

# APPLICATION OF ANALYTICAL NOISE MODELS USING NUMERICAL AND EXPERIMENTAL FAN DATA

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## ABSTRACT

**In this study, both tonal and broadband analytical noise models based on low computational costs simulations as well as experimental data are applied and validated. Various inflow conditions are considered to verify the ability of the tonal noise model to reproduce the tendencies between the different cases. The tones predictions are in very good agreement when using the numerical input data. For the prediction based on the experimental data the reduction of the tones with the flow control devices is well captured and the frequency-dependent variation of the tones amplitude is reproduced. The broadband noise is well predicted by a trailing-edge noise model over a wide frequency range and confirms that the trailing edge noise is the main broadband noise source mechanism in case of isolated axial fans.**

## INTRODUCTION

Due to strict governmental restrictions and the increasing consideration of comfort factors the acoustic performance of fans, whose usage is subject of our everyday life, becomes more and more important. Although numerical methods are widely used in the design process of fans, the calculation of the acoustics of fans is still a challenging issue. Since the computational costs for direct accurate acoustic simulations are too high for industrial applications, the noise emission predictions are often based on empirical models and iterative, lengthy experimental measurements. In the present work, analytical models of noise specific mechanisms will be investigated for a ducted low-speed axial fan. Such models will require temporal averaged input data that can be obtained by unsteady Reynolds Averaged Navier-Stokes (URANS) simulations.

In case of isolated ducted axial fans, the major annoyances are the perceived overall noise level and its tonal content (subjective noise). The latter is generated by the fluctuating blade forces induced by an inlet distortion, and radiated at the blade passing frequency (BPF) and its harmonics. Large flow structures sucked into the fan section, can be accelerated and amplified near the blade leading edge and create a non-axisymmetric and time-varying incident velocity field to the rotor blade (Sturm and Carolus, 2013a). The large flow structures are related to any flow asymmetry and installation effects. Moreover, they may be unsteady and responsible for unsteadiness of the sound levels at the BPF (Sturm and Carolus, 2014).

In the present work, the inlet distortion generation mechanism and the associated tonal noise will be experimentally investigated on a 5-blade low-speed radial fan installed in a cylindrical duct where a controlled pressure-rise is applied. The bell mouth ended-duct open in an anechoic wind-tunnel. The flow conditions inside the anechoic room yields a massive and slow flow recirculation that is responsible for the inlet distortion. The tonal noise related to the upstream distortion can be strongly reduced using flow conditioners. Hot wire measurements of the inflow are conducted in a plane one diameter upstream the fan for three inflow conditions investigated. Unsteady SAS simulations are performed to complement the experimental database and provide input for an analytical software.

Two inflow boundary conditions are investigated: a case of pure axial inflow and the free inflow case of the experiments is also achieved numerically by using the hot wire data.

The tonal content of the fan noise spectra is investigated using an analytical prediction tool OPTIBRUI that uses the analytical response of Amiet (1976b) for stacked flat plates to discretize the rotor blade geometry and the acoustic power upstream of the fan is obtained by using Goldstein's acoustic analogy (Goldstein, 1976). Several cases will be considered for the analytical tonal model: the hot-wire measurements for the different inlet flow conditions and several extraction planes from the two numerical simulations, in particular the amplification effects of the distortion is investigated. The broadband component of the fan is thought to be the self-noise mechanism and is not affected by the flow conditioner (Sturm and Carolus, 2013b). Consequently, the model based on the generalized Amiet's model (Roger and Moreau, 2005; Rozenberg et al., 2010) with pressure fluctuations reconstructed according to Rozenberg's model (Rozenberg et al., 2012) will be applied only on the results of the simulation with the disturbed inflow to predict the broadband noise of the fan.

The paper is organized as follows: in the methodology section the experimental configuration and the hot-wire measurements are first described, then the numerical parameters of the simulations and the analytical models and their specific inputs are finally presented. In the results section, the extraction from the numerical simulations is first analyzed, then the tonal prediction based on the numerical and experimental data are compared with the experimental sound power levels. Finally the trailing-edge noise predictions are compared with the acoustic measurements.

## METHODOLOGY

### Experimental Configuration

In this study the standardized test rig for acoustic measurements of fans at the University of Siegen is used. This test rig has been used in several studies (Sturm and Carolus, 2013a,b), and hence only the main features are repeated here. The fan takes air from a large anechoic room and exhausts into a duct with an anechoic termination. The flow inlet is located off-center in the reverberant floor. The low pressure fan unit contains five cambered and swept blades which are designed with an in-house design software for axial fans. The rotor is manufactured and balanced with very high precision to avoid any non-aeroacoustic sound sources. A bell mouth type inlet nozzle with a  $1/4$  rotor diameter radius is employed. Thin supporting struts are positioned one rotor diameter downstream of the rotor. No other obstructions are present.

The acoustic measurements are conducted by using three microphones (Brüel & Kjaer type 4190) with a radial distance of 1.3m from the leading edge of the rotor and an angular spacing of  $35^\circ$  from the axis of rotation. The signals are recorded with a sampling frequency of  $f_s = 25.6\text{kHz}$  for an acquisition time of 10 s. The evaluated spectra are an average of the three microphones and have a frequency resolution of  $\Delta f = 1\text{Hz}$ . For all levels, the reference pressure is  $p_0 = 2 \times 10^{-5}\text{ Pa}$ . All measurements were performed at a rotational speed of  $n = 3000\text{ rev/min}$  and at the fan design operating flow rate coefficient  $\phi_{design} = 0.195$ .

The application of two inflow conditioners as shown in Fig. 1, allows the investigation of the tonal noise related to the inflow distortion. The Hemispherical Flow Conditioner (HFC) consists of a combination of a fluid conditioner structure, which reduces the lateral turbulence, and a downstream layer of wire mesh to reduce axial disturbances (Scheimann and Brooks, 1980). A second inflow conditioner is a plane layer of small tubes, inserted into the duct upstream of the fan (Tubular Flow Conditioner, TFC). The axial position of the TFC with respect to the rotor is shown in Fig. 2. Geometric details of the flow conditioners have been described by Sturm and Carolus (2013b).

To capture spatial variations of the flow field in the intake a cross section of the duct one fan diameter upstream of the leading edge is scanned by a 3D hot-wire probe. The anemometer operates in a constant-temperature mode. The measured signals are temperature corrected by using a temperature probe. The probe is calibrated in a low turbulence wind tunnel.

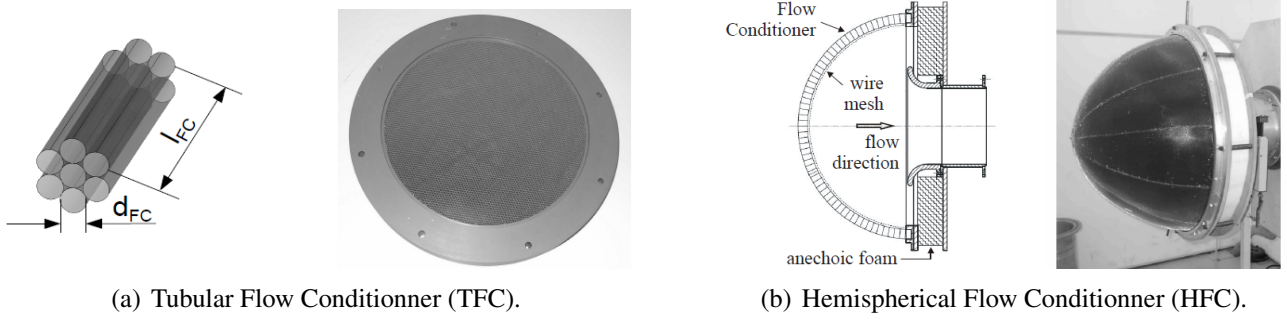


Figure 1: Inflow conditioners.

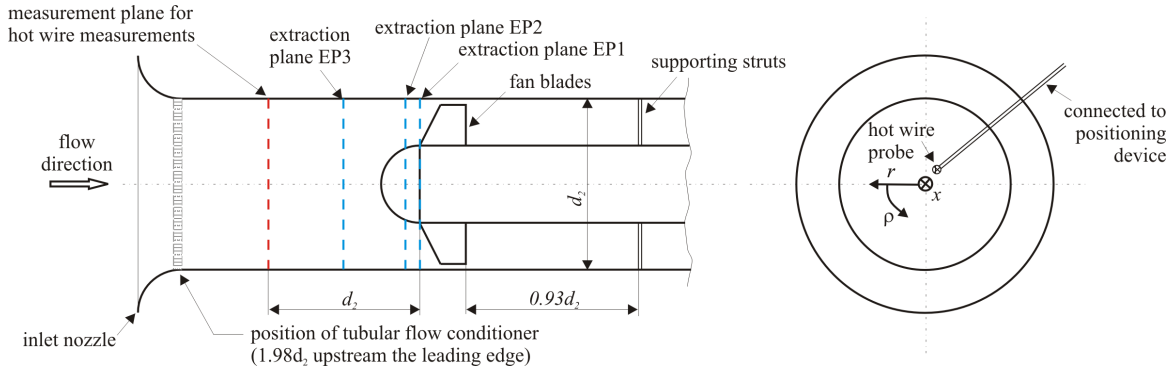


Figure 2: Lateral view of the axial fan section (left) and top view (right).

The orientation of the polar coordinate system for the hot wire measurements is indicated in Fig. 2 (right). The probe is attached to an automatic two axis positioning system, which is able to move the probe in circumferential and radial direction without disturbing the flow inside the duct. The device has a positioning accuracy of 0.02 mm and  $0.15^\circ$ . The uniformly arranged measurement points in the cross sectional plane of the duct have an angular distance of  $\Delta\varphi = 5^\circ$  and radial distance of  $\Delta r = 5$  mm, respectively. Together with the center of the plane, this leads to 2017 measurement points. All measurements are conducted over a time interval of two seconds with a sampling rate of  $f_s = 25.6$  kHz.

The inflow velocity profiles will be evaluated in terms of the axial velocity  $c_m$ , the circumferential velocity  $c_u$  and the radial velocity  $c_r$ , according to the coordinate system indicated in Fig. 2. Due to the fact that the data of the hot wire measurements are not recorded simultaneously at the different measurement points, only time averaged data will be evaluated assuming a statistically stationary state and used as a stationary inlet boundary condition for the simulations.

### Numerical Simulations

For the numerical simulations the fan section, as depicted in Fig. 2, is modeled using the commercial meshing tool ANSYS IcemCFD. As shown in Fig. 3, the computational domain includes the ducted fan with the hub and is divided into three subdomains, where the rotating rotor domain is separated by sliding mesh interfaces from the stationary subdomains. The inlet is located one fan diameter upstream of the leading edge, while the outlet is placed two diameter downstream the trailing edge. The small tip gap of  $s/d_2 = 0.1\%$  is fully resolved by the numerical mesh. The unstructured mesh consists of 21.4 million cells including 5 prism layers with a 10% growth rate at walls and refined regions in the vicinity of the blades.

The pressure outlet boundary condition is set to a gauge pressure of 0 Pa. Two cases, which differ in the inlet boundary condition are investigated: (i) a pure axial, undisturbed inflow achieved by a

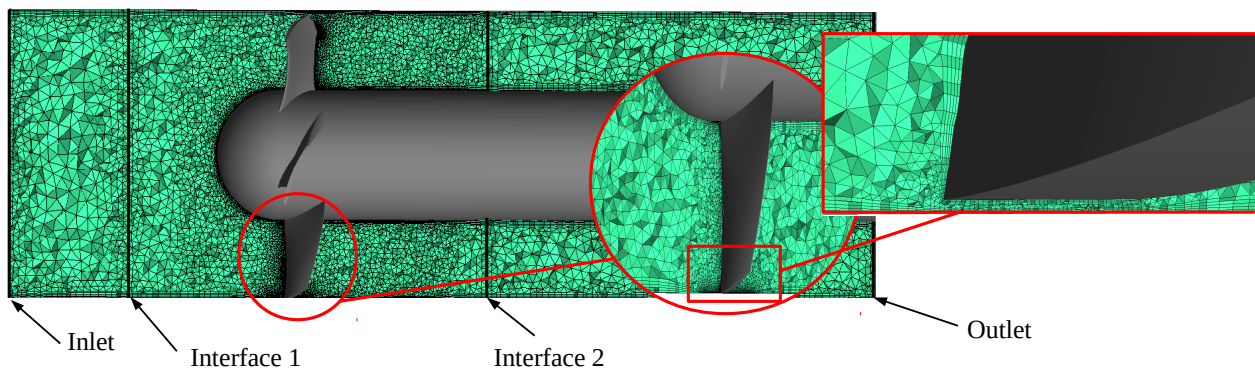


Figure 3: Computational domain and numerical mesh of the simulated fan section.

constant inlet velocity according to the design flow rate and (ii) a disturbed inflow where the velocity data of the hot wire measurements is used for the inlet boundary condition. While the latter are the real inflow conditions, the former represents an ideal inflow condition which can be achieved only numerically.

The simulations are conducted using the commercial solver ANSYS Fluent 14.5.7. Although time averaged simulation results are sufficient for the application of the presented noise models, the SAS approach is chosen to obtain instantaneous data which can be compared with intermediate results of the acoustic models like the spectra of the blade pressure fluctuations. All quantities are solved using second order schemes. The SIMPLEC algorithm is used to solve the pressure-velocity coupling. The time step is  $5 \times 10^{-6}$ s, which leads to a resolution of 4000 time steps per blade-passage. The input data for the tonal noise model is extracted at three different planes shown in Fig. 2: extraction plane EP1 at the leading edge, extraction plane EP2 one quarter chord length upstream of the leading edge and extraction plane EP3 midway between the leading edge and the interface upstream of the rotor.

### Tonal and Broadband Noise Models

This section presents the tonal and broadband analytical models used for the acoustic predictions and comparison with experimental measurements. These models are integrated in the OPTIBRUI software for noise prediction of axial turbomachine developed at Université de Sherbrooke. The geometrical modeling of the fan is identical for the two noise mechanisms. The fan is mounted in an annular duct of constant section. The cylindrical reference frame  $\mathcal{R}_d(r_d, \theta_d, z_d)$  is fixed to the duct with its axial direction corresponding to the machine axis oriented towards the exhaust of the duct. In order to account for the complex blade geometry, the blade is split into several annular strips of identical span  $dr$ . At each strip  $r_d$ , the blade element is described as a flat plate of constant chord  $C(r_d)$  and stagger angle  $\chi(r_d)$ . The radial stacking of the blade elements is parametrized by the lean and sweep angles  $(\varphi_{LE}(r_d), \psi_{LE}(r_d))$  that define the positioning of the blade element leading-edge with respect to a rotational and a meridional plane crossing the blade leading-edge at the hub. Similar angles at the trailing-edge  $(\varphi_{TE}(r_d), \psi_{TE}(r_d))$  can be deduced from the chord and stagger parameters. Each annular strip can be unwrapped into a rectilinear cascade geometry. A local Cartesian reference frame  $\mathcal{R}_c(x_c, y_c, z_c)$  is defined to compute the blade element acoustic response.

#### Tonal noise model with in-duct propagation

The rotor is assumed to be located in an infinite annular duct with a constant section and rigid walls. For the acoustic propagation, only an inviscid mean axial flow of Mach number  $M_a$  is considered. Within these assumptions, the Goldstein's analogy (Goldstein, 1976) provides the acoustic pressure in the duct resulting from the force  $\vec{f}$  exerted by the blade surface  $S$  on the fluid using the

annular duct Green's function  $G$ :

$$p(\vec{x}, t) = \int_{-T}^T \iint_S \frac{\partial G(\vec{x}, t | \vec{x}_0, t_0)}{\partial x_{0,i}} f_i(\vec{x}_0, t_0) dS(\vec{x}_0) dt_0 \quad (1)$$

with  $t_0$  and  $t$  the emission and observer times and  $\vec{x}_0$  and  $\vec{x}$  the emission and observer positions respectively in the duct coordinate system.

The Green's function can be written on the duct mode basis using the indices  $n$  and  $j$  for the circumferential and radial modes respectively in the frequency space:

$$G(\vec{x}, t | \vec{x}_0, t_0) = \frac{\Im}{4\pi} \sum_{n=-\infty}^{+\infty} \sum_{j=0}^{+\infty} \frac{E_{nj}(r_0) E_{nj}(r) e^{\Im n(\theta - \theta_0)}}{\Gamma_{nj}} \int_{-\infty}^{\infty} \frac{e^{-\Im(\gamma_{nj}^{\pm}(x-x_0) + \omega(t-t_0))}}{\kappa_{nj}} d\omega \quad (2)$$

with  $k_0 = \omega/c_0$ ,  $E_{nj}$  the duct radial function depending on the eigenvalue  $\chi_{nj}$  of norm  $\Gamma_{nj}/2\pi$ ,  $\gamma_{nj}^{\pm} = \frac{M_a k_0 \pm \kappa_{nj}}{\beta^2}$  the axial acoustic wavenumber and  $\kappa_{nj}^2 = k_0^2 - \beta^2 \chi_{nj}^2$  the cut-off criteria. The superscript  $\pm$  is related to the direction of propagation for  $z > z_0$  and  $z < z_0$  respectively.

The force can be related to the pressure jump across the blade:  $\vec{f} = \Delta P \vec{n}$  with  $\vec{n} = \text{sign}(\Omega) \vec{e}_{y,c}$  the local normal of the blade. The pressure jump is assumed to be uncorrelated between blades and can be decomposed into a Fourier series to account for the rotational periodicity. The Fourier coefficient of the  $p$  harmonics is written  $\Delta P(\vec{x}_0, t_0)$ .

A change of reference frame is required to exchange the integration order by integrating over a fixed surface. For  $T \rightarrow \infty$  and uniformly distributed blades, the acoustic pressure becomes the summation of tones. The acoustic pressure at the harmonic frequency  $sB\Omega$ :

$$\hat{p}^{\pm}(r_d, \theta_d, z_d, t) = \sum_{n=-\infty}^{+\infty} \sum_{j=0}^{+\infty} E_{nj}(r) e^{\Im(n\theta - \gamma_{nj}^{\pm} z_d)} P_{nj,sB-n}^{\pm} e^{-\Im sB\Omega t} \quad (3)$$

The modal coefficient can be computed by splitting the surface integration first along the circumferential extent of the blade then along its span:

$$P_{nj,sB-n}^{\pm} = \frac{-\text{sign}(\Omega) B}{2\pi \Gamma_{nj} \kappa_{nj}} \int_{RH}^{RT} \left\{ -\Im Q_{23} \frac{\partial}{\partial r_0} - \frac{n}{r_0} Q_{22} + \gamma_{nj}^{\pm} Q_{21} \right\} E_{nj}(r_0) \times e^{\Im(\gamma_{nj}^{\pm} z_{d,LE}(r_0) - n\theta_{d,LE}(r_0))} I_{nj,sB-n}^{\pm} dr_0 \quad (4)$$

with  $(x_{d,LE}, y_{d,LE}, z_{d,LE})$  the coordinates of the leading-edge and  $I_{nj}^{\pm}$  the chordwise integral of the pressure jump. The hub-to-tip spanwise integration in Eq. (4) is replaced by a discrete summation over the annular strips of the blade. The chordwise integral  $I_{nj}^{\pm}$  is approximated assuming a chordwise wavenumber without sweep or lean effects. Following the acoustic flat plate response of Patterson and Amiet (1976), the pressure jump chordwise integral can be computed analytically and related to the harmonics of the incident upwash velocity gust  $\hat{w}_p$  on the leading-edge considering:

$$I_{nj,sB-n}^{\pm} = \frac{C}{2r_0} \int_0^2 \widehat{\Delta P}_{sB-n}(x_c^*) e^{\Im \frac{c}{2} \left( \frac{n}{r_0} \sin \chi + \gamma_{nj}^{\pm} \cos \chi \right) x_c^*} dx_c^* = 2\pi \rho_0 W \hat{w}_{sB-n} (\mathcal{L}_{nj}^1 + \mathcal{L}_{nj}^2) \quad (5)$$

with  $W$  the relative velocity and  $\mathcal{L}_{nj}^1$  and  $\mathcal{L}_{nj}^2$  the blade response and the back-scattering of the trailing-edge respectively for supercritical gusts defined in (Reboul, 2010).

Finally the acoustic power is obtained by considering only the cut-on modes that fulfill  $\kappa_{nj}^2 > 0$  (Meyer and Envia, 1996):

$$\Pi_s^{\pm} = \sum_{n=-\infty}^{+\infty} \sum_{j=-\infty}^{+\infty} \frac{\Gamma_{nj} \beta^4 k_{0,s} \kappa_{nj}}{\rho_0 c_0 (k_{0,s} \pm M_a \kappa_{nj})^2} |P_{nj,sB-n}^{\pm}|^2 \quad (6)$$

### Broadband noise with free-field propagation

For the case of broadband self-noise model, the acoustic propagation is computed in free-field following the extension of Amiet's theory (Roger and Moreau, 2005) applied for each uncorrelated blade strip (Moreau and Roger, 2007; Guedel et al., 2009). The radiated sound field is calculated by integrating the induced surface sources on the actual chord length,  $C$  and the strip span,  $dr$ , assuming convection of frozen turbulent boundary-layer eddies past the trailing edge. In the assumption of large aspect ratio  $dr/C$ , the spectral density of the sound pressure in the far field is obtained using:

$$S_{pp}(\vec{x}_c, \omega) = \left( \frac{\omega C y_c}{2\pi c_0 S_0^2} \right)^2 \frac{dr}{2} \left| \mathcal{L} \left( \frac{\omega \bar{k} z_c}{U_c S_0} \right) \right|^2 \Phi_{pp}(\omega) l_z(\omega) \quad (7)$$

where  $\mathcal{L}$  is the aeroacoustic transfer function, derived analytically (Amiet, 1976a), and taking into account leading-edge back-scattering, supercritical and subcritical gusts (Roger and Moreau, 2005).

$\Phi_{pp}(\omega)$  is the wall-pressure spectrum upstream of the trailing-edge that has to be reconstructed from the RANS or URANS simulation or directly extracted in the case of the SAS simulations while  $l_z(\omega)$  is the corresponding spanwise correlation length described by Corcos' model.

In case of rotation, the far-field sound pressure level of a low solidity fan with  $B$  uncorrelated blades is given by an integration over all possible circumferential positions of the single airfoil formulation Eq. (7) (Paterson and Amiet, 1979):

$$S_{pp}(\vec{x}_d, \omega) = \frac{B}{2\pi} \int_0^{2\pi} \left( \frac{\omega_e(\Psi)}{\omega} \right)^2 S_{pp}^\Psi(\vec{x}_c, \omega_e) d\Psi. \quad (8)$$

The factor  $\omega_e(\Psi)/\omega$  accounts for Doppler effects due to the rotation (Sinayoko et al., 2012).

The wall-pressure spectrum upstream of the trailing-edge is obtained by using Rozenberg's spectral model (Rozenberg et al., 2010, 2012) which is developed for attached turbulent boundary layer with adverse pressure gradient and accounts for Reynolds-number effects. This model is based on several characteristic parameters of the boundary layer. Velocity profiles in the boundary layers of the pressure side and the suction sides at each strip are extracted near the trailing-edge from a phase-averaged snapshot (over a fan revolution) of the SAS simulation. The total pressure allows defining the boundary layer thickness  $\delta_{99}$ . The outer velocity  $U_e$ , the boundary layer thickness, the displacement thickness  $\delta^*$  and the wall shear stress  $\tau_w$  are averaged over the 5 blades. Similarly the wall pressure is extracted along the blade surface at each strip to compute the pressure gradient  $\frac{dp}{ds}$  near the trailing-edge that is averaged over the 5 blades.

## RESULTS

### **Parametric Study of the excitation of the fan**

The main input of the tonal noise mechanism is the relative velocity distortion seen by the leading edge. Along the simulation, the instantaneous velocity is extracted in three planes shown in Fig. 2. As can be observed in Fig. 4(a), the instantaneous velocity field is strongly influenced by the rotor potential field, even in the plane EP2 or EP3. Such velocity would generate high and unphysical harmonic at the blade rotation harmonic. To avoid this spurious effect, the potential effect of the rotor must be removed. The velocity is then averaged in the duct reference frame. A preliminary study has shown that 20 instantaneous fields to cover each blade passage period were sufficient to successfully remove the potential effect. The averaged field over a blade passage for the disturbed simulation is clearly non-axisymmetric as shown in Fig. 4(b).

Seven blade passages have been extracted once the blade force components have reached a stationary evolution. The Fourier coefficients of the upwash components of the relative velocity at mid span of the blade in the EP2 extraction plane are given in Fig. 5 for the disturbed and undisturbed simulations. The coefficient amplitudes are 10 times higher in the disturbed simulation than in the

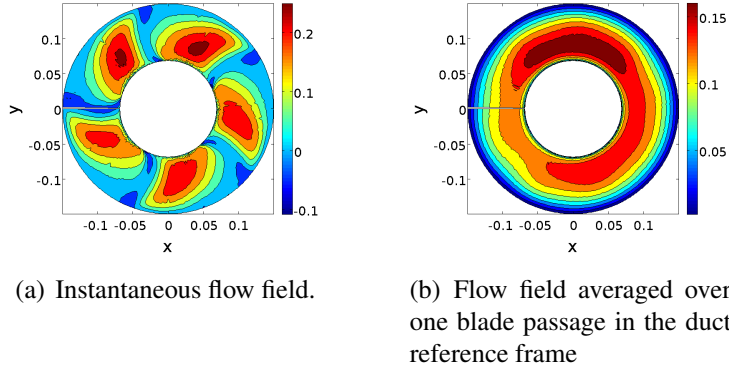


Figure 4: Dimensionless azimuthal component of the absolute velocity in the EP1 plane for the disturbed case.

undisturbed one. Also, some variations can be observed between the successive blade passage periods of averaging, but the fluctuations are higher in the disturbed case than in the undisturbed case.

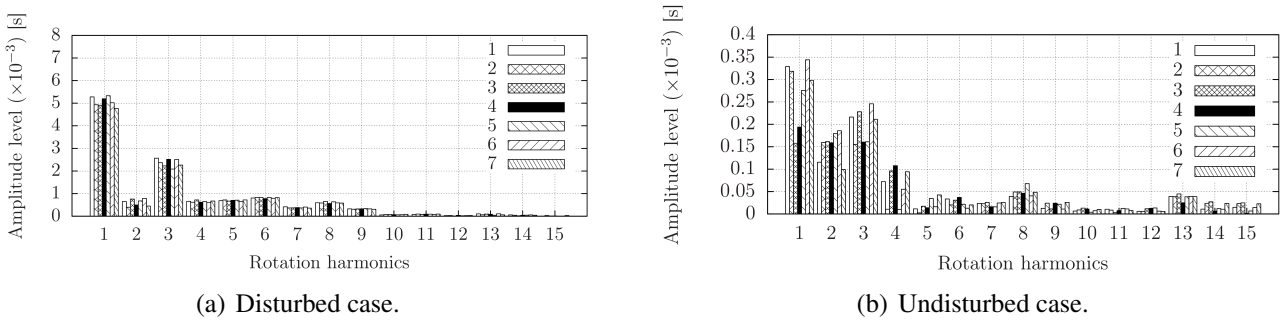


Figure 5: Fourier coefficients of the upwash component of the relative velocity at the mid-span of the blade in EP2. Seven successive blade passage periods are shown with different patterns.

### Tonal Noise Prediction based on Numerical Simulation

The noise predictions computed for the first blade passage period distortion are compared with the experimental measurements in Fig. 6. The predictions obtained from the three extraction planes are in very good agreement with the tones measured in the experiments for both the disturbed and undisturbed case. In particular the level of the BPF tone reduction of 10 dB is similar to the experimental reduction obtained with the HFC inflow device. The extractions made in EP3 upstream of the annular section provide the higher tone levels. The initial disturbances observed in EP3 are diminished as they are propagated downstream. This could be due to either numerical dissipation or related to the extra distortion caused by the bulk-head that is not present in EP3.

The tonal noise prediction based on the seven successive blade passage periods that have been extracted are shown in Fig. 8. As observed in the experiments the harmonics levels are fluctuating from one blade passage period to another. The levels of fluctuations are increasing in extraction plane closer to the blade leading-edge. The captured fluctuations might be related to secondary flow and strong unsteadiness captured close to fan blade by the SAS turbulent model.

### Tonal Noise Prediction based on Hot-Wire measurements

The hot-wire measurements recorded one duct diameter upstream of the fan for the three inflow conditions can also be used as distortion input in the analytical model. To account for the duct contraction yielding flow acceleration and pre-swirl effects an amplification factor is computed for



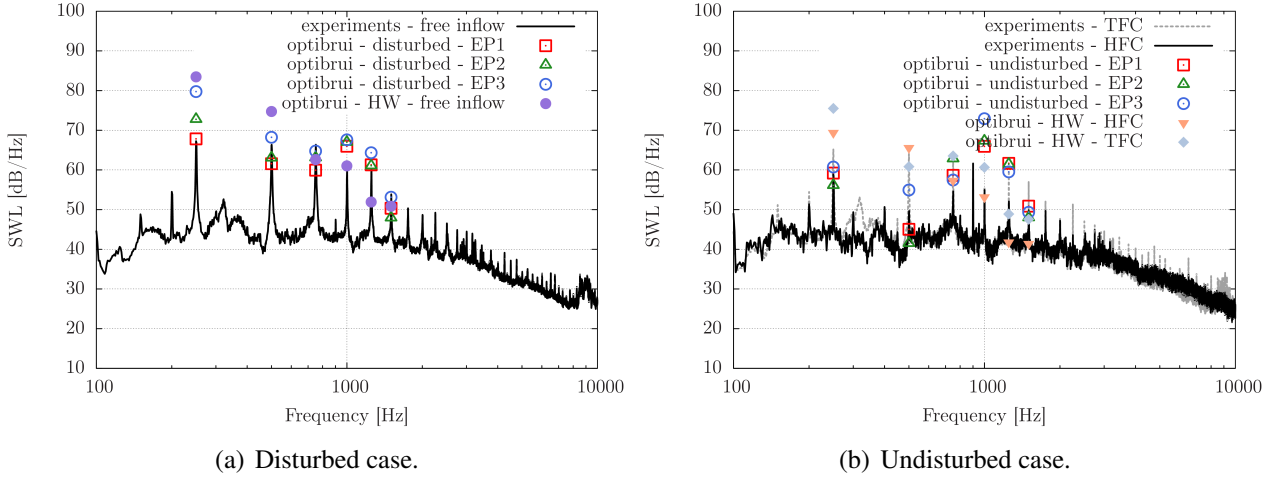


Figure 6: Comparison of the acoustic prediction from the tonal model based on numerical simulations with experimental measurements upstream of the duct intake.

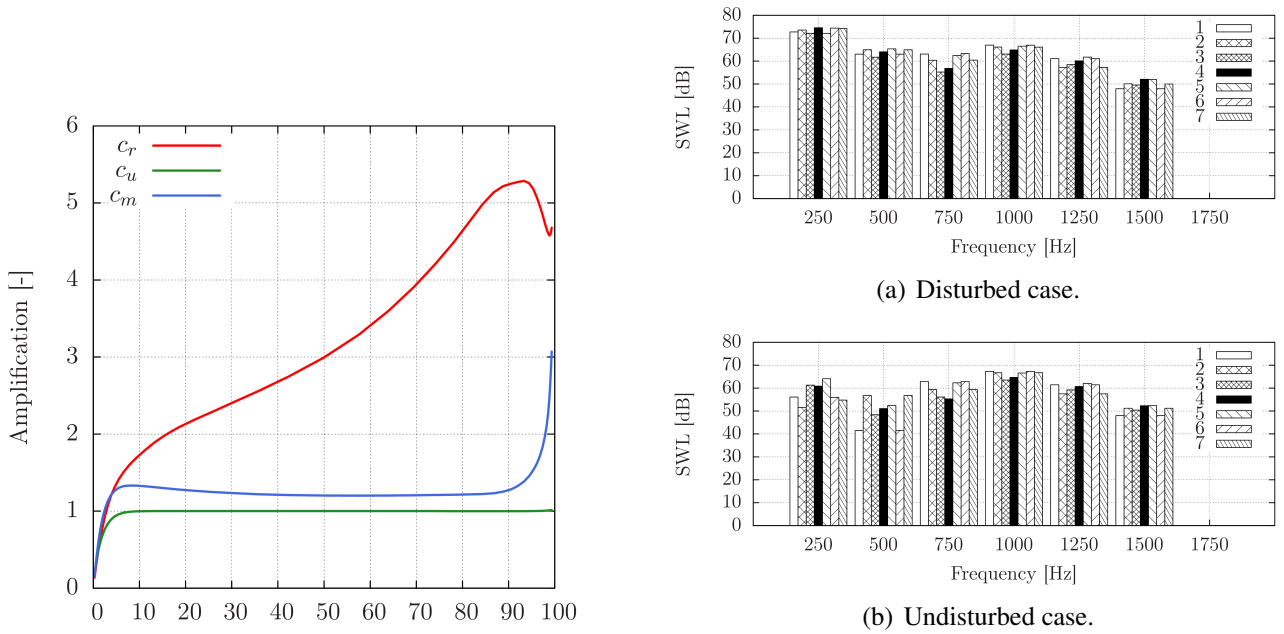


Figure 7: Amplification factor for the axial and circumferential velocity component.

Figure 8: Acoustic predictions from the tonal noise model for seven successive blade passage velocity fields extracted at the EP2 location.

each relative velocity components between the inlet plane and the EP1 plane. The velocity is averaged in the duct reference frame over a fan rotation to remove any potential effects that could affect the calculation of the amplification. The calculation considers only the annular area between the hub and the shroud and is an azimuthal average. The radial, azimuthal and axial components of the amplification factor is shown in Fig. 7. A smoothing of the hot-wire measurements was required to avoid unphysical high tones related to point to point variations in the measurements. The acoustic predictions are shown in Fig. 6 with the filled symbols. As for the predictions based on the numerical results extracted in EP3 plane the tones are higher than the measured ones. This is attesting that the upwash velocity fluctuations are dumped while there are convected towards the fan. The reduction of the tones with the flow control devices is well captured. Except for the second harmonics, the HFC input data yields a prediction of tonal noise lower than the TFC one, showing that the former provides



cleaner inflow conditions.

### Broadband noise based on Numerical Simulation

For the trailing-edge noise model, the input data are extracted from an instantaneous field of the disturbed simulation at the five blade strip locations close upstream of the trailing-edge. The input parameters for the wall-pressure spectra are the boundary layer thicknesses, the maximum wall-shear stress along the boundary layer profile and the pressure gradient at the wall. The flow topology is shown in Fig. 9 where the trailing-edge is on the left hand side. The streaklines allow identifying a large flow detachment at the hub and a small one near the trailing-edge at the tip. The data extraction are extracted just upstream of the recirculation close to the tip at 95% of the chord. At mid span the shear-stress lines are well aligned and parallel identifying a well attached boundary layer. According to the pressure contours, higher pressure gradient are found on the upper part of the blade span. The acoustic predictions are compared with the experimental spectra in Fig. 10. The contribution to the noise from each strip is also provided. Because of the flow detachment the acoustic contribution from the first strip has a higher roll-off slope. Surprisingly the second strip has a larger contribution at low frequency than the other strips, probably a consequence of the flow deflection coming from the hub recirculation. Finally the total spectra is within 3 dB agreement of the broadband noise level captured in the experiments, except above 7 kHz where motor drive additional noise could explain the higher levels in the experiments.

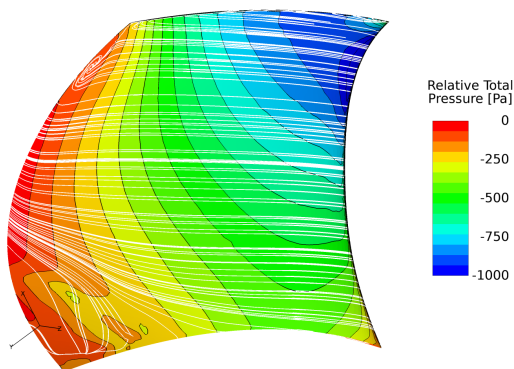


Figure 9: Visualization of the flow topology on the blade. The static pressure is shown with color contours and the streaklines are shown with the white lines.

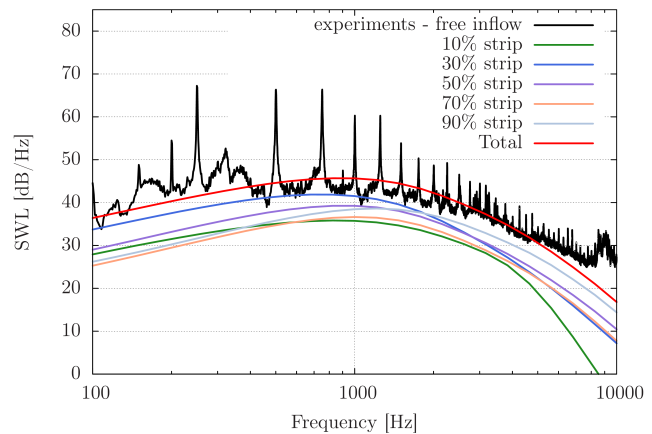


Figure 10: Comparison of the acoustic prediction from the trailing-edge noise model with experimental measurements upstream of the duct intake. The total noise spectra as well as the contributions from the 5 blade strips is provided.

### CONCLUSIONS

Unsteady numerical simulations have been performed using the SAS turbulent model on a 5-blade low-speed axial fan in a duct for which several flow and acoustic measurements have been performed. Inflow conditioners have shown to play a important reduction effect of the tonal noise. Two cases for the simulation have been considered to mimic the effect of the flow conditioners at the duct inlet. An undisturbed simulation using a uniform axial velocity inlet condition reproduces the hemispherical flow conditioner (HFC) experimental case. A disturbed simulation reproduces the free inflow condition. The inlet profiles are radial, circumferential and axial velocity maps that have been measured downstream of the bell-mouth using a rotating hot-wire test rig. Analytical tonal and broadband noise prediction available in the OPTIBRUI software have been computed using data

extracted from both simulations. The input for the tonal noise are the relative velocity maps upstream of the fan. An averaging procedure in the duct reference frame allows removing the potential effect of the rotor on the velocity components. The tone predictions for both the disturbed and undisturbed cases are in very good agreement with the free and HFC inflow experiments. The influence of the extraction plane is limited. Data extracted upstream of the bulk head provide tone levels 5 dB above the predictions from data extracted in the annular section. The distortion is dumped while propagating towards the fan. Assuming a quasi-steady evolution and using extractions from successive blade passage periods, fluctuations of the tone levels have been captured as identified in the experiments. Finally the analytical trailing-edge model has been applied to predict the broadband noise component of the acoustic spectra. The wall-pressure spectra is reconstructed from boundary flow data extracted close to the trailing-edge. The wall-pressure spectra model accounts for the Reynolds effects and the adverse pressure that develops along the chord. The acoustic model provides a good agreement with the broadband noise level in the measurements. The contributions from each strip can be related to the flow topology of the boundary layer on the blade suction side demonstrating the relevance of the wall-pressure spectra model.

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