

Sound Emissions of Axial Fans with Leading-Edge Serrations on different spanwise Locations

Czwielong, Felix¹

**Institute of Process Machinery and Systems Engineering
Friedrich-Alexander University Erlangen-Nürnberg
Cauerstr. 4, 91058 Erlangen, Germany**

Krömer, Florian²

**Institute of Process Machinery and Systems Engineering
Friedrich-Alexander University Erlangen-Nürnberg
Cauerstr. 4, 91058 Erlangen, Germany**

Becker, Stefan³

**Institute of Process Machinery and Systems Engineering
Friedrich-Alexander University Erlangen-Nürnberg
Cauerstr. 4, 91058 Erlangen, Germany**

ABSTRACT

Axial fans are used in a variety of technical systems such as air conditioners, trains, cars and ventilation systems. In these systems, axial fans are a dominant source of noise. Based on the investigations of flat-plate fans and airfoils, it has been found that serrated leading edges reduce the sound radiation compared with a straight leading edge. In this study, the sound emission of axial fans with leading-edge serrations, applied to different spanwise locations of the fan blade, was investigated. For that three fans with different leading-edge configurations (fully serrated, serrations on the outer 2/3 and 1/3 of the fan blade) and a reference fan without serration were examined experimentally. The results showed that leading-edge serrations influence the sound emissions of the axial fans. For high volume flow rates, the serrations led to a reduction of the sound radiation - the fan with serrations on the full span showed the highest sound reduction. For low volume flow rates, the sound emission was increased with the leading-edge serrations.

Keywords: Axial Fan, Leading Edge, Serration

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¹cz@ipat.fau.de

²kro@ipat.fau.de

³sb@ipat.fau.de

1. INTRODUCTION

Axial fans are used in a large number of technical systems. In most cases, these are used to generate a volume flow rate to remove heat from the system and protect it from overheating. Axial fans are used, for example, in air conditioning systems, automobiles, trains, computers and cooling circuits. Therefore, axial fans are machines that are part of our daily lives. According to current forecasts, the world demand for cooling power continues to increase, which means that the use of axial fans as part of the cooling system will also increase [1]. In addition to the positive aspect of cooling, the increased sound pressure level caused by most cooling systems is a physical and psychological disturbance for the surrounding people [2, 3]. Due to the fact that axial fans are rotating machines, they are a primary source of noise in systems and therefore a contributory cause of nuisance. For this reason, various approaches have been pursued in recent years for the acoustic optimization of axial fans with the aim of reducing the noise generated by the fan. In this context, several studies were carried out, which showed that sinusoidal serrations applied on the leading edge of the fan blades led to a reduction of the radiated sound emissions.

Based on airfoil investigations it could be shown that leading-edge serrations are an effective measure to reduce airfoil noise. This effect was more pronounced with high turbulent inflow conditions upstream the airfoil [4, 5]. Besides that, the effect of noise reduction could be observed on flat-plate fans. The sinusoidal leading-edge serrations were applied to simple plate fans without any skew. Noise reductions of up to 11 dB could be detected with free inflow conditions. A comparison of the various sinusoidal leading edge variations showed differences of up to 7 dB. The quietest variant was the leading-edge serration with the highest amplitude α_{LE} and the smallest wavelength λ_{LE} [6, 7]. The same results for the best serration could be determined by Biedermann et al. [8]. They investigated the influence of the leading-edge serrations on a ducted, unskewed low-pressure axial fans. They achieved a overall noise reduction in a regime below 2 kHz of up to 3,4 dB under various inflow conditions.

This experimental study is intended to be a continuation of Krömer's investigations [6, 7]. The leading-edge serrations are applied to the blades of low-pressure axial fans. Furthermore, the influence of different spanwise location of the serrations on the sound emission of the fan is investigated. Therefore four different axial fans are examined (one without leading-edge serration as reference, one with serrations on the outer 1/3 of the fan blade, one with serrations on the outer 2/3 of the fan blade and a fully serrated fan). In Section 2 the properties of the serrations and the fan design will be discussed. Section 3 describes the experimental setup and the measurement techniques. The results of these experimental investigations are discussed in section 4. Finally, a conclusion is given in section 5.

2. FAN DESIGN AND LEADING-EDGE PARAMETERS

All four investigated axial fans has nine unskewed fan blades and operate at a constant rotational speed of $n = 1250$ rpm. The rotational direction is clockwise. The reference fan without leading-edge serrations is shown in figure 1. The diameter of all fan hubs is $d_{\text{hub}} = 247,5$ mm and the total diameter of the whole fans is $d_{\text{fan}} = 495$ mm. The fans are operating in a duct with a diameter of $d_{\text{duct}} = 500$ mm, so the gap between the fan tip and the duct is $s_{\text{tip}} = 2,5$ mm.

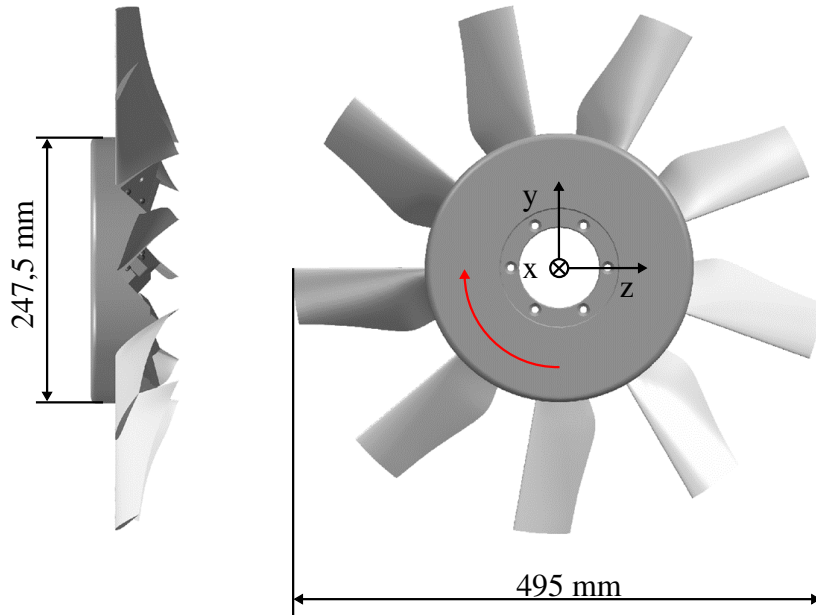


Figure 1: Schematic sketch of the reference fan without serrations on the leading edge, the rotation direction is clockwise.

The profile of the fan blades is a NACA airfoil type NACA 4510 [9]. The stacking line for the single fan blade section is in all cases the leading edge, see figure 1. Because of this, the leading edges of the fans are centric. The mean chord length of the fan blades is $\bar{l}_c = 69,6$ mm. All investigated fans were designed according to blade element theory, for more information see [7, 10].

The parameters of the leading-edge serrations were selected according to the findings of Krömer et al. [6, 7]. The three fans with serrations have all the same kind of sinusoidal serration. Figure 2 shows a sketch of the two used parameters α_{LE} and λ_{LE} .

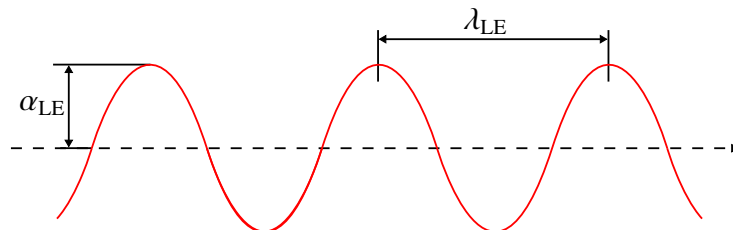


Figure 2: Parameters of the leading-edge serrations: amplitude α_{LE} and wavelength λ_{LE} .

In this study the amplitude of the sinusoidal serrations is $\alpha_{LE} = 16,7\%$ and the wavelength of the sine is $\lambda_{LE} = 6,7\%$. Thereby, both parameters are given in relation to the mean chord length \bar{l}_c of the fan blades. In figure 3 the four different fan blade geometries are shown. The different designs have the same blade surface area, but they differ in the spanwise location of the leading-edge serration. A fan blade of the reference fan without serrations (USK) is shown in figure 3 (a). The sketch (b) shows the fan blade with serrations located at the outer 1/3 of the span (USK13). The fan blade with serrations on the outer 2/3 of the span (USK23) is shown in figure 3 (c). A fully serrated fan blade (USK33) is shown in (d).

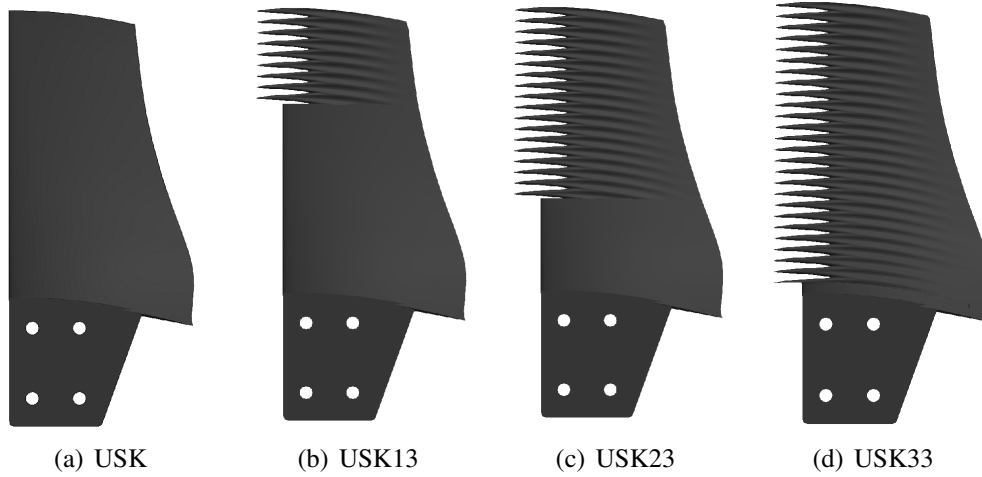


Figure 3: Schematic sketch of the investigated fan blade designs: (a) base line fan blade without serration, (b) fan blade with serration on the outer 1/3, (c) fan blade with serrations on the outer 2/3 and (d) fan blade with serrations on the whole span.

3. EXPERIMENTAL SETUP

In the following section, the aeroacoustics test bench is shown on the one hand, and the measurement technology used and the associated measurement setups are described on the other hand. The methods for the determination of the aerodynamic and acoustic characteristic of the axial fan as well as the characterization of the suction-side sound field are included.

3.1. Inlet test chamber and fan installation

Figure 4 shows the schematic design of the axial fan inlet test chamber. This test rig is designed according to ISO 5801 [11]. In order to be able to measure the sound emissions of the fans, the test chamber was built as an anechoic acoustic room. The inside of the chamber is equipped with absorbers on the walls and floor. These sound-reducing measures reduce the influence of external sound sources to the acoustic measurements inside the chamber. The quiescent sound pressure level within the test range is $\bar{L}_p = 28$ dB for the frequency range $f \in [0,1 \text{ kHz}; 10 \text{ kHz}]$. The air sucked in by the test fan is drawn in through the bellmouth inlet of the test bench. The bellmouth is standardized according to ISO 5801 [11] and is used to determine the volume flow rate \dot{V} . An auxiliary fan and a butterfly damper is located downstream of the bellmouth inlet for setting

the operating point. Before the flow reaches the acoustic chamber, the sound pressure of the inflow is reduced via a splitter-typed silencer. Inside the anechoic room a flow straightener is installed. This ensures that the flow is homogeneous and low in turbulence intensity on the suction side of the test fan. The volume of the acoustic room is $V = 22 \text{ m}^3$. A differential pressure sensor can be used to measure the difference of the total pressure to the static pressure Δp_{ts} . The sensor is connected to the pressure taps inside the chamber and to an ambient pressure sensor outside the chamber.

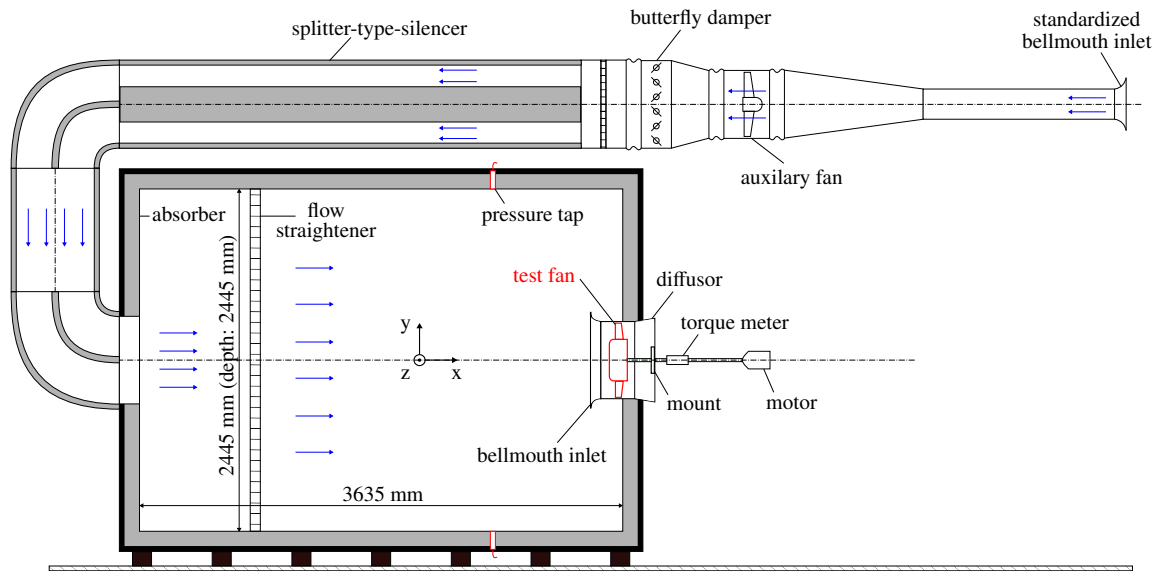


Figure 4: Standardized inlet test chamber according to ISO 5801.

For the fan support four non-centric struts were used, see figure 5. This mounting arrangement was chosen to reduce sound generation due to rotor-stator interactions.

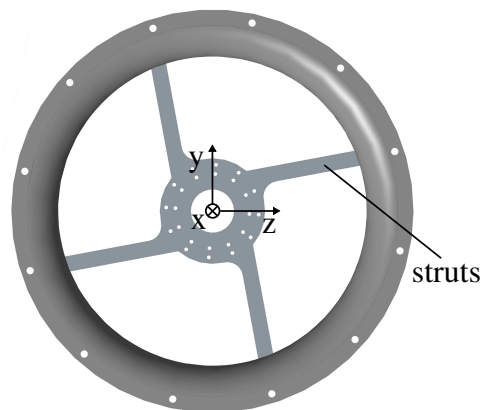


Figure 5: Sketch of the non-centric struts configuration for the fan support.

At free inflow conditions the mean turbulence intensity is $\overline{Tu} = 7 \%$ at the leading edge of the test fan for a volume flow rate of $\dot{V} = 1,4 \text{ m}^3/\text{s}$ [7]. This value for the mean turbulence intensity stays nearly constant for other volume flow rates [6].

3.2. Measurement setup for the characteristic curves

The axial fan test bench determines the total-static pressure difference Δp_{ts} from the suction side of the test fan to the ambient environment. The actual volume flow rate \dot{V} is measured at the standardized bellmouth inlet. The desired operating point is set by means of the auxiliary fan and the butterfly damper (see Figure 4). Inside the chamber the temperature T and the density ρ are measured so that the measured pressure difference Δp_{ts} can be converted to normalized pressure difference $\Delta p_{ts, \text{norm}}$ according to Equation 1. This serves a better comparability of the characteristic curves, which can be recorded under different environmental conditions.

$$\Delta p_{ts, \text{norm}} = \frac{\rho_{\text{ref}}}{\rho} \cdot \Delta p_{ts} \quad (1)$$

The reference density is $\rho_{\text{ref}} = 1,2 \text{ kg/m}^3$. A torque meter and a speed sensor are used to measure the rotational speed n and the required torque M_s of the fan. In this investigations, the rotational speed was kept constant to $n = 1250 \text{ rpm}$. Before the torque measurements, the torque offset $M_{s, \text{off}}$ was determined. This parameter contains the torque which is generated due to bearing friction. For this purpose, the torque of the drive unit without mounted fan was measured at a speed of $n = 1250 \text{ rpm}$. The torque offset $M_{s, \text{off}}$ was subtracted from the measured torque M_s with installed fan. All measured data's are recorded with a sampling frequency of $f_s = 1 \text{ kHz}$ and a measurement time of $t_m = 15 \text{ s}$. The total-static efficiency η_{ts} of the fan can be determined according to Equation 2 based on these measured data's.

$$\eta_{ts} = \frac{\dot{V} \cdot \Delta p_{ts, \text{norm}}}{M_s \cdot 2 \cdot \pi \cdot n} \quad (2)$$

3.3. Measurement setup for the sound field

To characterize the sound field, the sound pressure was measured on the suction side of the axial fans inside the anechoic chamber. Seven 1/2-inch microphones (M1-M7), from Brüel&Kjær type 4189-L-001, were used. The microphones were placed in a semicircle with a radius of $R = 1000 \text{ mm}$ around the axial test fan, see figure 6. The height of the microphones corresponds to the rotation axis of the fan. A NEXUS 2690-A from Brüel&Kjær was used as microphone amplifier. The data acquisition was done with a PXIe-1075 front-end with 24-bit PXIe-4492 data acquisition cards from National Instruments. The measurement time for the acquisition of the sound pressure was set to $t_m = 30 \text{ s}$. The sampling rate was $f_s = 48 \text{ kHz}$ resulting in a total number of measurement points of $N_s = 1440000$. The characterization of the sound field was performed at a stationary operating point simultaneously with the measurement of the aerodynamic characteristics curves. For the evaluation of the results based on the sound pressure spectra, the spectra's were energy averaged over the microphones (M1-M7). The mean sound pressure level was determined in the frequency domain of $f \in [0,1 \text{ kHz}; 10 \text{ kHz}]$.

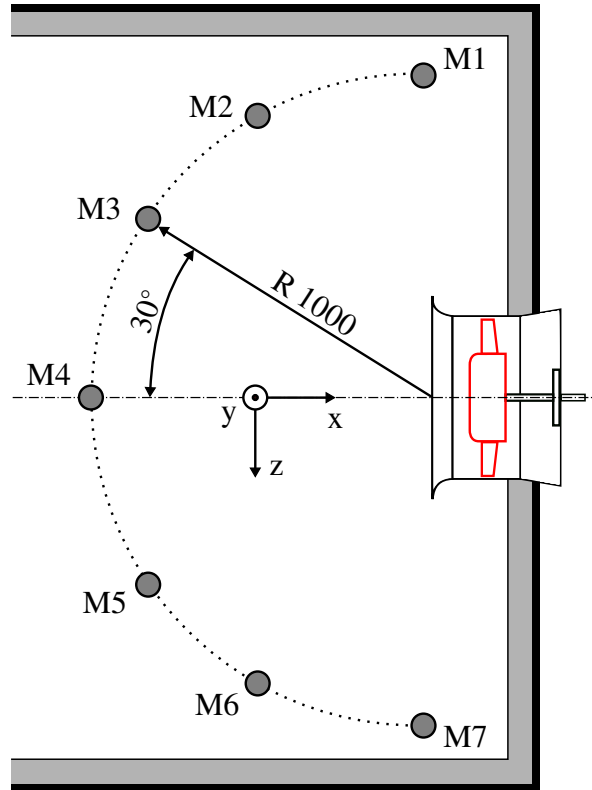


Figure 6: Measurement setup for the suction-side sound pressure of the fans, top view of the anechoic chamber.

4. RESULTS AND DISCUSSION

In the subsection 4.1 the results of the characteristic curve of the four axial fans with different spanwise location of the leading-edge serrations are discussed. In the subsection 4.2 the sound pressure spectra of the fans at four selected operating points are examined.

4.1. Aerodynamic and acoustic characteristic curves of the fans

The modification of the leading edge of axial fans not only has an effect on the acoustic sound radiation of the machines but also influences their efficiency and the generated pressure jump [12]. Figure 7 (a) shows the aerodynamic characteristic curves of the investigated axial fans. It can be clearly seen that the pressure jump decreases with the increase spanwise location of the serration of the leading edge. Thus the reference fan (USK) produces the largest pressure difference Δp_{ts} and the fully serrated fan (USK33) the smallest pressure difference Δp_{ts} . The same effect occurs for the efficiency curve (7 (b)). The highest efficiency of $\eta_{ts} = 42,5 \%$ is achieved by the reference fan (USK) at a volume flow rate of $\dot{V} = 1,3 \text{ m}^3/\text{s}$. The shape of the efficiency characteristic curves remains the same for all investigated fans with the serrations, but it sings down to lower efficiency values. Thus the fan USK13 achieves a maximum efficiency of $\eta_{ts} = 35.7 \%$, the fan USK 23 has an efficiency of $\eta_{ts} = 33.2 \%$ and the efficiency of the fully serrated fan is $\eta_{ts} = 31.5 \%$. Considering the differences in the efficiency, it can be concluded that serrations on the outer 1/3 of the fan blade has the greatest influence on the efficiency of the fan.

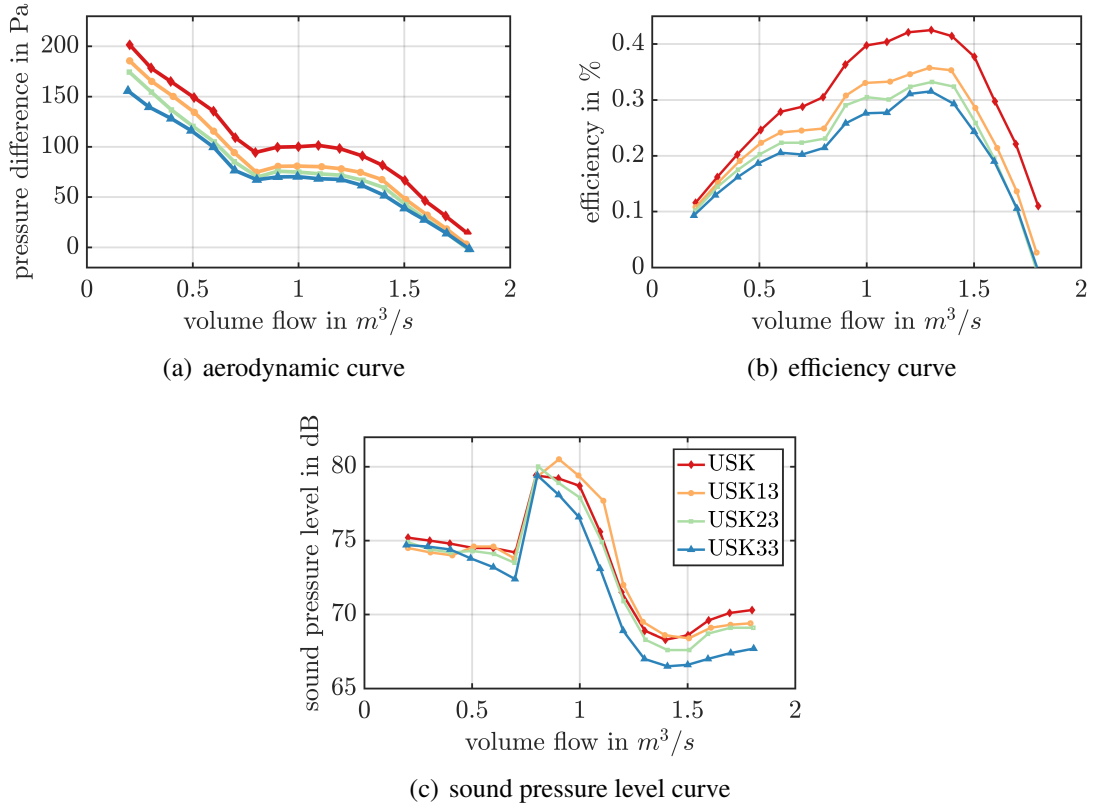
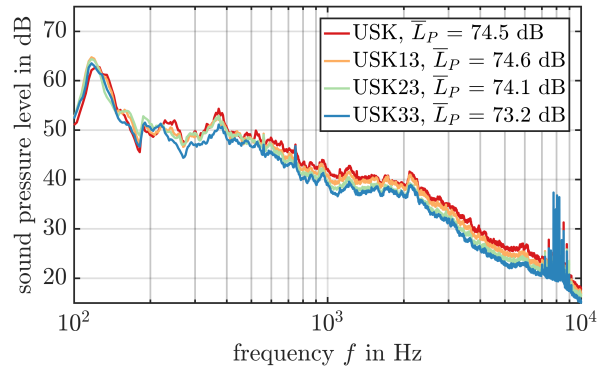


Figure 7: Characteristic curves of the investigated fans: (a) total-static aerodynamic curve, (b) total-static efficiency and (c) plot of the averaged sound pressure level at different volume flow rates.

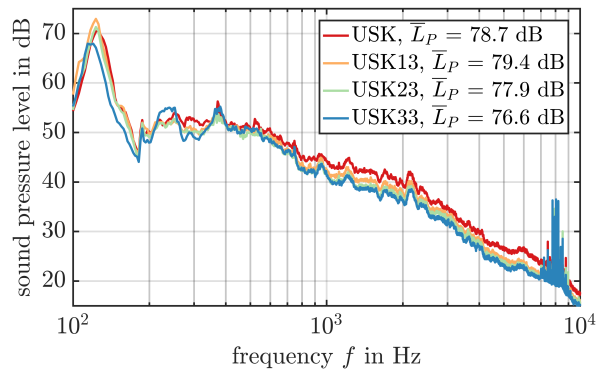
Figure 7 (c) shows the characteristic curve for the acoustic of the fans. The averaged sound pressure level \bar{L}_p of the fan USK33 is the lowest over nearly the whole range of volume flow rate \dot{V} . The minimum of the sound pressure level is reached in the range close to the highest efficiency at a volume flow rate of $\dot{V} = 1,4 m^3/s$. For the fan USK33 is the minimum $\bar{L}_p = 66,5$ dB. At the same operating point, the sound pressure level of the fan USK23 is $\Delta\bar{L}_p = 1,1$ dB higher, for the fan USK13 $\Delta\bar{L}_p = 2,1$ dB higher and for the reference fan USK $\Delta\bar{L}_p = 1,8$ dB higher. The differences show that the largest reduction in sound pressure level occurs between the two different spanwise locations of the serrations USK23 and USK33, at the operating point of $\dot{V} = 1,4 m^3/s$. Based on the characteristic curves, this means that the serrations in the inner third of the fan blades have the greatest influence on the radiated sound emissions of the low-pressure axial fans.

4.2. Sound pressure spectra

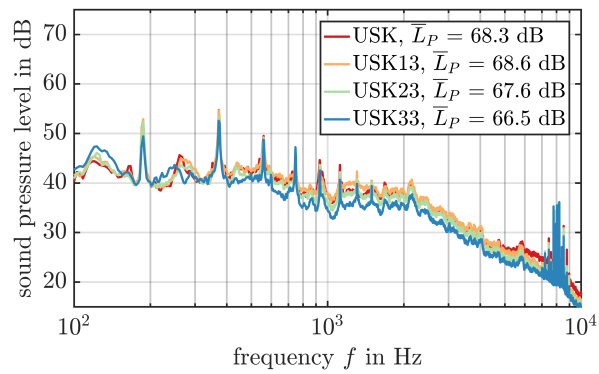
In order to get a better understanding of the effect of the different spanwise locations of the serrations, the sound pressure spectra's for different operating points are considered in the following. Figure 8 shows the averaged sound pressure spectra's \bar{L}_p for the volume flow rates (a) $\dot{V} = 0,6 m^3/s$, (b) $\dot{V} = 1,0 m^3/s$, (c) $\dot{V} = 1,4 m^3/s$ and (d) $\dot{V} = 1,8 m^3/s$.



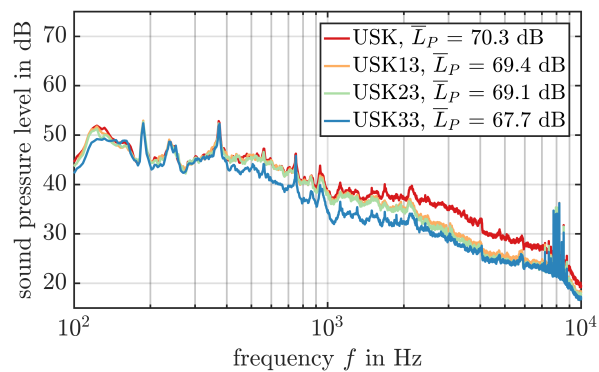
(a) $\dot{V} = 0,6 \text{ m}^3/\text{s}$



(b) $\dot{V} = 1,0 \text{ m}^3/\text{s}$



(c) $\dot{V} = 1,4 \text{ m}^3/\text{s}$



(d) $\dot{V} = 1,8 \text{ m}^3/\text{s}$

Figure 8: Averaged sound pressure spectra's for different volume flow rates \dot{V} .

The sound emissions at small volume flow rates (figure 8(a) and (b)) are determined by a narrow-band peak below the blade passing frequency $f_{\text{BPF}} = n \cdot z = 187,5$ Hz, where z is the number of fan blades (here nine fan blades). This subharmonic peak can be associated with fan tip noise caused by the backflow from the pressure side to the suction side of the fan via the tip gap [7, 13, 14]. At all operating points (figure 8 (a-d)), a clear separation of the individual sound pressure spectra starts at a frequency of $f = 400$ Hz. The frequency range up to a frequency of $f = 2$ kHz is assumed that sound generation due to turbulent inflow at the leading edge is dominant. This means that the modification of the leading edge leads to a reduced sound radiation from the leading edge of the fan. For small volume flow rates (figure 8 (a-b)), the radiated sound pressure level decreases with increasing the spanwise location of the serrations. For larger volume flow rates (8 (c-d)), only the fan USK33 has a significant difference to the reference fan USK in this frequency range. This indicates that in the overload range ($\dot{V} > 1,3$ m³/s) the serrations near the fan hub play an important role in the acoustic optimization of the fan. Frequencies greater than $f = 2$ kHz can mainly be assigned to the sound generation mechanisms in the boundary layer and the interaction between the boundary layer and the trailing edge of the fan blades [7, 15]. The sound pressure spectra's show that in this range the sound pressure level decreases with increasing the spanwise location of the serrations. This can be attributed to the fact that the leading-edge serrations generates a higher momentum exchange in the boundary layer of the fan blade. This leads to a boundary layer which is more resistant against boundary layer separation. For large volume flow rates (figure 8 (d)), in the overload range, it can be seen that the serrations near the blade tip in particular reduce the sound pressure level in this frequency range. Leading-edge serrations applied over a larger spanwise range of the fan blade only leads to a slight reduction of the sound pressure level in the frequency range from $f = 2$ kHz compared with the reduction caused only by the serrations in the outer 1/3.

5. CONCLUSIONS

In this experimental study, four different axial fans with free inflow conditions were investigated with respect to their sound radiation. The fans were modified with different leading edges. A reference fan with a straight leading edge without serrations was used. The other fans had different spanwise locations of the leading-edge serrations. Fans with serrations on the outer 1/3 and outer 2/3 of the fan blade were examined. In addition, a fully serrated fan was part of the investigations. Measurements of the suction-side sound pressure and the aerodynamic characteristics curves were carried out in a standardized axial fan test rig with an integrated anechoic chamber.

Based on the results, it could be shown that the use of leading-edge serrations reduces the aerodynamics and thus the efficiency of the fans. Especially the application of serrations in the area of the blade tip had a negative effect on the efficiency. In addition to this effect, the acoustic characteristic curves of the fans showed that from a volume flow rate of $\dot{V} = 0,8$ m³/s the fan with fully serrated leading edges produced the greatest reduction in the sound pressure level \bar{L}_p . A maximum reduction of the sound pressure level of $\Delta\bar{L}_p = 2,6$ dB could be achieved in comparison to the reference fan without leading-edge serrations.

On the basis of sound pressure spectra's, it could be determined that the sound pressure level \bar{L}_p for the frequency range $f \in [0,4 \text{ kHz}; 2 \text{ kHz}]$ decreases with the magnification of the span width on which the serrations were attached to the leading edge. This means that the serrations over the entire span (USK33) improve the sound emissions at the leading edge as well as in the boundary layer of the fan blades. In the overload range ($\dot{V} > 1,3 \text{ m}^3/\text{s}$) a reduction of the sound pressure level \bar{L}_p in the range of $f \in [0,4 \text{ kHz}; 2 \text{ kHz}]$ could only be achieved with the fully serrated fan. This indicates that in this operating range the serrations near the fan hub are of importance. However, in the overload range ($\dot{V} > 1,3 \text{ m}^3$) for the frequency range greater than $f > 2 \text{ kHz}$ the results of the sound pressure spectra's showed that most of the noise reduction takes place through serrations near the blade tip.

With this experimental study it could be determined that the use of leading-edge serrations leads to a reduction of the sound pressure level of low-pressure axial fans. However, this benefit is accompanied by a deterioration of the aerodynamic behavior of the fan. Based on these investigations fully serrated leading edges should be selected for the most silent variant of an axial fan. In order to continue these investigations and to clarify further influences of the spanwise location of the leading edges serrations, following studies should consider different inflow conditions on the one hand and on the other hand include fans, which have serrations only in the inner part of the fan blades. The change of the inflow conditions to higher and lower turbulence intensities would allow more generally statements regarding the aerodynamics and acoustics of axial fans with leading-edge serrations. The consideration of axial fans, which have leading-edge serrations only in the inner part of the fan blades, near the fan hub, would allow more precise conclusions to be made on the described acoustic phenomena.

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