Noise mechanisms in a radial fan without volute

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Abstract. The noise emission of a large radial fan with 7 backward curved blades is investigated. When decreasing the flow rate, the broadband levels below 1 kHz increase and the tones decrease, and a broad hump appears at about 80% of the BPF. The present numerical investigation aims at identifying the mechanism responsible for the broadband hump in the spectra. The configuration is simulated using the Lattice Boltzmann PowerFLOW solver for four flow-rate conditions. The low flow-rate condition is run longer to achieve statistical convergence at low frequencies. The simulations capture reasonably well the acoustic spectra and in particular the broadband hump at lower flow rate. The flow in the fan shows an unsteady flow detachment in some blade passages shed from the leading edge of the blade, while other blade passages experienced a fully detached flow along the whole blade chord. A spectral proper orthogonal decomposition is applied to identify the modal structures of the flow responsible for this mechanism. Several azimuthal modes are obtained from the decomposition for the different frequencies in the acoustic spectra. In addition an axial acoustic mode in the fan bowl is found at a frequency in the broadband hump.

1. Introduction

Radial fans are commonly used in heating and air conditioning (HVAC) systems for their ability to deliver high pressure rise with low flow-rate compared to axial fans. They can be installed in an adapted volute or in a large plenum when static pressure rise is of main interest. In the latter case, the interaction of the flow induced by the rotor with stationary parts of the assembly is reduced yielding limited tonal noise emergence at the blade-passing harmonics in the far-field acoustic spectra. Instead, some broadband humps at different frequencies appear. The aeroacoustic noise sources in low-speed radial fans have been investigated by several authors both experimentally [1–3] and numerically [3–5].

The challenge of noise predictions with numerical simulations for a low-speed radial fan, rises from the requirement to model the full fan geometry in its installation environment and the low frequency content to be resolved. In addition, the extremely complex and naturally unsteady turbulent flow fields in the blade passages make the analysis of the numerical results difficult. The rotational effects and the highly cambered blade typically used in such fans with low blade count can induce jet-wake patterns, large secondary flows and massive separations which can fill the whole blade passage depending on the operating conditions [2, 6]. In addition to the complex...
flow structures, several authors [1, 7, 8] have identified experimentally naturally developing azimuthal modes also called rotating instabilities which appear at all operating points, but are particularly amplified at low flow rate conditions. The interactions of these modes identified in the blade passages, and in the discharge are related to the noise emissions at frequencies different from the blade passing harmonics [1, 3].

In the present work, the noise source mechanisms are numerically investigated on a large radial fan with a low blade count using the Lattice Boltzmann Method as implemented in PowerFLOW allowing a detailed resolution of the turbulent field over a long physical time and direct acoustic predictions. A comprehensive analysis of the flow field is provided at a low flow rate operating condition for the fan installed without volute between two large plenum. In the aim of analyzing the modal content and coherent flow features at different frequencies, the spectral proper orthogonal decomposition (SPOD) is applied on time-resolved numerical results. SPOD, as first introduced by Lumley [9] for the analysis of turbulence, has been shown to represent physically meaningful coherent structures while providing an optimal base of dynamic modes [10]. The objectives of the present paper is to relate the modal content of the flow fields, also called rotating instabilities [1, 11], to the aerodynamic flow features which can be related to the aerodynamic performance and noise sources.

The paper is organized as follows. The investigated radial fan installation and experimental results are first described in Section 2. The numerical configuration and parameters are briefly described and the validation of the numerical model are provided in Section 3. The flow features, acoustic results and aeroacoustic source analysis for a low flow rate operating condition are analyzed in Section 4. The SPOD modes computed from the low flow rate operating condition are analyzed in Section 5 and related to classical rotating instability theory. Some conclusions and perspectives are drawn in Section 6.

2. Experimental investigation

2.1. Configuration description

The investigated radial fan of diameter $D = 355$ mm shown in figure 1 (left) has 7 backward curved blades made of metal sheet painted for a smooth finish surface. The blades weld to the flat hub disk and the curved rotating shroud are simple circular profiles extruded perpendicularly to the flat hub disk. The fan is installed without housing. The Ziehl-Abegg anechoic facility (Künzelsau, Germany) shown in figure 1 (center) consists of two large anechoic rooms separated by a partition. A bell-mouth attached to the partition guides the flow into the fan installed in the downstream plenum (figure 1 (left)). The gap between the rotating shroud of the fan and the convergent intake is of 2 mm. The fan is tested at constant rotational speed $\Omega = 2975$ RPM for various flow rates, yielding a blade-passing frequency (BPF) of 347 Hz.

![Figure 1. Left: view from downstream of the investigated radial fan attached to its bell-mouth. Center: anechoic facilities at Ziehl-Abegg. Right: measured acoustic power downstream of the fan.](image-url)
2.2. Acoustic results
For different flow-rates, the acoustic sound power levels measured downstream of the fan exhibit different features that are shown in figure 1 (right). The tones at the blade passing frequency harmonics decrease with flow rate. The broadband noise level and sub-harmonic hump levels increase as the flow rate decreases. The main objective of the present work is to investigate the modal content of the turbulent flow in the present configuration causing the emergence of the sub-harmonic humps in the noise spectra. The turbulent flow is investigated using numerical simulations as described in the next section.

3. Numerical investigation
3.1. Numerical parameters
The present simulations use the PowerFLOW solver 5.5c based on the Lattice Boltzmann Method (LBM). The approach is naturally transient and compressible providing a direct insight into hydrodynamic mechanisms responsible for the acoustic sources but also into acoustic propagation in the set-up and outside in free-field.

Instead of studying macroscopic fluid quantities, the LBM tracks the time and space evolution of a truncated particle distribution function on a lattice grid. The particle distribution evolution is driven to the equilibrium by the so-called collision operator, approximated by the BGK model. The discrete Lattice-Boltzmann equations are solved with 19 discrete velocities for the third order truncation of the particle distribution function, which has been shown sufficient to recover the compressible Navier-Stokes equations for a perfect gas at low Mach number in isothermal conditions[12, 13]. In PowerFLOW, a single relaxation time is used, which is related to the dimensionless laminar kinematic viscosity [13]. This relaxation time is replaced by an effective turbulent relaxation time that is derived from a systematic Renormalization Group procedure detailed in [14]. It captures the large structures in the anechoic room but also the small turbulent scales that develop along the blade and wall surfaces in a large eddy simulation manner. The particular extension developed for rotating machines can be found in Perot et al. [15].

The computational domain includes the two coupled anechoic rooms as shown in figure 2 (left). The square cross section of the simulation volume has a width of 10 m and the upstream and downstream plenums are 30 m and 20 m long respectively. The atmospheric pressure $p = 101,325$ Pa is imposed on the left red surface, while the volume flow-rate $Q_v$ is imposed on the downstream red surface. The convergent and detailed geometry of the fan are considered. Only the motor and its shaft are discarded in the numerical set-up. The mesh resolution in the domain is defined through successive embedded volumes, outlined in figure 2, the mesh size being decreased by a factor of 2 between them. The resolution around the fan blades and the convergent wall is 1 mm as shown on a fan cross section in figure 2 (right). The volume around the fan is defined as a rotating volume at a rotational speed $\Omega$. Extended logarithmic wall functions to account for pressure gradients [16] are applied on all wall surfaces. The total numerical domain counts 98 million cells and 7 million surface elements.

Simulations are performed for several flow rates. First the setup is initialized from uniform flow at rest with a coarse mesh, then the final mesh described above (figure 2 (right)) is used for performance prediction. The convergence is obtained after one fan revolution, and data are acquired for six fan revolutions. Static pressure is recorded at the exact locations of the microphones in the experiments. In addition, some cross-sections at mid-blade height and blade surface are recorded at high frequency sampling for frequency analysis. The numerical model presented above is validated through the comparison of performances with experimental measurements acquired on the certified ISO 5801 Ziehl-Abegg facilities.
3.2. Global performance results
In the downstream volume, a large recirculation is established. Hence, the precise volume flow-rate $Q_v$ through the fan is obtained from integration over an upstream hemispheric surface 1 m upstream of the fan. The pressure is averaged on two cross-section planes located at 2 m away from the partition wall. The total to static pressure rise $\Delta P_{ts}$ is obtained by difference considering the total and static pressures on the upstream and downstream planes respectively. The axial torque $M_x$ is obtained from surface integration of the axial angular momentum. The numerical results are compared to the experimental global performances in figure 3. The dimensionless flow, pressure and efficiency coefficients are defined respectively as:

$$\phi = \frac{4 Q_v}{\pi^2 D^3 \Omega}$$
$$\psi = \frac{2 \Delta P_{ts}}{\rho \pi^2 D^2 \Omega^2}$$
$$\eta = \frac{\Delta P_{ts} Q_v}{2\pi \Omega M_x}$$

(1)

Figure 3. Comparison of global performances between experimental and numerical results: pressure coefficient (left) and efficiency coefficient (right) as function of flow coefficient.

The total to static pressure coefficient is well predicted with the present LBM numerical model except for the higher flow-rate. The torque is over-predicted due to under-resolved boundary layer considering the 1 mm cell size at the blade surface. Nevertheless, the total to static efficiency is about 0.04 lower than the measured one. The predicted global performances are considered as satisfactory enough to provide relevant inputs for the aeroacoustic driving mechanisms.

4. Aeroacoustic results at low flow rate
In the following, the lower flow-rate simulation at $\phi = 0.13$ is considered for the aeroacoustic and modal analysis. The simulation has been continued for 50 fan revolutions to provide sufficient statistical samples for the modal analysis up to 100 Hz.
4.1. Aerodynamic flow investigation

The flow patterns inside the fan for the considered low flow-rate condition are highly non uniform in the different blade passages as can be seen in figure 4. The static pressure in the discharge has a sort of modal structure with 7 lobes. Inside the fan, strong flow detachment can be observed in three blade passages, while in the remaining four blade passages flow detachments appear limited to the upstream part of the suction side. In the center of the fan the velocity seems to be distributed with a four-lobe pattern. With the $\lambda_2$ criterion (second invariant of the velocity gradient tensor) isosurface the complexity of the turbulent flow in the blade passages is clearly demonstrated. Three passages show large vortical structures covering the full passage. They shed large coherent structures in the discharge. The four other blade passages show a typical jet-wake pattern, where the main flow is ejected along the pressure side, while coherent vortical structures develop along the suction side and create a blockage.

![Figure 4](image-url)

**Figure 4.** Instantaneous flow patterns at mid-height of the fan blades. (Left) Static pressure contours, (center) velocity magnitude (in the local reference frame: absolute in the bowl and discharge area, relative in the blade passages) and projected streamlines, $\lambda_2$ isosurface $-100 \text{s}^{-1}$ colored with vorticity magnitude (right).

4.2. Acoustic results

![Figure 5](image-url)

**Figure 5.** Acoustic spectra from numerical probes located in the downstream plenum. The modal shape identified in the first mode of the SPOD analysis are shown in red. $m$ corresponds to the azimuthal order identified.

The acoustic spectra downstream of the fan are shown in figure 5. As for the experiment, the broadband spectrum is dominated by peaks around frequencies lower (roughly at 78%) than the harmonics of the blade passing frequency. The main emission is not emitted in phase by all blades yielding a lower frequency than the BPF. This agrees with previous observations by Mongeau et al. [1] where this mechanism is associated with rotating modes. According to their theory, the main peak would be associated to a 7-lobes azimuthal mode rotating at lower rotational speed than the fan (blue dashed line). The spectral proper orthogonal decomposition is considered as an appropriate tool to investigate the modal content at a given frequency [10].
5. Modal analysis

5.1. SPOD technique
To relate the acoustic emission to the modal content of the turbulent flow already highlighted in section 4.1, the SPOD technique is applied in the stationary reference frame. From volume data recorded at a sampling frequency of 9,700 Hz during 50 fan revolutions, the pressure and velocity components are interpolated onto a structured stationary grid of a volume cylinder of 500 mm radius including the fan and the fan intake. The rotating domain is rotated at its instantaneous position in phase with the stationary domain before the interpolation. The latter is performed with a weighted distance method using 35 points of interpolation. The volume mesh has 320 points in the azimuthal direction, 60 points in radius. The Welch’s method is applied to the snapshot sequence using an overlap of 50% between time windows and 30% of zero padding. The frequency realization ensemble corresponds to \( n_{blk} = 40 \) blocks of frequency resolution 35 Hz. For each frequency, an eigenvalue problem of size \( n_{blk} \times n_{blk} \) is solved [10], providing an optimal decomposition that represents the dynamics of the flow field. SPOD modes at the same frequency and at different frequencies are uncorrelated [10], providing a limited number of modes to analyze to obtain the understanding of the dominant dynamics. In addition, the obtained eigenvalues spectrum can be used to identify the number of dominant modes. Contrary to an azimuthal mode decomposition, the SPOD technique does not a priori assume any shape of the dynamics. Both the modal shape and the energy associated to the modes are the results of the eigenvalue decomposition at each frequency.

5.2. Modal analysis from the SPOD modes
The SPOD eigenvalue spectra on a cylindrical surface in the inlet bowl at \( D/3 \), and on two cross-section planes, one crossing the gap between the convergent and the fan rotating shroud and one at the mid-height of the blades, are shown in figure 6. The vertical lines correspond to frequencies identified in the acoustic spectrum. The planes crossing the fan are obviously strongly influenced by the blade passing dynamics. Yet lower frequencies appear, especially in the range identified in the acoustic spectra. In the bowl, the dominant frequency is not the blade passing frequencies but the main one observed in the acoustic spectra in figure 5. The first SPOD mode is largely dominant by up to three orders of magnitude, proving that the dynamics at several frequencies can be represented by a single mode function. Hence only the first SPOD is shown. The SPOD decomposition is done on the pressure and the three velocity components simultaneously, only the mode shapes for pressure and axial velocity will be shown.

![Figure 6. SPOD eigenvalue spectra from the decomposition at a cylinder in the inlet bowl (left), at fan-convergent gap axial position (center) and at mid-height of the fan blades (right).](image)

The first SPOD mode at 140 Hz is shown in figure 7 for the upper plane at the gap location and for the mid-height plane of the fan blades. The mode eigenfunction corresponds to an azimuthal mode of order 3. It is more coherent in the intake part of the fan than in the discharge at the gap axial location. In the plane at the gap position, the lobes are centered on the convergent wall near the leading edge of the blade. This mode is driving the flow-detachment and the vortical structures along the fan rotating shroud observed in figure 7 (right).
The first SPOD mode at 200 Hz is quite similar to the previous one as can be seen in figure 8 but the mode eigenfunction corresponds to an azimuthal mode of order 4. The lobes are moved to a further downstream location in the blade passages and secondary lobes can be distinguished at the trailing-edge of the blades.

The first SPOD mode at 260 Hz is an axial mode in the bowl as can be seen in figure 9. This frequency does not emerge (compared to other frequencies) in the mid-section of the fan blades.

Other azimuthal modes are observed at different frequencies and the orders \((m)\) are reported on the acoustic spectra in figure 5 (right). Note that these modes are rotating at a slightly lower frequency (95 to 100\%) than the fan. The same rotational speed is obtained by performing an azimuthal Fourier decomposition and measuring the phase time variation (not shown here). These findings somehow differ from the rotating instability theory [1]. This could be explained by the particular acoustic mode coupling in the axial direction very close to the main hump of the acoustic spectra.

6. Conclusions
The aeroacoustic noise sources of a large radial fan with 7 backward curved blades are investigated using the Lattice-Boltzmann PowerFLOW solver. The simulations provide a good estimation of the aerodynamic performances. The turbulent flow in the fan at low flow-rate is further analyzed. The flow at the mid-height of the fan shows an unsteady detachment in some blade passages shed from the leading edge of the blade, while other blade passages
experience a fully detached flow along the whole chord of the blade. The far-field acoustic directly computed by the flow solver captures a broadband hump at a frequency lower than the blade passing frequency. A spectral proper orthogonal decomposition is applied to identify the modal structures of the flow responsible for this mechanism. Several azimuthal modes are obtained from the decomposition for the different frequencies in the acoustic spectra. In addition an axial acoustic mode in the fan bowl is found at a frequency falling in the broadband hump.

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