flow simulations. Also in contrast with the valve alone measurements, the largest mean flow velocities are present in the vertical plane, where the distance between the valve and its housing is smallest. Finally, due to the non-uniform inflow velocities, the length of the wake of the valve is significantly increased for the valve with bend case. This is noticed by a smaller value of the centerline velocity and large velocity values, with a very small boundary layer thickness for the bend with valve measurements in comparison to the valve alone measurements.

Figure 67: Mean velocity profile [m/s] at a distance of 1.5D downstream of the valve and comparison between the valve with bend (blue) and valve alone (red) test case (left: horizontal axis; right: vertical axis).

The turbulent intensity levels downstream of the valve are shown on the left and right of Figure 68 for, respectively, the horizontal and vertical axis. The results for the valve with bend case (blue lines) are, in Figure 68, also compared with the valve alone (red lines) test case. As expected, both due to the fact that the incoming turbulence levels have increased due to the presence of the bend, as well as, due to the fact that the wake region has been extended, the turbulence intensities for the valve with upstream bend case are, especially close to the centerline, significantly larger in comparison to the valve alone test case. On the horizontal axis a similar pattern as for the valve alone case is observed but the maximum turbulent intensity has increased from 50% to almost 75%. For the vertical axis, a maximum intensity of 67% is observed with a strong decrease towards the duct walls, which is significantly differing from the vertical axis of the valve alone case where an almost constant turbulent intensity was observed.
7.2.2 Discussion of the PIV-results

In this section the results obtained with the PIV-setup are discussed in full detail for the valve alone and valve with upstream bend test case. Since both the HWA and PIV measurements are carried out on a horizontal XY-plane and a vertical XZ-plane, the same sign convention as for the hot-wire measurements, shown in Figure 61, is used. The analysis on the PIV results is limited to the discussion of the mean flow velocities and a comparison/cross-validation with the HWA results. The data, which are made available for the consortium partners, however, contain also the instantaneous velocity fields of which only a limited number are shown in this deliverable.

7.2.2.1 Valve alone

The time-averaged velocity on a horizontal measurement plane in both the x- and y-direction is shown in Figure 69. Since, in contrast to the HWA results, a rather uniform stream-wise velocity pattern is observed, in combination with the positive y-velocity components in the vicinity of both the walls, it can be expected that the PIV results for the valve alone in a horizontal plane are erroneous. Measures have been taken to search for possible reasons for this to happen but no firm conclusions have been made yet. As such, these data will not be discussed in the remaining of this deliverable. If these data are needed in the future of the project for the validation of the (un-)steady CFD simulations, the measurements will be repeated in order to obtain more reliable PIV results for the valve alone test case on a horizontal measurement plane.

Figure 68: Turbulent intensity levels [%] at a distance of 1.5D downstream of the valve and comparison between the valve with bend (blue) and valve alone (red) test case (left: horizontal axis; right: vertical axis).
Figure 69: Contour plot of the time-averaged x- (left) and y- (right) velocity for the PIV measurements on the valve alone carried out on a horizontal plane.

Figure 70 shows the time-averaged velocity field, obtained using PIV in both the x- and z-direction for the valve alone test case, carried out on a vertical plane. Both the stream-wise and vertical velocity components show that the PIV acquisition plane is still located in the wake of the butterfly valve, in the vicinity of the reattachment of the flow field, which is evidenced by the decrease in the z-velocity components. Along the axial position (x-axis) the mass flow rate is not conserved. This is, on the one hand, caused by the presence of 3D flow phenomena, which are not captured with the 2D measurements. The main reason is, however, the fact that, based on the HWA, the largest flow velocities are found in the close vicinity of the walls. Since the high-speed camera images are only calibrated for a square calibration plate with a size equal to 58mm, with it center located at the centerline of the duct, the processed results in the acquired camera images, lying outside this domain, are obtained using extrapolation. However, in this region, i.e. in close vicinity of the walls, the large curvature of duct causes diffraction of the camera images. As such, the results in the vicinity of the wall (i.e. \( z < -30 \text{mm} \) and \( z > 30 \text{mm} \)) should be carefully interpreted. This, in principle, also holds for all PIV data at axial distances which are located at too large distances upstream or downstream from the center point of the calibration plate (i.e. \( x < -30 \text{mm} \) and \( x > 30 \text{mm} \)).
When looking at the axial mean flow velocity along the centerline, shown in Figure 71, the discrepancy between the horizontal and vertical plane measurements is confirmed. For the vertical plane measurements, an increase in the axial flow velocity is noticed with a slight asymptotic behavior at large $x$-coordinates. As such it can, indeed, be expected that both the PIV and HWA measurements are carried out in the neighborhood of the flow reattachment point in the wake of the valve.

![Mean x-velocity profile along the centerline](image1.png)

**Figure 71:** Axial velocity along the centerline for the PIV measurements on the valve alone carried out on a horizontal (blue) and vertical plane (red).

Since both PIV and HWA measurements are carried out downstream of the valve, the outcome of the different aerodynamic measurement techniques can be compared for the vertical line, located at $1.5D$ downstream of the valve. The result of this cross-validation is shown in Figure 72. An excellent agreement between the HWA and PIV results is obtained both in amplitude and vertical variation of the axial velocity component. It can also be noticed that the PIV results are, as mentioned before, only reliable for $-30\text{mm} < z < 30\text{mm}$. This analysis was also carried out for the horizontal line and confirmed the erroneous results of the PIV data on the horizontal plane. As such, the results are not shown in this deliverable.

![Mean x-velocity profile at x=1.5D downstream of the valve](image2.png)

**Figure 72:** Comparison of the time-averaged axial velocity along a vertical line, located at $x=1.5D$, between PIV (red) and HWA (blue).
An instantaneous velocity field for the PIV measurements on the vertical plane is shown in Figure 73. Due to the large turbulent intensity levels (based on the post-processing of the PIV results turbulent intensity levels between 50% and 60% are found, corresponding to the HWA-results shown in Figure 64) a large number of small scale turbulent structures are present in the flow field and the local flow velocities reach peak values up to 75 m/s, being more than double the maximum mean flow velocity. It is likely that these peak values, as well as the maximum mean flow velocity will further increase when the distance to the valve is decreased. As such, a compressible (un-)steady CFD-simulation is indispensable to yield accurate numerical data in WP4 for this application.

![Contour plot of the instantaneous velocity magnitude with additional vector plot for the PIV measurements for the valve alone carried out on a vertical plane.](image)

**Figure 73:** Contour plot of the instantaneous velocity magnitude with additional vector plot for the PIV measurements for the valve alone carried out on a vertical plane.

### 7.2.2.2 Valve and Bend

The time averaged velocity fields, obtained by the 2D PIV measurements on the valve with upstream bend test case, are shown in Figure 74 and Figure 75 for, respectively, the horizontal $XY$-plane and vertical $XZ$-plane. A very similar stream-wise variation of the mean flow behavior in comparison to the valve alone measurements on a vertical plane, shown in Figure 70, is obtained: the axial $x$-velocity at the centerline increases in the downstream direction while, for the horizontal plane, positive and negative $y$-velocities are noticed for, respectively, negative and positive $y$-values. The magnitude of the time-averaged $y$-velocities decreases towards the downstream direction, indicating the PIV measurement plane is located in the wake of the butterfly valve and no reattachment of the streamlines has occurred. Similar conclusions can be drawn for the vertical plane measurements.
Figure 74: Contour plot of the time-averaged x- (left) and y- (right) velocity for the PIV measurements for the valve with upstream bend, carried out on a horizontal plane.

Similar as for the valve alone measurements, the accuracy of the measurement data which are located at large distances from the center point (i.e. points which are outside the calibrated region) should be interpreted carefully as the results are likely to be influenced by interpolation errors, caused by diffraction of the high speed camera images. As such, the most reliable measurement area extends from -25mm>x>25mm for the stream-direction and from -25mm>y>25mm, respectively, -25mm>z>25mm for the horizontal and vertical direction.

Figure 75: Contour plot of the time-averaged x- (left) and y- (right) velocity for the PIV measurements for the valve with upstream bend, carried out on a vertical plane.

The mean x-velocity along the centerline is shown in Figure 76 and a good agreement of both the magnitude and stream-wise increase of the x-velocity is obtained between the horizontal and vertical plane measurements. As such, the measurement of the centerline velocity can be assumed to be repeatable without much influence of slight variations in the inflow conditions. The shape of the centerline is very similar to the valve alone measurements (shown in Figure 71), with an asymptotic increase of the stream-wise velocity in the downstream direction, indicating the PIV measurement plane is still located in the wake of the valve. The magnitude of the time-averaged x-velocity is, however, slightly smaller which indicates that the presence
of the upstream bend causes an increase in the length of the wake region downstream of the valve. This was also noticed in the discussion of the HWA results in section 7.2.1.2.

Figure 76: Axial velocity along the centerline for the PIV measurements for the valve with upstream bend carried out on a horizontal (blue) and vertical plane (red).

A comparison between the HWA results and PIV data for the horizontal and vertical variation of the time-averaged stream-wise velocity is shown in Figure 77 for, respectively, the horizontal and vertical line, located 1.5D downstream of the valve. Similar as for the valve alone measurements, an excellent agreement between the HWA and PIV results is obtained in the region where the PIV images are calibrated (i.e. $25mm > y > 25mm$ and $-25mm > z > 25mm$, for, respectively, the horizontal and vertical plane. At positions outside this region, the PIV data show decreased velocities in comparison to the HWA results, while the largest flow velocities are occurring in the vicinity of the duct walls. Similar as for the valve alone measurements, this partly explains why the mass flow rate seems to increase in Figure 74 and Figure 75.

Figure 77: Comparison between PIV (red) and HWA (blue) of the time-averaged axial velocity along a horizontal line (left) and vertical line (right), located at $x=1.5D$, for the valve with upstream bend case.

A snapshot of the instantaneous velocity magnitude for both the horizontal and vertical plane is shown in Figure 78. A highly turbulent velocity field is observed with peak velocities going
up to 75m/s. Based on an in-depth analysis of the PIV results, turbulence intensity levels reach values up to 60%-70%, which corresponds to the HWA measurements. As such, also the valve with upstream bend case offers severe challenges for the (un-)steady CFD simulations, carried out in WP4, requiring a proper grid spacing and a compressible implementation.

Figure 78: Contour plot of the instantaneous velocity magnitude with additional vector plot for the PIV measurements for the valve with upstream bend carried out on a horizontal (left) and vertical (right) plane.

7.2.3 Discussion of the acoustic multi-port results
The last type of measurements, carried out for the detailed experimental aeroacoustic analysis of the butterfly valve with inflow distortion is the active acoustic multi-port characterization. In this section the acoustic multiport results of the valve alone measurements and the valve with upstream bend measurements are discussed in detail using both the reflection and transmission coefficients, obtained using the passive scattering matrix determinations, and the active source vector using the previously defined scattering matrix and previously measured reflection matrix of the test rig termination under operational conditions.

7.2.3.1 Valve alone
For the valve alone measurements, an additional multi-port characterization has been carried out for a quiescent medium. This enables, on the one hand, a further validation of the data-acquisition approach and post-processing routines. On the other hand, this analysis provides useful additional validation data for the numerical simulation of the (passive) acoustic propagation through the valve. First of all, it allows to validate the numerical discretization of the geometry without the need of mean flow (RANS) results. Secondly, it enables to evaluate the different numerical post-processing routines to obtain the multi-modal scattering matrix coefficients numerically without adding to much additional complexity caused by the presence of the mean flow. Finally, a comparison between the different numerical flow-acoustic propagation methods, envisaged within this project, can be made and their performance to model the influence of the presence of a highly non-uniform flow on the passive acoustic scattering coefficient can be easily evaluated.
The magnitude of the different transmission and reflection scattering matrix coefficients are shown in Figure 79, for the different modes which are excited within the frequency range under consideration. At low frequencies (up to, approximately, 1500 Hz) in the plane wave region the result shows an excellent agreement with an analytical 1D model of a duct system with a sudden area contraction-expansion with an infinitesimally small length (i.e. similar to a diaphragm) with a contraction ratio equal to $1 - \cos \alpha = 0.134$, $\alpha$ being the valve angle, equal to $30^\circ$. In this case, for the low frequency region the transmission coefficient linearly decreases with increasing frequency while the reflection coefficient linearly increases. The slope of the linear decrease and increase of, respectively, the transmission and reflection coefficient is determined by the contraction ratio of the obstacle, in this case the butterfly valve. This phenomenon is clearly predicted by the valve alone without flow measurements.

At higher frequencies, but still in the plane wave region, the assumption of an infinitesimally small valve length does not longer hold since the acoustic wavelength becomes of the same order of magnitude as the length of the valve body. In this region, a sudden decrease and afterwards increase of the transmission coefficient is noticed and an opposite phenomenon of the reflection coefficient is observed. Above the cut-on frequency of the first circumferential mode and its complex conjugate a slight decrease of the transmission coefficient is noticeable with increasing frequency.

As no flow is present, a fairly reciprocal behavior (i.e. a same behavior of the upstream and downstream propagating modes) is expected and present in the different reflection and transmission coefficients, shown in Figure 79. Also both the (-1,0) and (1,0) show an almost identical behavior, as expected. As such, it can be concluded that enough confidence is guaranteed for the accuracy of these data to validate the different numerical prediction techniques for the valve alone case in the absence of a flow field.
Figure 79: Reflection and transmission scatter matrix coefficients of the valve alone measurements without flow for the, from top to bottom, (0,0)-upstream mode, (1,0)-upstream mode, (-1,0)-upstream mode, (0,0)-downstream mode, (1,0)-downstream mode and the (-1,0)-downstream mode.

The transmission and the reflection scattering matrix coefficients for the multi-port characterization with flow are shown in Figure 80. In comparison to the no flow results, the results are much ‘noisier’. This is due to the influence of the aerodynamic pressure fluctuations, caused by the high turbulence intensity levels, which, especially at the downstream microphone array, fully masked the acoustic signals. The coherence levels of the microphone signals with respect to the loudspeaker excitation signal, indeed significantly drop to values below 0.2. This phenomenon is more apparent for the transmission and reflection coefficient of the upstream propagating modes. Nevertheless, the scattering matrix still shows a realistic prediction of the reflection and transmission behavior of the valve. As
such, the capability of the multi-modal decomposition technique, to discard and filter out non-acoustic microphone signals is evidenced.

![Reflection and transmission scatter matrix coefficients](image)

**Figure 80:** Reflection and transmission scatter matrix coefficients of the valve alone measurements with flow for the, from top to bottom, (0,0)-upstream mode, (1,0)-upstream mode, (-1,0)-upstream mode, (0,0)-downstream mode, (1,0)-downstream mode and the (-1,0)-downstream mode.

The scattering matrix results for the valve alone with flow show that the presence of the flow field has a significant influence on the passive transmission and reflection characteristics of the valve. At low frequencies, below 1700 Hz, the transmission coefficient remains almost constant with, due to convective amplification and dissipation, a slightly lower value of the downstream propagating mode in comparison to the upstream propagating mode. Similar as for the no flow results a sudden increase in the transmission coefficient is noticed close to the
cut-on frequency of the first circumferential modes. At frequencies, larger than the cut-on frequency a similar transmission behavior as for the no flow case is observed with a slight decrease of the transmission coefficient for increasing frequency. The outcome of the various numerical flow-acoustic prediction techniques, which will be used in WP4 to simulate this problem, should provide further information on the accuracy of the obtained experimental results.

The magnitude of the active source vector coefficients is shown in Figure 81. Based on these results, it can be concluded that most of the active noise, generated aerodynamically by the turbulent flow over the valve is radiated towards the downstream direction, while the noise levels, propagated upstream, are significantly (i.e. more than 20dB) smaller. This might be caused by the fact that the major noise generating mechanisms are being generated downstream of the valve and that the valve itself provides an efficient upstream shielding effect for the upstream propagates, i.e. most of the aerodynamically generated sound waves are reflected by the valve into the downstream direction.

The aerodynamically generated acoustic field, propagating in the downstream direction, shows a very broadband spectrum with very high acoustic levels, close to 120 dB. This was also experienced qualitatively during the experimental campaign where the stepped sine loudspeaker excitation, generating a signal of, approximately, 140 dB at the microphone positions was barely audible inside the semi-anechoic room. For the upstream propagating source field some tonal phenomena at 1000 Hz and 1700 Hz are observed for the (0,0) mode. A comparison with the numerical CAA simulations is, however, needed to verify whether this phenomenon is actually occurring or is the result of measurement inaccuracies at these frequencies.

For the active noise generating components of the first circumferential mode and its complex conjugate, a gradual decrease of the sound pressure levels with increasing frequency is observed. For these modes, also a broadband active noise spectrum is observed for both the upstream and downstream propagation with a preferred radiation towards the downstream direction.
Figure 81: Magnitude (dB) of the active source vector coefficients for the valve alone measurements with flow for the, from top to bottom, (0,0) mode, (1,0) mode and (-1,0) mode.

7.2.3.2 Valve and Bend

The magnitudes of the passive transmission and reflection scattering matrix coefficients for the valve with upstream bend measurements are shown in Figure 82. Both the magnitude and frequency-dependent behavior of the transmission and reflection coefficients is very similar to the valve alone with flow measurements, indicating the presence of the bend does not significantly change the acoustic transmission characteristics of the system. This can be explained by the fact that the valve causes the largest blockage of the acoustic field while the bend causes only minor changes in the reflection and transmission characteristics. Given the large similarity between the valve and bend measurements and the valve alone measurements with flow, this might indicate that the presence of upstream distortions and the accompanying change in both the upstream and downstream time-averaged velocity field, evidenced by the analysis of the HWA and PIV results does not significantly change the propagation characteristics. This will be verified in WP4 by comparing the numerical flow-acoustic propagation results obtained for both the valve alone and valve with upstream bend measurements with the experimental results, discussed in this deliverable.
Figure 82: Reflection and transmission scatter matrix coefficients of the valve with upstream bend measurements with flow for the, from top to bottom, (0,0)-upstream mode, (1,0)-upstream mode, (-1,0)-upstream mode, (0,0)-downstream mode, (1,0)-downstream mode and the (-1,0)-downstream mode.

The magnitude of the active source vector coefficients for both the upstream and downstream propagating modes is shown in Figure 83 for the valve with upstream bend measurements. Similar as for the valve alone measurements the major radiation occurs in the downstream direction and in this direction the aerodynamically generated sound field has a very broadband nature. At low frequencies, below the cut-on frequency of the first circumferential mode the results of the active source vector coefficients are very similar, both in amplitude and frequency-dependent behavior in comparison to the valve alone measurement which might be an indication for the fact that the presence of the upstream bend and the corresponding
increased incoming turbulence levels do not change the low-frequency aerodynamic noise generating mechanism.

Above the cut-on frequency of the circumferential modes, however, the active source components are modified. This is mainly noticeable in the active source vector components which contribute to the active source spectrum of the downstream propagating circumferential modes (1,0) and (-1,0). In comparison to the valve alone measurements, the active source vector causes an increased aerodynamic excitation of these modes and the decay of these components with increasing frequency is less pronounced. This might indicate that the presence of the upstream bend and the accompanying change of the turbulent field, leading to a significant increase of the turbulent intensities, measured downstream of the valve, is predominantly responsible for a different high-frequency content of the aerodynamically generated sound field.

![Figure 83: Magnitude (dB) of the active source vector coefficients for the valve with upstream bend measurements with flow for the, from top to bottom, (0,0) mode, (1,0) mode and (-1,0) mode.](image)
8 Ranking of ramp noise sources

Within WP2 also acoustic measurements, using both far-field microphones and acoustic beamforming techniques, are employed around one aircraft in order to identify and rank the dominant sources affecting ramp noise. These measurements are carried out by EMBRAER. As agreed with the different consortium partners, the results obtained from this analysis are considered to be confidential. As such, they described in full-depth in the restrictive deliverables D2.3 and D2.7, but are omitted in this deliverable which is publicly accessible.
9 Conclusions

In this deliverable the results of WP2: “Experimental characterization of flow-acoustic and multi-component interactions” are summarized. On the one hand, the dominant sources affecting ramp noise have been identified by EMB and ranked in a general manner through regular far-field microphone and acoustics beamforming measurements. On the other hand an in-depth measurement campaign is successfully carried out to determine the aerodynamic and aeroacoustic behavior of a simplified but realistic fan and butterfly valve ECS mock-up with specific focus on the influence of upstream and downstream obstacles on the aerodynamic performance and the aerodynamic noise generation characteristics and its radiation inside a flow duct system. Both the butterfly valve and the fan are provided by LTS, thus ensuring the engineering relevance of the proposed test cases.

In order to study the fan and butterfly ECS mock-ups experimentally, 2 dedicated test-facilities are designed, assembled and thoroughly validated by KUL and KTH in collaboration with KTH. The operational flow conditions and the geometrical configuration for both set-ups have been determined using a preliminary aerodynamic and acoustic survey on a large number of operational conditions. The final selection of the operating point is made based on the engineering relevance of the acquired far-field acoustic spectra; as well as on the feasibility to numerically predict in WP4 the aeroacoustic behavior of these configurations with the numerical prediction techniques, developed in WP3.

Consequently, detailed flow and acoustic measurements are successfully carried out and validated using Hot Wire Anemometry (HWA); Particle Image Velocimetry (PIV), and active acoustic multi-port characterization techniques for a fan with upstream distortions, a fan with downstream distortion and a valve with inflow distortion. As a result, further insight is gained on the influence of downstream and upstream distortion on the aeroacoustic behavior of both the fan and butterfly valve and a large experimental flow and acoustic database, made available for the consortium partners, is obtained.

The outcome of this measurement campaign will be used in WP4 to validate the different numerical prediction techniques to simulate the aeroacoustic noise behavior of these devices including an assessment of the multi-component interactions:

- HWA data obtained upstream of the different test objects will be used to generate proper inflow boundary conditions for the (un-)steady flow simulations.
- HWA and PIV results will be used to validate the outcome and accuracy of the (un-) steady flow simulations.
- The scattering matrix obtained with the acoustic multi-port characterization will be used to evaluate the outcome of the different (convective) acoustic propagation simulation techniques.
- The active components of the acoustic multi-port characterization will be, finally, used to validate the different hybrid CAA (Computational AeroAcoustics) simulations.
10 References


