ON THE FLUID MECHANICAL AND ACOUSTIC MECHANISMS OF SERRATED LEADING EDGES

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SUMMARY

On the basis of experimental investigations, numerical flow simulations were carried out on a fan with leading edge serrations. The fan operates in turbulent inflow and a good agreement with the experiment could be achieved. It was found that leading-edge serrations generate longitudinal vortices that are temporally and spatially stable and thus improve the flow over the airfoil. In addition, a vortex could be generated which reduces the leakage across the tip gap. The stabilization of the flow over the blade reduces the pressure fluctuations on the blade surface, which in turn leads to less sound radiation. The found physical mechanisms of serrations have great effects on simple fan geometry, but are reduced in optimized fans, which weaken their positive effects on the flow field and the sound emission.

INTRODUCTION

Due to ever-increasing legal requirements, as well as the electrification of the automobile, it is becoming more and more important that axial fans operate as quietly as possible. This is because axial fans are one of the most common and loudest sound sources in systems such as cars, trains and air conditioning systems. They often operate in disturbed inflow conditions created by heat exchangers or protection grids on the suction side [1]. Disturbed inflow conditions can lead to the sound radiation of the fans being amplified even further and cannot be estimated in advance due to the high complexity of the flow. If, as in the case of electric cars, other sound sources such as the combustion engine also disappear, the noise of the axial fan appears as the dominant source and causes harm to the health of the people in the immediate vicinity. In order to protect these people, the legal regulations on noise emission are tightened, which leads the manufacturers of axial fans to face challenges regarding the design of quiet axial fans. For this reason, an attempt has been made

to reduce noise emission by modifying the geometry of the axial fan blades. In particular, the leading edge of the fans, which acts as the contact between the fan blade and the incoming turbulent flow, was investigated. Various modifications of leading edges such as serrations, slots, waves or porous materials have been considered in the past [2-7]. However, often the application of these modifications is based on trial and error and a basic understanding of the physical sound reduction mechanisms are not investigated. In addition, it appears that the application of the modification was usually only suitable for a specific fan or a specific inflow, and was often accompanied by a loss of aerodynamic properties [8, 9]. In this study, based on good sound reduction properties of serrated leading edges, which have been determined experimentally [10], we focus on high resolution numerical simulations of leading edge serrations. With the help of the numerical simulations, a better understanding of the physical mechanisms of leading edge serrations will be derived.

AXIAL FAN DESIGN

The general considerations of these investigations are based on the idea that the leading edge modifications to be investigated should be analyzed as far as possible without any other influencing factors. For this reason, the leading edge serrations are investigated on simplified flat-plate fans. This allows to understand the aerodynamic and aeroacoustic effects of the serrations without having to consider any influence of skew, twist or blade profiles first. For the experimental and numerical investigations, two flat-plate fans with straight leading edge are used. The two fans differ in that the basic model has no serrations on the leading edge and the second has serrations on the leading edge (see Fig 1).

(a) Base fan (Ref).   *(b) Serrated fan (A167λ67).*
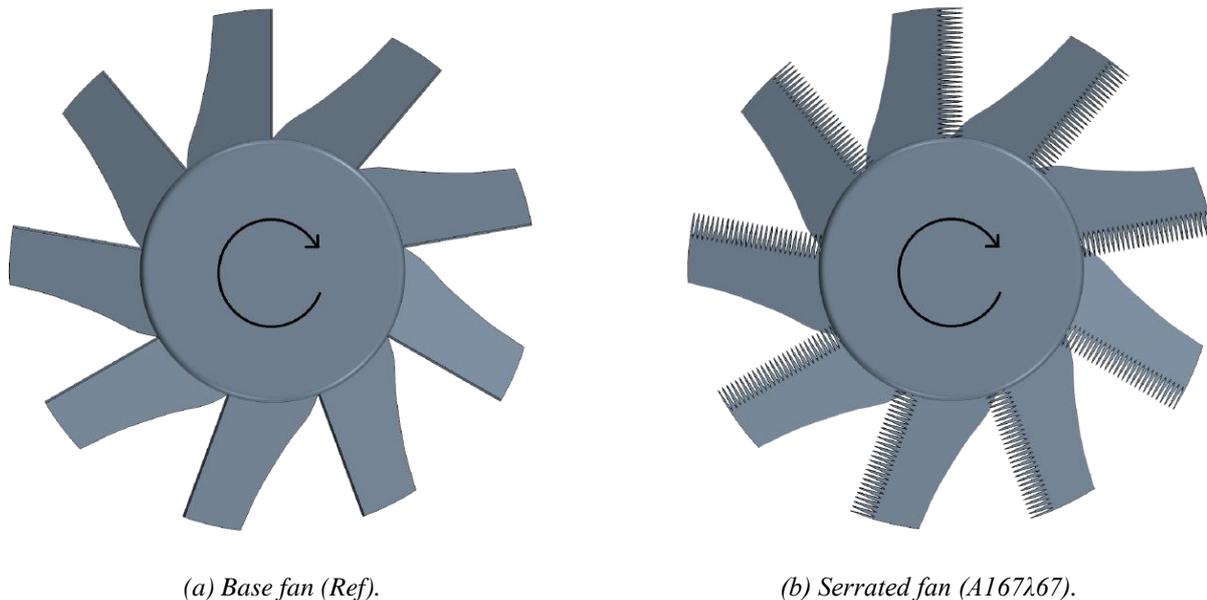
Figure 1: Investigated axial fans with and without leading edge serrations. Rotational direction clockwise.

The two axial fans are operated in a short duct segment with a diameter of d = 500 mm. The total diameter of the fans is d_{fan} = 497 mm and the resulting tip gap is s = 1.5 mm. The design speed of the fans is n_{fan} = 1486 rpm. The choice of these two fans is based on previously conducted experimental studies, which have already shown a great aerodynamic and acoustic potential of the selected serrations [2, 10]. Figure 2a shows the amplitudes and wavelengths of the serrations. The amplitude of the serrations is α_{LE} = 16.7 % and the wavelength is λ_{LE} = 6.7 % of the mean chord length of the axial fan (l_c = 72 mm).The serrations were designed in such a way that the blade surface between the base fan and the fan with serrations is identical.

(a) Parameters of serrations.   *(b) 3D Leading Edge*
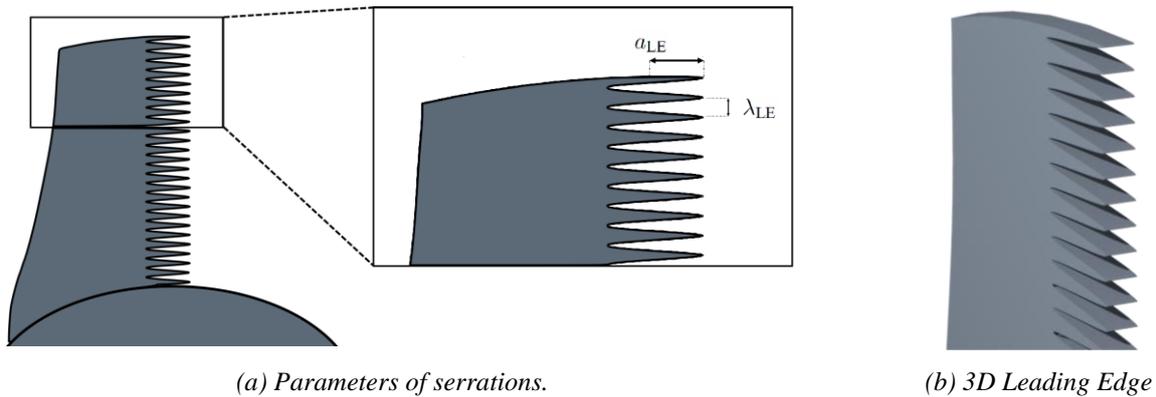
Figure 2: Parameters of the leading edge serrations.

EXPERIMENTAL SETUP

In these investigations, the numerical simulations of the leading edge serrations are in the focus of attention, which should serve to obtain a better understanding of the physical mechanisms of the serrations. Nevertheless, experimental investigations are also inevitable in this context. On the one hand, these serve to select the most efficient serrations in advance from an acoustic and aerodynamic point of view. On the other hand, they are needed for the aerodynamic and acoustic validation of the simulation results. In addition, a transfer of the serrations to profiled fans will be investigated by means of experiments at the end of this paper. This should help to find out how the effects of the serrations are influenced by other parameters like the blade profile or the blade twist. Figure 3 shows the test setup for the aeroacoustic investigations of the axial fans. The investigations were carried out in an axial fan test rig designed in accordance with DIN EN ISO 5801 [11]. On the suction side of the fans, the test rig has an anechoic chamber, which is used to analyze the emitted sound field [12]. For this purpose, seven microphones are used, which are positioned at a radius of R = 1m from the inlet nozzle of the axial fan. The measurement of the aerodynamic parameters (volume flow rate and total-to-static pressure rise) will take place simultaneously with the measurements of the sound pressure. In these investigations, the axial fans are operated with disturbed inflow. These are generated by a rigid turbulence grid, which has a mesh size of t_{mesh} = 24 mm and a bar width of t_{bar} = 4 mm. For further information on the test rig, the turbulence grid, the measurement technique and the test procedure, refer to the literature [3, 10, 12].
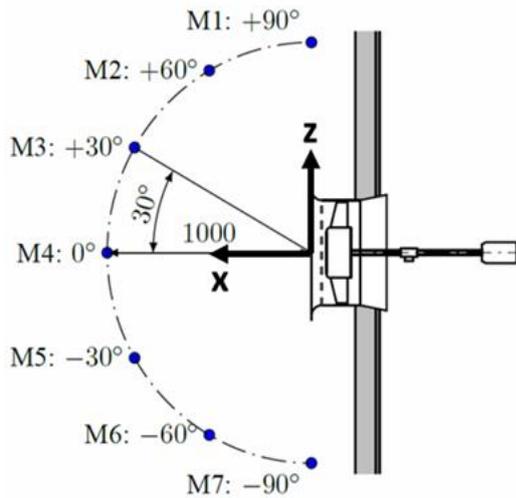
(a) Scheme of the experimental setup. *(b) Photo of the test setup.*

Figure 2: Experimental setup for the characterization of the sound emissions. The seven microphones used are free-field microphones from B&K and indicated here by the abbreviations M1-M7.

SIMULATION SETUP

The numerical investigations are carried out with the STAR-CCM+ software (Version 11.06.010) [13]. The most important components of the reference case are shown in Fig. 3. The setup used corresponds to that of the experimental investigations. The inlet and outlet area of the simulation is realized by two cylinders with a length of l = 2500 mm. This length corresponds to ten times the hub diameter of the fans. A trimmed mesh based on a hexadominant mesh structure is used to mesh the regions [13]. The grid structure of the mesh is gradually refined from the outlet and inlet to the rotor. The resulting mesh is shown in Fig. 4. The meshes used have a y+ value of y+ < 1 everywhere. In addition to these global mesh structures, the simulation mesh has been further refined in the area of the turbulence grid and the fan tip gap. The cell sizes for the different regions are given in Tab. 1.
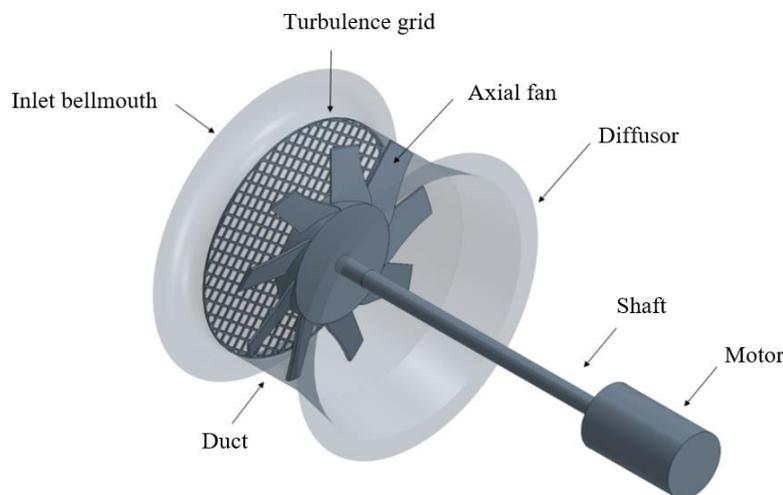
Figure 3: Illustration of the fan with duct system and turbulence grid.
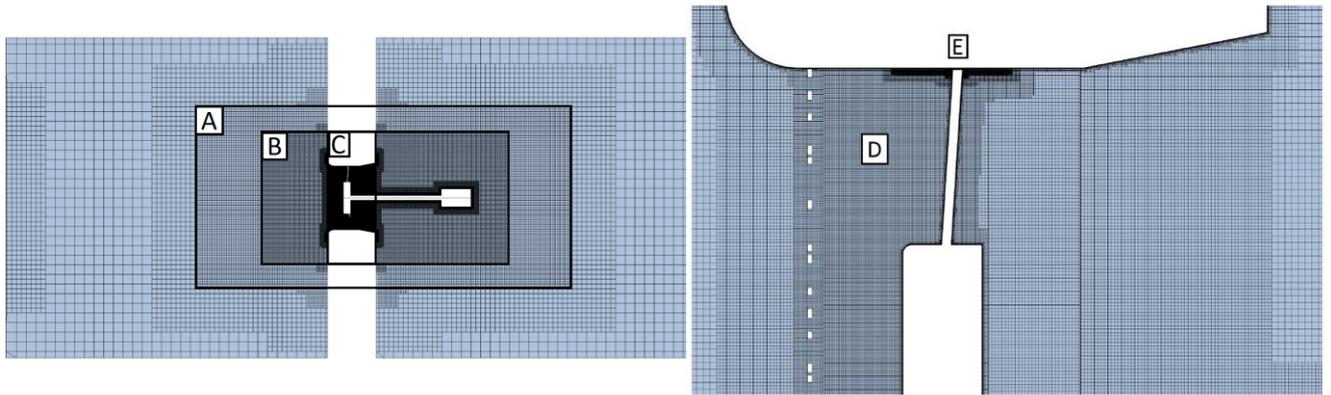
(a) Trimmed Mesh of the whole domain.   (b) Refinement in the fan region.

Figure 4: Topology of the simulation grid for the reference fan.

Table 1: Cell sizes of the different regions.

Regions	A	B	C	D	E
Cell size in mm	30	15	3	1.5	0.51

For the reference fan, the total size of the computational mesh is 90 million cells. For the fan with serrations, the total number of grid cells is 180 million due to the high resolution of the leading edge serrations. For this, cells in the area of the leading edge with a size of 0.5 mm were required. In order to be able to analyze time-related processes in the flow, a LES simulation was carried out for the cases. The fluid was assumed to have a constant density of ρ = 1.18 kg/m³. An inlet mass flow rate of $\dot{m}$ = 1.298 kg/s, corresponding to a volume flow rate of $\dot{V}$ = 1.1 m³/s, was selected. This volume flow reflects the optimum point of the fans at a speed of n = 1486 rpm. The outlet is defined as a pressure outlet with p = 0 Pa. For prior RANS simulations (SST-k-ω-model) 5760 CPUh were used for the stationary flow field. For the LES simulation with the WALE subgrid scale model used, 10 inner iterations with a 2nd-order discretization and a time step of t=20 µs were used. Here, 12 revolutions of the fan were calculated, with data collected starting with the sixth revolution. The physical total time is therefore t = 0.48 s. For the reference fan 135,200 CPUh and for the serrated fan 230,400 CPUh were used. The acoustic radiation of the axial fan is calculated via the FW-H integral method.

VALIDATION OF THE SIMULATIONS

In order to be able to assume that the numerical simulation also represents the real flow conditions and the real sound generation, a validation of the LES simulation to the experimental values of the characteristic curves must first be available. This forms the basis for a physical analysis of the effects of leading edge serrations on aerodynamics and aeroacoustics. With the reference fan, the simulation delivers a result that deviates by 1.7 Pa from the measurement of the operating point. For fan A167λ67 the deviation in pressure rise at the investigated operating point is 2.1 Pa. Based on these deviations, it can be assumed that the numerical calculations correctly reflect the real flow conditions and the operating point. Figure 5 shows a comparison of the sound radiation of the fans in the experiment and the sound pressure spectra calculated from averaging the 7 microphones signals in the simulation. There is a good agreement between the measurements and the numerically calculated FW-H integral formulation up to a frequency of 6 kHz.

(a) Sound pressure level spectrum (Ref).   *(b) Sound pressure level spectrum (A167λ67).*
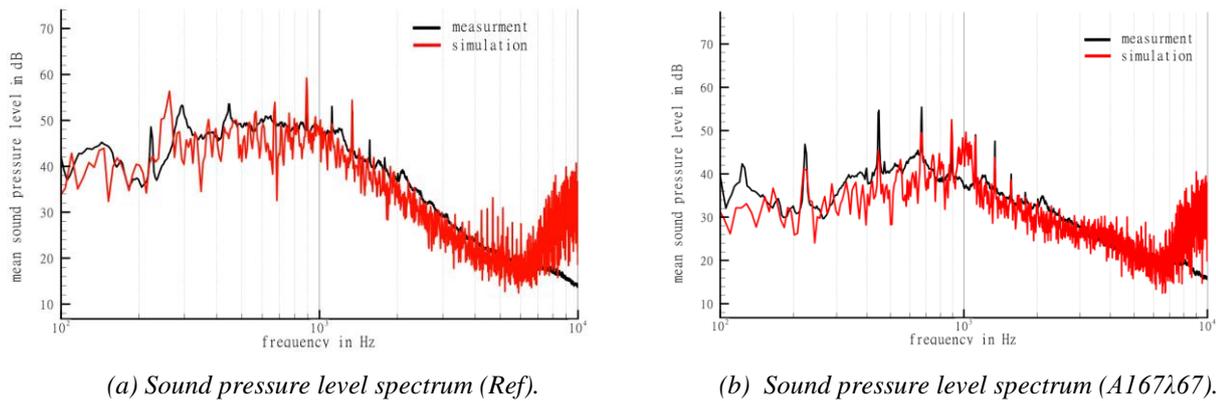
Figure 5: Comparison of the acoustic sound pressure spectrum between measurement and simulation for the investigated fans.

In addition, the fan A167λ67 induces a lower broadband sound in the frequency range up to 2 kHz. The dominant sound sources of the low-frequency broadband noise are usually located at the leading edge of the axial fans. In the case of the serrated fan, these are reduced by the leading edge modification, resulting in lower sound radiation in this region. The measurements indicate, that this measure can decrease the overall sound pressure level by 6.6 dB compared to the reference fan. In addition, it can be determined that the fan with serrations produces a slightly higher sound radiation from around 3 kHz. The sound pressure spectra confirm that the accuracy of the simulations is even so good that such effects, triggered by small-scale vortices, can be reproduced realistically.

IMPACT OF SERRATIONS ON THE FLUID DYNAMICS AND AEROACOUSTICS

Based on the LES simulations, it can be concluded that the serrations at the leading edge cause longitudinal vortices to form. Thereby, as seen in Fig. 6, two counter-rotating vortices are formed at each serration of the leading edge. Longitudinal vortices stabilize the flow and lead to a better overflow of the blades. In addition, the vortices created are stable over time, whereas with the reference fan, vortices change over time and travel towards the tip gap. In addition, the vortices of the leading edge serrations interact with the vortex system in the tip gap of the axial fan and create a dominant vortex (see Fig. 7), which leads to a reduced leakage flow across the tip gap of the fan. The flow through the tip gap is decelerated. Both the improved stability of the vortices across the blade and the reduction in leakage across the tip gap result in the axial fan with the leading edge serrations A167λ67 having a higher pressure rise in this case and thus a higher aerodynamic efficiency.
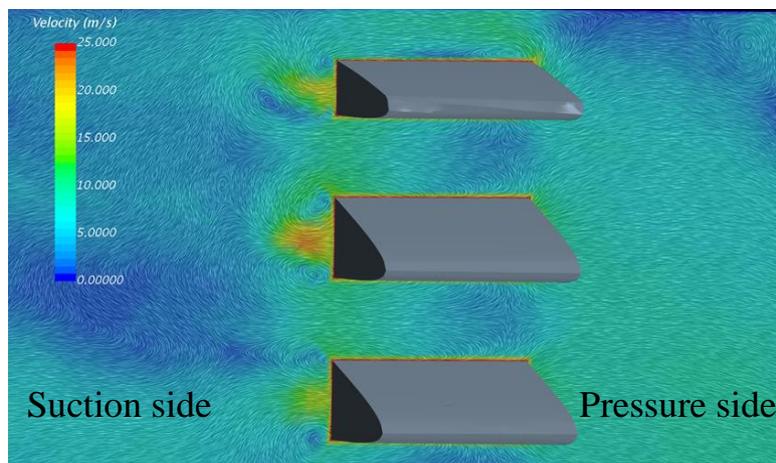
Figure 6: Flow field at the serration of the A167λ67 fan. On the suction side, longitudinal vortices form due to the serrations.
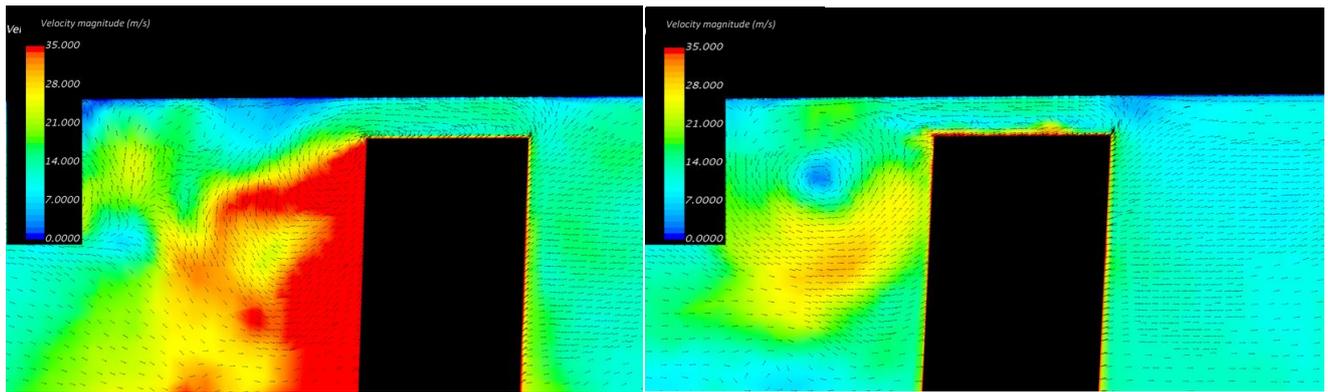
(a) Flow in the tip gap of the reference fan. (b) Flow in the tip gap of the serrated fan A167λ67.

Figure 7: Velocity magnitude in the tip gap of the investigated axial fans.
The serrated fan lowers the leakage over the tip gap due to the serrations.

In addition, the improved flow guidance by the serrations can be identified by the decrease in radial flow velocity. In the case of the reference fan, there is a radial flow component, which directs the flow towards the tip gap. There, an interaction takes place between the radial flow and the tip gap flow, which in the end results in an increased turbulent kinetic energy in the tip gap area. The improved flow guidance by the stabilizing contra rotating vortex pair at the serrated fan prevents the radial flow (see Fig. 8).
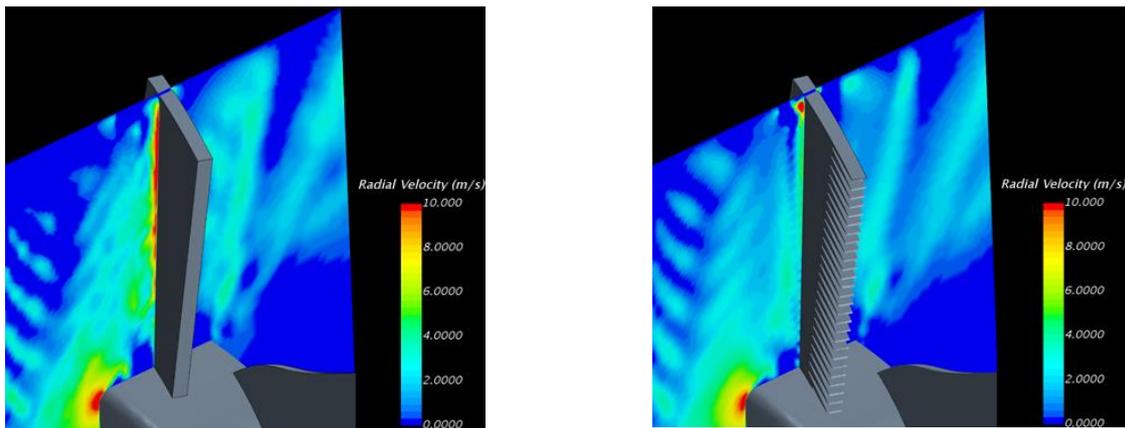
(a) Radial flow velocity (Ref). (b) Radial flow velocity (A167λ67).

Figure 8: Radial flow velocity over the fan blade.

The improved flow guidance due to the leading edge serrations is also reflected in reduced surface pressures on the blade surface (see Fig. 9 (a-b)). The reduced surface pressures combined with the reduced tip gap flow result in lower acoustic source terms for the axial fan with serrations. Figure 9 (c-d) shows the surface source terms from the FW-H formulation calculated from the axial microphone (M4) signals. The total source term distribution is reduced by the serrations and only in the area of the blade tip and at the valleys of the serrations, increased source terms occur. Small sound sources are formed on the leading edge serrations due to small-size vortices, which are the cause of the slightly increased sound radiation above 3 kHz.

TRANSFER TO ADVANCED AXIAL FANS

Simplified plate fans were used to understand and analyze the physical mechanisms of leading edge serrations. In the following, we will examine how the effects of leading edge serrations behave when applied to an aerodynamically optimized axial fan. The blades of the fan used are based on a NACA 4510 airfoil. The reference fan is described in the literature [1, 10]. The same leading edge modifications are used on this fan as before on the flat-plate fan. In contrast to the flat plate fans, the pressure rise is reduced due to the serrations on the optimized fans (see Fig. 10

(a)). To compare the acoustics, the speed of the serrated fan is increased so that it and the reference fan have the same operating point (total-to-static pressure rise). Figure 10 (b) shows the radiated sound pressure spectrum at a flow rate of 1.3 m³/s.

(a) Pressure on the blade surface (Ref).

(b) Pressure on the blade surface (A167λ67).

(c) Source terms FW-H of reference fan.

(d) Source terms FW-H of fan A167λ67 fan
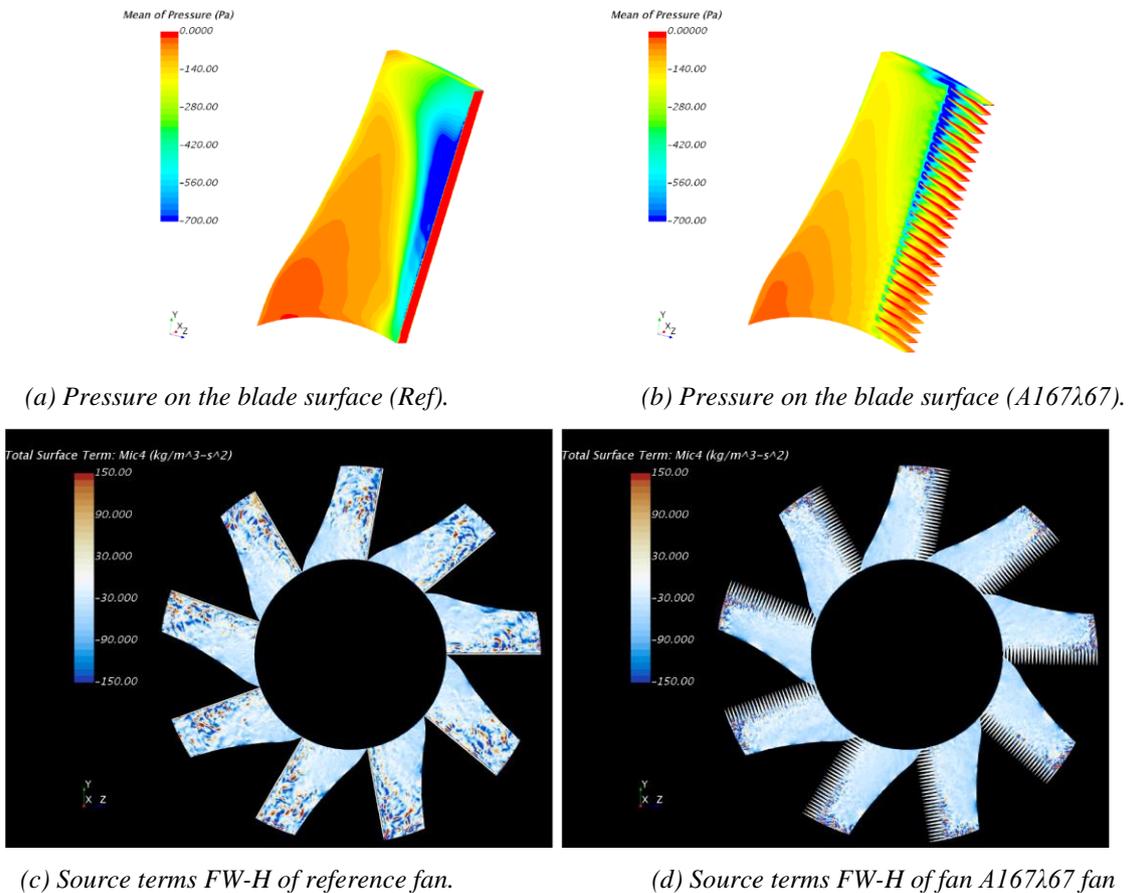
Figure 9: Pressure on the blade surface and source terms of the axial fans calculated from FW-H (time = 0.48s).

With regard to the operating point of the axial fans with the same characteristic curve, it was found that in the partial load range the serrations lead to higher sound radiation and in the design point and overload range they induce lower sound emissions. Thus, a large part of the effect of serrations is lost with optimized fans or even has a negative effect when the fans are incorrectly designed or operated. This is because with optimized fans, the flow across the airfoil is already significantly improved and the serrations can achieve almost no additional stabilization of the flow, despite the formation of vortices.

(a) Characteristic curves at same rotational speed.

(b) Sound pressure spectra at same pressure rise.
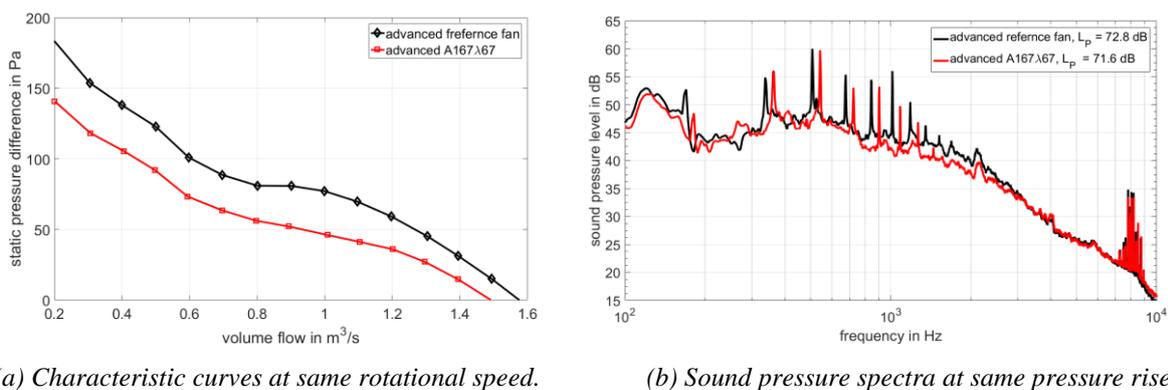
Figure 10:.Characteristic curves of the advanced fans with and without serrations at the same rotational speed and averaged sound pressure spectra of the fans at the same pressure rise (increased rotational speed of A167λ67)

CONCLUSION

Based on numerical investigations on simplified flat-plate fans, the fluid dynamic and acoustic effects of leading edge serrations could be derived. Leading edge serrations lead to a stabilization of the flow by generating counter-rotating vortices. These reduce the tip gap flow on the one hand and a radial flow over the blade on the other hand. The more stable flow results in lower surface pressure fluctuations on the blade surface. This reduces the strength of acoustic source terms on the blade surface and the sound radiation can be reduced. Once the same serration design is transferred to optimized axial fans, most of the positive effects are lost. It was found that a large part of the acoustic benefit is removed and the pressure rise of the fan decreases. The reason for this is that an optimized fan already has a stabilized flow pattern over the fan blade and the serrations can only make a small contribution to this. In this context, the prevailing flow must be accurately known in order for the serrations to provide an advantage.

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