Numerical flow noise simulation of an axial fan with a Lattice-Boltzmann solver

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Abstract – In this paper, a transient and compressible solver based on the Lattice-Boltzmann Method/Very Large Eddy Simulation (LBM/VLES) approach is employed to predict unsteady flow physics and flow-induced noise generation of a low-pressure axial fan. Aerodynamic and aeroacoustic measurements provided by the European Acoustics Association (EAA) benchmark platform are used for validation purposes. Boundary and design operating conditions are applied to the numerical model to replicate the experimental setup. Simulation and experimental data are compared, showing an excellent agreement in terms of fan efficiency with less than 1% deviation, as well as broadband and tonal noise within 0.7 dBA in terms of overall sound pressure level. An advanced post-processing analysis is performed to shed light on the noise generation mechanism in tip clearance. It is observed that both fine random turbulent structures and large coherent vortices are generated in the tip gap. The continuous impingement of the fine turbulence with the following blades and the blade itself is responsible for the radiation of broadband noise, while the interaction between the large coherent tip vortices, spinning at a lower angular velocity with respect to the fan shaft, and the following blades leads to the generation of narrowband peaks at sub-harmonics of the blade-passing frequencies. Finally, a beamforming analysis further confirms that the main noise sources are located in the blade tip clearance and tip regions.

Keywords: Aeroacoustics, Computational Fluid Dynamics, Lattice-Boltzmann Method, Axial fans, Sub-harmonic noise

1 Introduction

The development of fast and robust computational aeroacoustic methods for predicting the aerodynamic sound generated by low-pressure axial fans is essential for several applications, such as aeronautical, high tech, industrial and automotive ones. In the latter, low-pressure axial fans are often used in cooling units of engines or battery stacks [1]. Both are nowadays essential in electrified vehicles and for battery durability during fast charging. In particular, during battery charging, a substantial amount of air-cooling is used to remove the excess heat. Concerning industrial applications, axial fans are one of the key components of heat pump systems, whose market is increasingly growing in order to reduce fossil fuel consumption and, consequently, CO₂ emissions. In principle, axial fans deliver the required airflow that a heat exchanger of an air-source heat pump needs to operate. However, axial fans produce a certain amount of noise that is emitted into the environment as a by-product [2, 3]. Hence, there is an urgent need to quantify such a negative impact in terms of noise radiation, thus motivating the development and assessment of relatively fast, reliable and accurate computational aeroacoustics methods for fan noise prediction and mitigation [4].

Besides the development of fast computational simulation tools, the importance of benchmarking data for various applications in aeroacoustics is key to validate these tools [5]. Regarding the application of a low-pressure axial fan and its noise radiation signature, two noteworthy benchmark cases exist [6, 7]. The herein-used experimental benchmark data was issued by the European Acoustics Association under the name “A Benchmark Case for Aerodynamics and Aeroacoustics of a Low-Pressure Axial Fan” [8]. The dataset includes aerodynamic performance, fluid dynamic quantities, sound measurements, and the fan geometry.

Over the years, several contributions to the scientific field have been made for low-pressure axial fans [9–16]. A known sound generation mechanism is located at the blade tip and the tip gap. Zhu et al. [13] investigated these sound sources in detail for an axial fan impeller, finding that the presence of narrow-band sub-harmonic humps in the...
far-field was due to the interaction of coherent vertical structures, formed in the tip clearance, with the fan blades. Furthermore, in [14], the impact of the tip gap size on the overall flow field for an axial fan was investigated. They found that the mechanism triggering the turbulent transition depends on the tip gap size, indicating the strong effect of the tip gap vortex wandering on noise emission. A common approach to suppress tip gap noise sources is to add a ring that connects the fan blades, which was investigated in [9]. During the design of the experimental benchmark setup, the flow is rectified again to avoid disturbed inflow conditions. The effect of these disturbed inflow conditions was previously studied and it was shown that the radiated sound can be reduced by using leading-edge serrations [15, 17]. This leading-edge modification further affected the stall onset of axial fan blades [15].

In [16], a hybrid aeroacoustic approach [18] based on the perturbed convective wave equation (PCWE) was presented and validated for a fan application. In [11, 12], the accuracy of the source term interpolation and a potential reduction of the computational cost of the above mentioned method were investigated. This hybrid aeroacoustic workflow was also applied to the EAA benchmark case in [10], where a Detached Eddy Simulation (DES) was used to resolve the underlying flow field. The validation showed good agreement for both aerodynamic quantities and sound pressure spectra. About 10 revolutions were considered for this investigation to quantify the acoustic results. Both analyses reasonably resolved the sub-harmonics peaks. However, the acoustic pressure spectra resolution provided no details on the blade passing frequency (BPF).

In recent years, the Lattice-Boltzmann Method (LBM) has been developed as an alternative Computational Fluid Dynamics (CFD) method for numerical simulations of turbulent flows. Contrary to Navier-Stokes equation-based approaches, which describe the fluid at continuum level, LBM describes the fluid in terms of a discrete kinetic energy equation for particle density distributions and the macroscopic flow properties are a direct result of the moments of these particle distribution functions [19–21]. The key advantages of LBM, compared to Navier-Stokes based methods, are the highly efficient parallelization, the simplicity of modeling complex fluids and the more straightforward handling of complex geometries and boundary conditions [20–22]. Moreover, due to the fact that LBM is low dissipative, compressible and provides an unsteady solution, it is intrinsically suited for aeroacoustic computations, without fundamentally requiring the coupling to any acoustic analogy.

The LBM was successfully applied to predict the flow-induced noise of several fan and fan housing geometries in [23]. Thereafter, the effect of blade sweep on the EAA benchmark fan has been numerically studied by using the LBM in [24]. A good agreement was found for the overall performance of three fans with different sweep configurations, reproducing, for example, broadband noise levels and spectral shapes within 1 dB over the whole frequency range. Such LBM simulations yielded insights into the noise emission mechanisms from the tip clearance, indicating that the sweep angle is a critical parameter for limiting blade-turbulence interaction and thus improving fan performance.

The primarily objective of this work is to assess the accuracy, prediction capabilities and the computational performances of LBM to predict the unsteady flow and the resulting noise radiated by a low-pressure axial fan. A best-practice for low-pressure fan noise prediction, based on the commercial LBM solver SIMULIA PowerFLOW® is presented. The EAA low-pressure fan benchmark is considered as a reference for validation, and it is shown that the proposed high-fidelity numerical approach is accurate for fan noise predictions, yet fast enough to be used in an industrial framework. The performance of the solver is reported for the sake of comparison with other CFD/CAA hybrid approach adopted in a previous work for the same benchmark [10], which are supposed to generate an equivalent amount of flow information at a similar level of fidelity. In addition to the validation of the numerical results, physical insights into the tip clearance noise generation mechanisms for a low-pressure axial fan are also provided.

The paper is structured as follows. Sections 2 and 3 briefly present the LBM numerical approach and the EAA benchmark, respectively. Section 4 describes the numerical set-up, including, among others, information on the resolution strategy and boundary conditions. In Section 5, after a mesh convergence study has been performed, the experimental and computational results with respect to aerodynamic and acoustic performance are compared, and both qualitative and quantitative insight into the noise generation mechanisms are further provided. Finally, the conclusion and summary of this work are drawn in Section 6.

2 Computational approach

In this study, the Lattice-Boltzmann Method (LBM) hybridized with a wall-modeled Very Large Eddy Simulation (VLES) approach implemented in the CFD solver SIMULIA PowerFLOW® is employed to compute the unsteady flow around the fan and its resulting far-field noise radiation.

2.1 Lattice-Boltzmann method

The LBM is based on the mesoscopic fluid description, in which representative collections of particles are tracked through the particle distribution function \( f(x, \xi, t) \), which represents the density of particles with microscopic velocity \( \xi \) at position \( x \) and time \( t \) [25]. The evolution in time of \( f(x, \xi, t) \) is given by the solution of the Boltzmann equation, which is discretized in the physical space and time, and the microscopic velocity domain into a prescribed set of velocity directions \( \xi \). For low-subsonic iso-thermal flows, the D3Q19 lattice scheme (19 velocity states in the three-dimensional space) is employed to recover to the Navier-Stokes equations [25].

The discrete-Boltzmann equation with an external body force reads as [26]:

\[
\begin{aligned}
-\frac{\partial f_i}{\partial t} + \sum_j \left( f_i^{eq} v_{ij} - f_i \right) &= \frac{1}{\Delta t} \left( \sum_j f_i^{eq} v_{ij} - \sum_j f_i v_{ij} \right) + \mathbf{F}_i \\
&= \frac{1}{\Delta t} \left( \sum_j f_i^{eq} v_{ij} - \sum_j f_i v_{ij} \right) + \mathbf{F}_i,
\end{aligned}
\]
\[ f_i(x + \xi_i \Delta t, t + \Delta t) - f_i(x, t) = C_i(x, t) + F_i(x, t), \]  
\[
\text{with } f_i \text{ and } F_i \text{ being the particle distribution function and external body force terms along the } i\text{-th lattice direction, respectively, while } \Delta t \text{ is the discrete time increment. The left-hand side of equation (1) corresponds to the advection of particles, while the right-hand side is the collision operator, governing the rate of change of } f_i \text{ due to particles collisions. The collision term } C_i(x, t) \text{ is modelled with the Bhatnagar-Gross-Krook (BGK) approximation [19, 27]:}
\]
\[
C_i(x, t) = -\Delta t/\tau[f_i(x, t) - f_{eq}^i(x, t)],
\]
\[
\text{where } \tau \text{ is the single relaxation time parameter, which is related to the fluid kinematic viscosity and temperature, while } f_{eq}^i \text{ is the particle distribution function at the thermodynamic equilibrium [28]. A regularized collision procedure is employed, resulting in significantly higher numerical stability and accuracy than the classical BGK collision operator [29]. Macroscopic flow quantities, such as density and momentum density, can be computed taking the appropriate moments of the distribution function } f_i:\]
\[
\rho(x, t) = \sum_i f_i(x, t), \quad \rho u(x, t) = \sum_i \xi_i f_i(x, t),
\]
\[
\text{while other physical quantities (i.e., pressure and temperature) can be determined through the thermodynamic relationships for an ideal gas [28].}
\]

In order to simulate rotating geometries around a fixed axis, the computational domain is decomposed into an outer (inertial) reference frame and a Local Reference Frame (LRF). The LRF is a volume sliding mesh that rigidly rotates with the rotating geometry so that no relative motion between it and the enclosed geometry occurs. Inside the LRF, equation (1) is used with the external body force term \( F_i \) being related to the inertial force associated with the non-inertial (rotating) reference frame [26]. Conversely, equation (1) without the body-force term \( F_i \) is solved in the inertial reference frame.

The LBM scheme is solved on a Cartesian mesh composed of cubic volume elements (voxels) distributed within different Variable Resolution (VR) regions. Since a grid resolution change by a factor of two is allowed between adjacent VRs, and an explicit time-marching scheme based on a unitary Courant-Friedrichs-Lewy (CFL) condition is used by the solver, the time step (based on the local grid size and lattice velocities) is also changed by a factor of two between adjacent VR regions. This time step adaption allows for an efficient speed-up of the transient flow simulation through a balanced domain decomposition based on the equivalent number of voxels updated at every time step.

The surface of solid bodies is automatically discretized by the solver within each voxel intersecting the wall geometry using planar surface elements (surfaces). No-slip and slip wall boundary conditions on each of these surface elements are implemented through a boundary scheme based on a particle bounce-back and a specular reflection process, respectively [20]. Therefore, very complex geometries can be quickly and automatically meshed by the LBM solver, simplifying the cumbersome manual work typically associated with the volume meshing step using body-fitted mesh-based CFD approaches.

Thanks to its inherent unsteady and compressible nature and to its very low dispersive and dissipative properties [30], the LBM is intrinsically suited for aeroacoustic simulations. Direct Noise Computations (DNC) can be performed without, in principle, the need for coupling with any acoustic analogy.

### 2.2 Very large eddy simulation

Although the LBM allows performing Direct Numerical Simulations (DNS), the computational cost associated with it is usually prohibitively expensive for cases of industrial relevance, and turbulence modeling is required. In the present work, the LBM scheme is therefore coupled to a modified two-equation RNG \( k-\epsilon \) model [31, 32], which is used to compute a turbulent relaxation time that is added to the viscous relaxation time, so that \( \tau \rightarrow \tau_{eff} \):
\[
\tau_{eff} = \tau + \tau_{turb} = \tau + C_\mu \frac{k^2/\epsilon}{T(1 + \tilde{\eta})^{1/2}}, \tag{4}
\]
\[
\text{where } C_\mu = 0.09, \text{ } T \text{ is the temperature, } k \text{ and } \epsilon \text{ are the turbulent kinetic energy and dissipation, respectively, and } \tilde{\eta} \text{ is a function of the local strain, vorticity and helicity parameters.}
\]

A swirl model incorporated into a modified RNG \( k-\epsilon \) model is employed to reduce the impact of the modeled eddy viscosity in regions of high vorticity, in order to locally resolve large anisotropic vortical structures where the underlying computational grid is fine enough [34]. It is worth mentioning that such an approach is not equivalent to that used in RANS (Reynolds-Averaged Navier-Stokes) method. Indeed, there is no explicit modification of the eddy viscosity in the LBM/VLES, in the sense that the modeled Reynolds stresses are not explicitly added to the governing equations, as in RANS approaches. Conversely, the RNG \( k-\epsilon \) model is used to modify the evolution of the system of particles towards the thermodynamic equilibrium (through a variation of the relaxation time) in a way that is consistent with the characteristic time scales of a turbulent flow, leading to Reynolds stresses that are the result of the LBM computation and not of a semi-empirical modeling [22]. Since the LBM uses a Cartesian mesh, different volume cell sizes along the three (or two) spatial dimensions and a stretch of the grid only in the wall’s normal direction is not possible. Hence, the resolution of the boundary layer down to the no-slip wall, which would require a wall-normal distance in viscous units \( (y^+ \leq 1) \) less than 1, is generally too computationally expensive. Therefore, to keep the computational cost relatively low, a wall function approach is generally used to model boundary layers on solid surfaces. Such a wall model is an extension of the standard law-of-the-wall formulation [35] and includes the effects of pressure gradients [36].
3 Validation data

The fan of the EAA benchmark is designed with the blade element theory for low-solidity fans. Regarding size and operating conditions (see Tab. 1), it is a typical fan to be used in industrial (heating, ventilation, air conditioning – HVAC) and automotive (thermal management cooling fan) applications.

The fan has no blade skew, and the design was not optimized for fluid dynamic or acoustic behavior. The blades consist of NACA 4510 profiles. Table 1 shows the fan design parameters. The circumferential velocity at the blade tip corresponds to a Mach number of 0.113. The Reynolds number based on the chord length is almost constant over the spanwise direction of the blade and suggests a turbulent flow. The experiment was conducted in a standardized anechoic inlet test chamber according to ISO 5801 (see Fig. 1).

The volume flow rate \( \dot{V} \) at the (benchmark) operating condition was adjusted by butterfly damper and an auxiliary fan in the inlet section. A flow straightener rectified the flow field in the first third of the inlet chamber. The duct was installed in the chamber wall, with the suction side facing inwards and the pressure side facing outwards. An electrical motor outside of the measurement chamber drove the fan, while torque and rotational speed measurements were obtained using a precision torque meter. To eliminate any influence of the bearing friction on torque measurements, a measurement of the torque offset was performed with the fan being removed. This offset was then subtracted from the measured torque when the fan was installed to retrieve the aerodynamic torque. The measured total-to-static pressure rise of the fan at the design point is \( \Delta p = 126.5 \) Pa, as the design pressure difference is not entirely reached due to unconsidered losses, such as tip flow. At the design point, the total-to-static efficiency according to,

\[
\eta = \frac{\dot{V} \Delta p}{2\pi n M_s},
\]

is \( \eta = 53.2\% \), with \( M_s \) and \( n \) being the torque and the rotational speed of the shaft, respectively. The reader can refer to [7] for more information about measurement setup.

Among other data, the available measurement results to be examined are found in [33] and contain the time signals obtained from seven 1/2 inch free-field microphones (type 4189-L-001, Brüel & Kjær) mounted on a planar semi-circular array with a radius of 1 m upstream of the inlet nozzle of the fan, as shown in Figure 2a. Furthermore, the time signals of the wall-pressure fluctuations at 15 transducers (XCS-093-1psi D, Kulite Semiconductor Products) located on the duct wall, and Laser Doppler Anemometer (LDA) measurements on the suction and pressure side of the fan blades, as shown in Figures 2b and 2c respectively, were recorded as well. The microphone and transducer signals are obtained with a sampling frequency of 48 kHz and a measurement length of 30 s. For all examined data, the precise locations are outlined in the relevant figures in the results section, and a detailed description is given in [7].

4 Simulation model

The computational setup provides a direct representation of the experimental one. It consists of an upstream chamber, an axial fan, a duct, a shaft and struts at the duct outlet and a motor drive (see Fig. 3a). A uniform flow is

![Figure 1. Experimental setup of the test chamber, adapted from [33].](image-url)
applied to the inlet of the digital test chamber, without the curved inlet and honeycomb flow straightener, for the sake of simplicity and numerical efficiency. In addition, to avoid acoustic reflections, a series of three large cubical damping zones between the test chamber and the simulation boundaries have been employed in the numerical setup. These damping zones are characterised by a gradual increase in dynamic viscosity, starting from the physical viscosity value in proximity of the chamber and rising by two orders of magnitude as they extend towards the outer volume region located at a significant distance from the chamber itself (Fig. 3b).

In order to properly discretize the computational domain, 12 VR zones are used, whose resolution varies by a factor of two compared to the adjacent VR (see Fig. 4). The finest grid resolution is applied to the tip gap and blade area on the suction side, where the highest velocity gradients and turbulent flow are expected. Concerning the resolution around the microphone area, 12 voxels per wavelength at 5.7 kHz have been considered to properly capture the acoustic propagation at sufficiently high frequencies. This means that in the present simulation, the microphone results for frequencies up to about 5.7 kHz can be trusted.

About the boundary conditions, the mass flow rate corresponding to the design operating point is set at the chamber inlet and ambient pressure outlet is set at the boundaries of the simulation domain. An acoustic sponge region is applied on the chamber inlet to reduce the acoustic reflections from the upstream wall. Non-reflective boundary conditions have been applied to the walls of the numerical test chamber to reproduce a digital anechoic environment consistent with the anechoic experimental chamber. Furthermore, to simulate rotating geometries around a fixed axis, such as the fan, an LRF (sliding mesh) is used as shown in Figure 4b and described in Section 2.1.

Finally, the simulation was initialized with a static pressure of 101325 Pa and quiescent flow conditions. The simulation time is set to 0.8 s, corresponding to a total of 20 revolutions: 5 revolutions for achieving convergence in terms of fan performance (settled mass flow and pressure rise) and 15 revolutions for the sampling of data. The wall-mounted and far-field probes are positioned as described in the experiment and recorded the pressure signal with a sampling frequency of 120 kHz.
This section compares experimental and computational results in terms of aerodynamic and acoustic performance, but also presents additional results that reveal insights about the sound generation mechanism of a low-pressure axial fan. The time signals are Fourier transformed (FFT) considering a bandwidth of 5 Hz for both the numerical and experimental datasets, resulting in 4 and 304 Welch’s windows, respectively, and Hanning windowed with 50% overlap.

5.1 Resolution study

A grid independence study is carried out with three different resolution sizes, but the same VR strategy as described in Section 4. Table 2 shows, for different grid variants, the size of the finest voxel $\Delta x$, the physical time step $t_n$, the total number of voxels, the computational cost on a 308 cores cluster with Intel Xeon Gold 6148 2.4 GHz, as well as the average $y^+$ values in proximity to the blades and pressure difference $\Delta p$ at the design point. The minor variations between the medium-resolution case and the fine one (3% on pressure rise) indicate the grid convergence in terms of mean aerodynamic quantity.

Table 2. Details of the different grid resolution levels with the corresponding computational costs and pressure rise.

<table>
<thead>
<tr>
<th>Case</th>
<th>$\Delta x$ [mm]</th>
<th>$t_n$ [s]</th>
<th>#Voxels</th>
<th>CPU time h</th>
<th>$y^+$ [-]</th>
<th>$\Delta p$ [Pa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coarse</td>
<td>0.625</td>
<td>$1.05 \times 10^{-6}$</td>
<td>$1.64 \times 10^7$</td>
<td>3500</td>
<td>41.2</td>
<td>119.9</td>
</tr>
<tr>
<td>Medium</td>
<td>0.415</td>
<td>$7.00 \times 10^{-7}$</td>
<td>$4.76 \times 10^7$</td>
<td>12000</td>
<td>30.3</td>
<td>123.8</td>
</tr>
<tr>
<td>Fine</td>
<td>0.312</td>
<td>$5.25 \times 10^{-7}$</td>
<td>$6.67 \times 10^7$</td>
<td>26000</td>
<td>23.6</td>
<td>127.7</td>
</tr>
<tr>
<td>Exp.</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>126.5</td>
</tr>
</tbody>
</table>

In addition, to examine the grid independence with respect to the acoustic quantities, Figure 5a shows the power spectral density (PSD) at microphone 4, for the different resolution variants. Both broadband and tonal noise results are weakly dependent of the employed grid resolutions, especially for the medium and fine ones, however some tonal peaks are slightly better captured in the fine simulation. A similar conclusion can also be drawn for the wall-pressure fluctuation results (see Fig. 5b, which shows the PSD at transducer 2). In view of the above, the results from the fine resolution grid are chosen to be further investigated and compared in detail with measurement data in the rest of the paper.

5.2 Mean flow

The numerical time-averaged static pressure field on a longitudinal plane cutting the test chamber is shown in Figure 6. Upstream of the fan, the pressure distribution is uniform, while a pressure jump is established between the suction and pressure sides of the fan, resulting in a higher static pressure downstream of it, as a consequence of the work done by the fan itself.

Furthermore, the comparison between experimental and simulated data for the pressure rise and efficiency at the design point is presented in Table 3, showing an accurate prediction by the LBM simulation, with a relative deviation of less than 1% observed for both quantities.

Figures 7a and 7b show the time-averaged and root-mean-square (RMS) of the velocity magnitude, respectively, on a longitudinal plane (perpendicular to the $z$-axis) across the test chamber. Upstream of the fan, the mean velocity field is quite uniform as a consequence the choice of neglecting the curved inlet and honeycomb straightener in the numerical setup. The flow is then accelerated by the fan, resulting in high flow velocity downstream it along the duct. A low velocity region can be noticed downstream of the hub up to the motor drive. No considerable interaction between the turbulent flow generated by the fan and the downstream shaft and the motor drive is observed (Fig. 7b). Furthermore, the RMS velocity result shows that the regions of highest turbulence levels are generated in the tip and root regions of the fan, which tend to coalesce and further mix-up as they are convected downstream.

Figure 4. Variable Resolution (VR) regions in overall view (a) and close-up, also showing Local Reference Frame (LRF) zone (b) around the fan.

Figure 6. Variable Resolution (VR) regions in overall view (a) and close-up, also showing Local Reference Frame (LRF) zone (b) around the fan.
In order to provide a quantitative validation of the aerodynamic near-field, the time-averaged axial velocity component $V_x$ on the suction and pressure sides at various positions along the radial coordinate $r/r_{duct}$ is shown in Figure 8. On the suction side, the numerical axial velocity profile is in fair agreement with respect to the experimental one. The flow is mostly accelerated over at inboard sections of the fan blade, while it attains lower axial velocity towards the duct wall (see Fig. 8a). On the pressure side, the numerical result is still in a fair agreement with the experimental data (see Fig. 8b), with the LBM simulation predicting the radial position of the axial velocity peak.

Table 3. Predicted pressure rise and efficiency compared with the measurement results.

<table>
<thead>
<tr>
<th></th>
<th>$\Delta p$ [Pa]</th>
<th>$\eta$ [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Exp.</td>
<td>126.5</td>
<td>0.532</td>
</tr>
<tr>
<td>LBM-Sim.</td>
<td>127.7</td>
<td>0.536</td>
</tr>
</tbody>
</table>

Figure 5. Power spectral density (PSD) at microphone 4 (a) and transducer 2 (b). Comparison between computational results of different resolution sizes and experimental ones.

Figure 6. Time-averaged static pressure field on a longitudinal plane.

Figure 7. Velocity magnitude field on a longitudinal plane: time-average (a) and root-mean-square (RMS) (b).

In order to provide a quantitative validation of the aerodynamic near-field, the time-averaged axial velocity component $V_x$ on the suction and pressure sides at various positions along the radial coordinate $r/r_{duct}$ is shown in Figure 8. On the suction side, the numerical axial velocity profile is in fair agreement with respect to the experimental one. The flow is mostly accelerated over at inboard sections of the fan blade, while it attains lower axial velocity towards the duct wall (see Fig. 8a). On the pressure side, the numerical result is still in a fair agreement with the experimental data (see Fig. 8b), with the LBM simulation predicting the radial position of the axial velocity peak.
similarly to the experiment (Fig. 8b). However, some deviations from 5 to 10% are observed compared to the experimental data for $0.63 < r/r_{\text{duct}} < 0.87$. A similar under-prediction has also been observed in a previous numerical study by Schoder et al. [10].

5.3 Wall pressure fluctuations

Figure 9 shows the comparison between the numerical and experimental PSD of the wall-pressure fluctuations for different transducers along the duct wall.

Upstream of the fan at the transducer 2, the peaks at the blade passing frequency (BPF = 225 Hz) and its higher harmonics ($2\times$BPF = 450 Hz and $3\times$BPF = 675 Hz), as well as the sub-harmonics humps ($\text{SH}_2 = 344$ Hz and $\text{SH}_3 = 514$ Hz), are well captured, as shown in Figure 9a. For transducer 7, which shows the highest broadband levels being positioned very close to the blade tip gap region, the tonal components are very well captured by the simulation. For transducer 13, downstream of the fan, the agreement with the experimental data is satisfactory, especially in terms of tonal components, while the broadband component is slightly underestimated at higher frequencies, most likely due to an excessive numerical dissipation of the turbulent structures convecting over the duct wall induced by a coarsening of the grid at the transducer location.

Finally, Table 4 compares the overall wall-pressure fluctuations, integrated between 100 and 6000 Hz, between
the experiment and simulation. It can be observed that the discrepancy between the two overall wall-pressure fluctuation results is considerably low, in agreement with the results shown in Figure 9. It is worth mentioning that the above-mentioned wall-pressure fluctuation results contain a superposition of both aerodynamic (dominant) and acoustic (minor) pressure contributions.

5.4 Acoustic field

The far-field noise spectra at the microphones placed upstream of the fan (Fig. 10) allow to assess the capability of the LBM to reproduce the sound generation mechanisms, as well as the resulting noise propagation. The numerical results shown in Figure 10 are directly obtained from the transient and compressible LBM simulation. As the 7 microphones are placed symmetrically with respect to the fan axis, only microphones 1 to 4 are shown in the following for the sake of conciseness.

For microphone 1, both broadband and tonal noise are predicted in a satisfactory way, with largest discrepancies taking place between the BPF at 225 Hz and the third sub-harmonic peak SH$_3$ at 514 Hz. The fact that the microphone 1 is located close to the wall is expected to be the cause for this discrepancy, and further investigation should be performed in this regard. A possibility could be to employ a lower acoustic absorption coefficient on the walls of the settling chamber in order to better resemble the experimental test rig. Regarding microphone 2, similarly to microphone 1, broadband and tonal noise are well predicted. However, the amplitude of the second sub-harmonic peak SH$_2$ is overestimated by roughly 5 dB, even though it matches well the experimental results in terms of frequency. Furthermore, for microphones 3 and 4, which are located closer to the axis of rotation, broadband and tonal components are both found in adequate agreement with the measurement data.

For all microphones, and more prominently, for microphones 3 and 4, the prediction of the first sub-harmonic hump SH$_1$ (at 153 Hz) does not accurately match the experimental result, neither in frequency nor in amplitude (e.g. Fig. 10c). This discrepancy could be attributed to the difficulty to capture the exact physical flow mechanism in the tip and gap regions either due to the insufficient size of the resolution or to the fact that the tip gap may vary between experiment and simulation, as the numerical model (fully rigid blade) does not account for the deflection due to aerodynamic and centrifugal forces, as also assumed in other cases previous [13, 38]. Moreover, the simulation time

<table>
<thead>
<tr>
<th>Transducer</th>
<th>2</th>
<th>7</th>
<th>13</th>
</tr>
</thead>
<tbody>
<tr>
<td>Exp.</td>
<td>106.3</td>
<td>123.9</td>
<td>112.1</td>
</tr>
<tr>
<td>LBM-Sim.</td>
<td>105.6</td>
<td>123.6</td>
<td>111.5</td>
</tr>
<tr>
<td>Abs. Diff.</td>
<td>0.7</td>
<td>0.3</td>
<td>0.6</td>
</tr>
</tbody>
</table>

Table 4. Comparison between numerical and experimental integrated wall-pressure fluctuations (100–6000 Hz) in dBA at the indicated transducers.

Figure 10. Comparison between numerical and experimental far-field noise power spectral density (PSD) at the microphones 1 (a), 2 (b), 3 (c) and 4 (d).
may not be sufficiently long to properly capture such low-frequency noise components in a statistical sense.

It is worth noting the presence of the tone at the BPF, for all the microphones analysed, despite the choice not to model the curved inlet and honeycomb straightener in the present study. This suggests that the presence of the BPF tone in the current simulation might be attributed to a generation mechanism other than the mild inflow non-uniformity generated by the curved inlet and honeycomb straightener. A solid Ffowcs-Williams & Hawkings' (FW-H) acoustic analogy analysis – not shown here for the sake of conciseness – considering all the relevant surfaces (fan, duct, walls, struts, shaft and motor) has been also performed for different far-field probes upstream and downstream of the fan in order to shed more light on this aspect. The solid FW-H results did not show the presence of a relevant BPF component in far-field noise spectrum of the struts, but only in the noise spectrum associated to the fan blades. This further suggests that the fan wakes impingement on the struts is also not responsible for the occurrence of the BPF tone in the noise spectra, whose origin could be attributed to some flow non-axisymmetry induced by the numerical simulation and interacting with the fan itself.

Overall, both narrowband and broadband noise components are predicted in a satisfactory way for all microphones, as also confirmed by the overall sound pressure level OSPL (100–6000 Hz) results in Table 5, which shows only a small discrepancy within a range of 0.7 dBA between experimental and numerical data.

Finally, Figure 11 shows instantaneous snapshots of the band-pass filtered pressure field on a longitudinal plane perpendicular to the z-axis for different narrow-band frequency bands centered around the SH1, BPF, SH2 and SH3 peaks, which provides a qualitative indication of the noise directivity at these frequencies.

From Figure 11a, it can be observed that the directivity of SH1 is rather uniform in all directions. A similar behaviour is manifested by the acoustic field at the BPF (Fig. 11b). The radiation at SH2 is more prominent at 45° with respect to the fan axis (Fig. 11c), as it can also be seen from the far-field noise results (Fig. 10). Concerning the SH3 (Fig. 11d), the noise radiates more efficiently along out-of-plane directions (i.e. outside the plane of the fan). Moreover, from all the frequency bands investigated, the distinction between aerodynamic and acoustic pressure components is evident. The former are limited to near-field flow region, i.e. in proximity of the fan blades and in the wake of the fan. Both upstream of the fan, where the microphones are located, and downstream of it at a sufficient distance from the wake

<table>
<thead>
<tr>
<th>Microphone</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Exp.</td>
<td>71.8</td>
<td>78.4</td>
<td>80.1</td>
<td>80.5</td>
</tr>
<tr>
<td>LBM-Sim.</td>
<td>71.4</td>
<td>78.7</td>
<td>79.8</td>
<td>79.8</td>
</tr>
<tr>
<td>Abs. Diff.</td>
<td>0.4</td>
<td>0.3</td>
<td>0.3</td>
<td>0.7</td>
</tr>
</tbody>
</table>

Table 5. Comparison between numerical and experimental overall sound pressure level (100–6000 Hz) in dBA at the indicated microphones.

Figure 11. Instantaneous band-pass filtered pressure on a longitudinal in a frequency range of SH1 = 153 ± 10 Hz (a), BPF = 225 ± 10 Hz (b), SH2 = 344 ± 10 Hz (c) and SH3 = 514 ± 10 Hz (d). Black dotted marks indicate the microphone positions.
of the fan, only acoustic waves propagate, indicating that this area can be reliably considered as an acoustic far-field.

5.5 Physical insights

To investigate the complex noise generation mechanisms occurring in the tip region area, and to correlate them with far-field noise analysis, first the instantaneous isosurface of $\lambda_2$ criterion colored by the vorticity magnitude is shown in Figure 12. A significant amount of random fine turbulent structures are generated in the blade tip gap region and continuously interact with the blade itself, resulting in non-constant unsteady loading and leading to the radiation of broadband noise. Moreover, larger coherent vortical structures are also created within the tip gap region and interact with the following blades, as depicted in Figure 13 by means of instantaneous flow streamlines. Since such coherent structures “swirl” (i.e. rotate) at a lower angular velocity compared to the fan rotational speed, they result in the generation of the sub-harmonic peaks, as previously pointed out by Zho et al. [13].

An intuitive way to explain the presence of the SH peaks can be illustrated through Figure 14, which shows the relative “swirling” frequency $f_{rel}$ of the coherent vortical structures on a plane perpendicular to the axis of rotation and about 10 mm downstream of the fan blades leading edge. Such a relative frequency is evaluated as:

$$f_{rel} = U_c(z f_{fan} - f_{V_t})$$

where $U_c$ represents the convection velocity coefficient, $z = 9$ is the number of blades, $f_{fan} = 24.77$ Hz is the fan shaft frequency and $f_{V_t}$ is an equivalent “swirling” frequency based on the time-averaged tangential velocity component $V_t$ of the mean flow through which the vortical structures are convected. Note that the quantity between round brackets in the equation above represents the frequency at which the fan blades would interact with turbulent structures convected at the exact same velocity as the underlying flow entrainment. Since turbulent structures are generally convected at a lower velocity compared to the velocity of the mean flow that carries them, a constant convection velocity coefficient $U_c = 0.7$ (usually taken between 0.6 and 0.8 [39]) has been considered to account for this effect in the first instance. In the limits of the approximation and assumption made (i.e. convection velocity not function of the frequency), this result indicates that the relative “swirling” frequency, at which the coherent vortical structures generated in the tip clearance region would impinge on the following blades, agrees in a satisfactory way with the central frequency of the SH1 hump $\sim 153$ Hz, as shown in Figure 10. The fact that such vortical structures are organized into multiple helices (up to 3, as shown in Fig. 13) leads to the occurrence of multiple blade-vortex interactions per blade-passage and corroborates the presence of the higher SH peaks.

Next, to obtain an indication of the wall-pressure fluctuations on the fan blades, Figure 15 shows the power spectrum of the surface pressure at four different frequency
ranges centered around the SH1, BPF, SH2 and SH3 peaks. As expected, the highest wall-pressure fluctuations occur at the blade tip, where an intense turbulence-blade interaction takes place. Interestingly, the highest pressure fluctuations levels are found for the SH2 and SH3, which are also the most dominant peaks observed in the far-field noise spectra.

Finally, the comparison between Clean-SC [40] beamforming noise maps on the blade suction side for the three different frequency ranges is shown in Figure 16. These noise maps are computed from FW-H acoustic signals evaluated from a single blade on a spiral array of 400 microphones of 200R radius, which is defined in a reference frame rigidly rotating with the fan for unsteady loading noise sources localization. The array is centered around the fan hub and located at 2R distance from it in the upstream direction. The beamforming antenna resolution at the minimum (f = 100 Hz) and maximum (f = 4300 Hz) frequencies of interest is approximately equal to 35% and 1% of the mean chord, respectively. The employed Clean-SC deconvolution algorithm is implemented in the beamforming code OptiB-BF by Dassault Systèmes. For all the three frequency ranges, the main noise sources are located in the

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Figure 15. Power spectrum pressure at the fan suction side in a frequency range of SH1 = 153 ± 10 Hz (a), BPF = 225 ± 10 Hz (b), SH2 = 344 ± 10 Hz (c) and SH3 = 514 ± 10 Hz (d).

Figure 16. Clean-SC beamforming noise maps on the blade suction side for three different frequency ranges: 100–700 Hz (a), 700–1900 Hz (b) and 1900–4300 Hz (c). Values below 60 dB are not shown.
blade tip region at a mid-chord location, where the interaction of the blade tip with the incoming turbulent flow shed by the tip of the preceding blade, as well as with the turbulent flow generated in the tip gap region, takes place. In particular, the unsteady loading noise sources appear to be more clustered around the tip gap for the frequency range covering the SH and BPF peaks (100–700 Hz). Conversely, for the other frequency ranges of interest (700–1900 Hz and 1900–4300 Hz), the noise sources are detected at slightly more inboard locations. Interestingly, for such frequency ranges, secondary noise sources are also found by the beam-forming algorithm at the leading-edge, due to its interaction with the incoming turbulent flow from the preceding blade; at the blade trailing-edge, as a consequence of the scattering of the turbulent flow convecting over the blade; and in the blade root region, due to the occurrence of some flow separation over the hub surface and innermost blade sections (as shown in Fig. 12).

6 Conclusive remarks

In this work, the LBM/VLES approach was successfully employed to predict the aerodynamic and aeroacoustic performance of the low-pressure axial fan of the EAA benchmark case. The numerical setup was conceived with the goal of efficiently and accurately predicting low pressure axial fan noise in an industrial context. A grid independence study was performed and demonstrated the quality of the convergence of both aerodynamic and aeroacoustics quantities. The computational effort of the finest employed grid was about 1300 CPU hours per fan revolution, but satisfactory aerodynamic and aeroacoustic results have been already achieved with a coarse mesh at a cost of 175 CPU hours/rev. The overall wall clock time on a 308 cores cluster corresponded to approximately 84 and 11 h for the fine and coarse simulations, respectively, making the proposed approach highly competitive in terms of turnaround times for industrial applications.

Flow quantities such as pressure rise, fan efficiency and axial velocity component on the suction and pressure side showed an excellent agreement with the measured data. In addition, the predicted levels of wall-pressure fluctuations along the fan duct wall, as estimated from the wall-mounted pressure transducers, agreed well with the experimental results. Direct noise calculation of the far-field noise also showed a quite satisfactory prediction of both broadband and tonal noise levels compared to the experiment.

An advanced analysis was performed to shed light on the noise generation mechanisms in the tip clearance. It was observed that both fine random turbulent structures and large coherent vortices generated in the tip clearance interact with the blade itself and the following blades. The continuous impingement of the former with the blades led to the radiation of broadband noise, whereas the interaction of the coherent tip vortices – spinning at a lower angular velocity with respect to the fan shaft – with the following blades resulted in the generation of sub-harmonic humps at frequencies lower than the blade-passing frequencies. Furthermore, the highest wall-pressure fluctuations at the tip of the fan blades occurred at frequencies centered around $\text{SH}_2$ and $\text{SH}_3$, thus explaining the more pronounced tones in the far-field noise at such sub-harmonics. Finally, a beam-forming analysis for different frequency ranges further demonstrated that the main sources of narrowband and broadband noise are located in the blade tip gap and tip regions.

Conflict of interest

The authors declared no conflict of interests.

Data availability statement

Data are available on request from the authors.

References