SOUND SOURCE LOCALISATION AT AN AXIAL CONTRA-ROTATING FAN BY MEANS OF PIV AND ROTATIONAL BEAMFORMING

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ABSTRACT

This paper describes the investigation of a contra-rotating axial fan. It has a shroud diameter of 300 mm and a rotational speed of 1920 min\(^{-1}\) and 1200 min\(^{-1}\) for the first and the second impeller, respectively. It is designed for an air-flow rate of 0.486 m\(^3\)s\(^{-1}\) and for total pressure difference of 180 Pa (120 Pa at the first and 60 Pa at the second impeller). The main focus lies on the correlation of flow structure and the noise emission. Therefore, the flow structure is investigated by means of Particle Image Velocimetry (PIV) and the noise emission with an Acoustical Camera. PIV is established at a non-rotating model of the blades. For localizing the noise sources, a rotational beamforming algorithm is applied allowing the determination of noise sources on a rotating fan.

KEYWORDS

FAN, CONTRA-ROTATING, BEAMFORMING, NOISE, PIV

NOMENCLATURE

Symbols
\begin{align*}
\gamma & \text{ m}^2/\text{s}^2 \quad \text{specific work} \\
D & \text{ m} \quad \text{diameter} \\
\dot{V} & \text{ m}^3/\text{s} \quad \text{volume flow rate} \\
c & \text{ m/s} \quad \text{fluid velocity, absolute System} \\
c & \text{ m} \quad \text{chord length} \\
c_L & \quad \text{lift coefficient} \\
n & \text{ 1/s} \quad \text{rotational speed} \\
r & \text{ m} \quad \text{radial position} \\
s & \text{ m} \quad \text{span width} \\
t & \text{ m} \quad \text{pitch} \\
u & \text{ m/s} \quad \text{velocity in the direction of rotation} \\
w & \text{ m/s} \quad \text{fluid velocity, relative System} \\
\alpha & \quad \text{angle of attack, absolute flow angle} \\
\beta & \quad \text{relative flow angle} \\
\lambda & \text{ m} \quad \text{wavelength} \\
\gamma & \quad \text{stagger angle} \\
\varepsilon & \quad \text{glide number} \\
\Delta p & \text{ Pa} \quad \text{pressure difference, pressure rise}
\end{align*}
### Indices

<table>
<thead>
<tr>
<th>A</th>
<th>first impeller in the flow direction</th>
<th>( \cdot )</th>
<th>inlet</th>
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<tbody>
<tr>
<td>B</td>
<td>second impeller in the flow direction</td>
<td>( \cdot \cdot )</td>
<td>outlet</td>
</tr>
<tr>
<td>m</td>
<td>meridian component of the velocity</td>
<td>s</td>
<td>shroud</td>
</tr>
<tr>
<td>u</td>
<td>in the direction of rotation</td>
<td>h</td>
<td>hub</td>
</tr>
<tr>
<td>e</td>
<td>Euler, ( D_e = \sqrt{\frac{1}{2}(D_e^2 + D_h^2)} )</td>
<td>( \infty )</td>
<td>freestream</td>
</tr>
</tbody>
</table>

### INTRODUCTION

Contra-rotating axial fans have been investigated since the 1930s (Bell and eKoster, 1942, Berlage, 1936, Traupel, 1959). Due to increasing efficiency demands and power density enhancement this technology is in focus since a short time again (Mistry and Pradeep, 2014, Nouri et al., 2012, Pundhir and Sharma, 1992).

Axially blowing fans are common in many fields of application. However, it shall be noted that there is a swirl occurring at the trailing edge of the blades due to the working principle. This swirl is not required in most cases and may have unfavourable influence on subsequent devices, e.g. a higher pressure drop or a lower heat transfer coefficient. As the static pressure rise is an evaluation criterion for the fan efficiency, there are different possibilities for converting the dynamic pressure of the swirl into static pressure rise. The most common application for rising the efficiency is the installation of outlet guide vanes.

Contra-rotating axial fans (CRF) are well known as one opportunity to increase the efficiency of a fan, too. The efficiency increase is driven by the conversion of the dynamic pressure of the swirl at the trailing edge of the first axial fan blades into a static pressure at the end of the entire stage. In comparison with fans using outlet guide vanes, a higher power density can be obtained.

### WORKING PRINCIPLE OF CONTRA ROTATING FANS

Due to the Euler equation, the specific work \( Y \) is defined by

\[
Y = u''c_u'' - u'c_u'.
\]  
(1)

The peripheral velocities of an axial fan stage remains constant as \( u'' = u' = u \), which transforms Eq. (1) to

\[
Y = u(c_u' - c_u),
\]  
(2)

with \( c_u \) as the component of the fluid velocity vector \( c = c_m + c_u \) with the same direction of \( u \) and \( ' \) as inlet and \( '' \) as outlet symbols. The relative velocity \( w \) of the rotating system is defined by the Eq. (3).

\[
w = c - u,
\]  
(3)

According to Eq. (2), a pressure rise in an axial impeller stage causes a change in the outflow \( c_u'' \) compared to the inlet conditions \( c_u' = 0 \). The second impeller, with an opposite direction of rotation, shall be designed without outlet swirl as there is already a swirl at the inlet \( c_u'' < 0 \). Figure 1 represents the velocity components for a contra rotating axial fan stage without outlet swirl, defined by \( c_u'' = 0 \).
Figure 1: Schematic representation of the velocity components
The fan configuration used in this paper corresponds to the schematic drawing as presented in Figure 2. Each impeller (A, B) is driven with a separate motor. Stationary components and rotating components are differently coloured. Stationary components are presented in black and rotating components are coloured in blue and red, according to their direction of rotation.

Figure 2: Schematic representation of the fan configuration

MEASUREMENT SETUP

Test Samples
The fan under investigation has a tip diameter of 300 mm, a hub diameter of 195 mm and rotational speeds of 32 s⁻¹ (1920 min⁻¹) and 20 s⁻¹ (1200 min⁻¹) for the first and the second impeller, respectively. It is designed for an air-flow rate of 0.486 m³ s⁻¹ (1750 m³ h⁻¹) and for a total pressure differences of 180 Pa (120 Pa at the first and 60 Pa at the second impeller).

The design of the fan has been carried out using the program CFTurbo (CFTurbo, 2017). The design in CFTurbo follows a strict conceptual approach that considers 1D-balance equations, empirical correlations and airfoil theory. The geometry description is completely parametric and allows therefore adjustments of any detail. Each impeller’s boundary conditions are set according to the power distribution (120 Pa and 60 Pa) and the respective speed in the conceptual design process. The same applies to the pre-swirl that is zero for the first impeller. For the second impeller that pre-swirl is equal to the swirl produced by the first impeller. NACA 4 Digit profiles have been used for the design, i.e. NACA 6309 and 6509 for the first and second impeller respectively. From the respective polares the lift coefficients as well as the glide
numbers were taken at an angle of attack $\alpha = 5^\circ$: $c_{L/A:B} = 1.15/1$ and $\varepsilon_{A:B} = 0.014/0.011$. With blade numbers of $n_A = 7$ and $n_B = 5$ resp. the pitch for each span is defined. Using information from the velocity triangles of each span the geometry of both impellers where determined using the fundamental relation between force balance and Euler equation and stagger angle definition:

$$\frac{c}{t} = \frac{2 \cdot \gamma}{c_L \cdot w_\infty \cdot u \left(1 + \frac{\varepsilon}{\tan(\beta_\infty)}\right)}$$

(4)

$$\gamma = \alpha + \beta_\infty.$$  

(5)

Results of the described preliminary design are given below with respect to the diameter of each span:

![Graphs showing stagger angle $\gamma$, solidity $c/t$, absolute velocities $c_2$ and relative velocities $w_2$.](image)

**Figure 3: Results from preliminary design**

The result of the design are blades with straight leading and trailing edges for both impellers although sweeping of the blades would have been possible, too. This modification is not applied since some means of trailing and leading edge treatment (such as saw tooth i.e. serrations) shall be executed afterwards. These modifications and shall be done on straight edges to compare the influence more easily.

The blades haven been modified due to acoustical reasons. Those of impeller A have a modified trailing edge with serrations according to Carolus (2012), Catalano (2012) and Howe (1991). Both, the unmodified standard blades and the blades with modified trailing edge are presented in Figure 4.
The amplitude $\hat{c}$ of the serration is dependent on the mean chord length $\bar{c}$ at $D_e = \sqrt{\frac{1}{2} (D_s^2 + D_h^2)}$ with a value of 42.3 mm and was set to 0.2 $\bar{c}$. The angle of the serrations is $60^\circ$ resulting in a wavelength $\lambda$ of 13.2 mm. The serrations have been completely cut from the blade, with the tips coinciding with the trailing edge. According to Howe (1991), the sound power level shall decline with $\Delta L_W = 10 \log \left( 1 + \left( 2 \frac{\hat{c}}{\lambda} \right)^2 \right)$ dB.

Another attempt is the modification of the leading edge. Different types of those edges are discussed by Corsini and Delibra (2013), Sodermann (1972), Hansen et al (1974), Hersh et al. (2012), Polacsek et al. (2011) and Johari et al. (2007). The design study 4L of Johari et al. (2007) with a sinusoidal leading edge is applied within this study. Both types of blades for impeller B are depicted in Figure 5.

The wavelength $\lambda$ is 0.5 $\bar{c}$ and the amplitude $\hat{c}$ is 0.12 $\bar{c}$ ($\hat{c} = 5.08$ mm and $\lambda = 21.15$ mm). The unmodified leading edge cuts the wave at the point of the highest gradient. For construction purpose, the span was dived into section with distance of one quarter of the wavelength. The chord length $c$ at each section was adapted by $\Delta c = \frac{c}{2} \sin \left( \frac{2 \pi r}{\lambda} \right)$, with $r$ as the local position of the span width $s$.

These four different blades types have been permutated to four different samples. Sample 1 has unmodified blades at both impellers. Sample 2 consists of modified impeller A and unmodified impeller B and vice versa in sample 3. Both impellers in sample 4 have modified blades (see Figure 6).
Both impellers are mounted at a stand-alone drive. These drives have opposite directions of rotation and are controlled separately. Each sample has seven blades at impeller A and five blades at impeller B.

Figure 6: Design studies under examination, from left to right: sample 1 with straight edges for impeller A and B, sample 2 with serrations at the trailing edge of impeller A, sample 3 with sinusoidal leading edge of impeller B, sample 4 serrations at trailing edge of impeller A and sinusoidal leading edge of impeller B.

Sound Measurements
Localizing rotating aero-acoustic sound sources is a difficult task for a conventional beamforming algorithm, as the sound sources are moving with the rotational speed of the fan. To overcome this problem, Kerscher et al. (2017) have introduced a virtual rotation in combination with functional beamforming (Sarradj, 2012, Sarradj, 2010, Dougherty, 2014, Sijtsma, 2007). Both algorithms are used here.

As the rotational beamforming algorithm uses a virtual microphone rotation, symmetric array geometries shall be preferred. A 48-channel ring-array with a diameter of 0.75 m was used here. The rotation speeds of the impellers were recorded with two laser rpm-meter facing towards the impellers from either side of the test stand. The camera in the centre of the ring array allows the concentric alignment of the ring array with the fans axis of rotation.

The measurements are carried out as described by Krause et al. (2018) at a test stand complying with ISO 5081 (2008). Category C is selected as mode of installation, implying that there is a pipe at the suction side and a free outlet at the pressure side of the fan. Deviating to the standard, a silencer is installed in front of the fan, which is necessary for the acoustic measurements, but shall have no influence on the fan operation itself. A schematic drawing of the test stand is presented in Figure 7. In order to meet the design point of fan operation, the pressure drop is adjustable with a throttle valve. The flow rate is measured by means of the inlet nozzle. The static pressure rise of the fan corresponds with the static pressure difference at the suction side of the fan against the environment. The rectifier is installed to achieve a constant and equal velocity profile. The fan is hold by a 4-arm hub.
Flow Measurements

The measurements are carried out by using particle image velocimetry (PIV, Raffel et al., 2007). PIV calculates the displacement of illuminated particles seeded into the fluid. The illumination is performed by a light sheet produced by a laser. This particle distribution within this plane is captured with a CCD-camera. The axis-of-view of this camera is normal oriented to the illuminated plane. The displacement between two subsequent pictures is calculated by cross-correlation algorithms and is performed for all particle ensembles within an area of typically 32 x 32 px² out of a 2048 x 2048 px² sized picture.

First attempts proved that it is very difficult to measure the velocity distribution within the rotating system. On the one hand, the cylindrical housing causes distortions at the light sheet and on the other hand the illumination of the fluid around the blades and the gap between the impellers is difficult to establish. This problems led to a redesign with a not-rotating setup of the blades, as depicted in Figure 8. Both, planar PIV and stereoscopic PIV have been applied to evaluate different aspects. Figure 9 represents a measurement setup for stereoscopic PIV.

To yield an impression of the interaction between impeller A and impeller B, the blades have three degrees of freedom. Both can be shifted in angular position and location.
Figure 8: Schematic drawing of the wind-tunnel test-rig with PIV measurement at fixed blades

Figure 9: Measurement setup for stereoscopic Particle Image Velocimetry

Since these blades are not rotating, the velocity components must be adapted to a fixed system. The amount and direction of the flow in the wind-tunnel corresponds to the velocity vector in the rotating system $w'_A$. Due to mass conservation the velocity $w_y$ remains constant throughout the test section. This shall be taken into account when interpreting the velocities measured at the trailing edge of the blade. The angle between $w'_A$ and $w''_A$ by assuming $c'_{A,m} = c''_{A,m}$ leads to wanted swirl component $c''_{A,u}$. This is illustrated in Figure 10. The blade B has to be aligned thoroughly to $w''_A$. 
Figure 10: Schematic drawing of the velocity components for wind-tunnel test with non-rotating blades and conversion of the velocities

The blades for these wind channel tests are 3d-printed straight extrusion from the profile at $D_e$ with a height of 56 mm.

RESULTS

Performance measurement
The performance obtained with the different modifications is denoted in Table 1. As one may see, an influence of the different modifications on the performance at the design is not negligible. At almost the same flow rate, the achieved pressure rise differs by 8%.

Table 1: results of the performance measurement

<table>
<thead>
<tr>
<th>Sample</th>
<th>$\Delta p$ in Pa</th>
<th>$V$ in m³/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>sample 1</td>
<td>144.67</td>
<td>0.4876</td>
</tr>
<tr>
<td>sample 2</td>
<td>136.03</td>
<td>0.4870</td>
</tr>
<tr>
<td>sample 3</td>
<td>140.14</td>
<td>0.4882</td>
</tr>
<tr>
<td>sample 4</td>
<td>132.02</td>
<td>0.4857</td>
</tr>
</tbody>
</table>

Sound Measurements
The results obtained by the Acoustic Camera are illustrated in Figure 11 and Figure 12. In each figure, the images are arranged horizontally for the individual samples and vertically for the two impellers. Figure 11 presents the results for the 5 kHz third-octave band and Figure 12 for the 2 kHz third-octave band. By applying the rotational beamforming algorithm, the sound sources can be localised at impeller A and impeller B and are depicted in colours in accordance to their sound pressure level. The blade contours are highlighted by white lines.

In Figure 11, corresponding to the number of blades at impeller A, seven single aero-acoustic sources and at impeller B five single aero-acoustic sources can be found.
Impeller B exhibits nearly the same acoustic behaviour for both third octave bands of 2 kHz and 5 kHz (compare bottom rows of Figs 8 and 9). Five noise sources can be identified clearly.
Figure 12: Acoustic image with color-coded sound pressure level of the 2 kHz third octave band for the different samples (horizontally oriented) and impeller A (top row) and impeller B (bottom row)

In the next step, the mean average of the sound pressure level for the blades was calculated. Every blade segment and the corresponding level was translated to the same angle in polar coordinates and the levels have been averaged at this position.
The aero-acoustic sound sources of impeller A are always located at the trailing edges and in the middle of the span in case of sample 1 and sample 4. At sample 2 and sample 3 they are located near the centre in chord and span direction. As one may see, the noise level is attenuated by applying trailing edge serrations (samples 2 and 4). Therefore it shall be assumed that the noise of sample 3 should be the same as for sample 1. In this special case the same level is not achieved. The underlying effects have to be investigated in further experiments.

The dominant aero-acoustic sound source of impeller B is the trailing edge, for samples 1 and 2. Obviously, a sinusoidal leading edge leads to a change of the most important noise source. Contrary to samples 1 and 2 (straight leading edge), samples 3 and 4 (sinusoidal leading edge) have the highest amount of noise emission at the leading edge. At this point it cannot be distinguished whether the noise at the trailing edge is diminished in comparison to the leading edge or the noise source is shifted from the trailing to the leading edge. Furthermore, the implementation of serrations at the trailing edges at the first impeller are leading to a noise reduction of the subsequent impeller, as being visible by comparison of samples 2 and 4. The colours are indicating this noise reduction in both cases. The measured differences in the sound pressure levels between samples 1 and 2 as well as between samples 3 and 4 is in the range of 1 dB for impeller B.
Contrary to the observations at 5 kHz, no specific noise sources could be detected on impeller A. The acoustic image shows a circular sound source. The noise emission of impeller A is probably significantly lower than those of impeller B. Hence, the noise emission of impeller B masks the noise sources from impeller A. These sources from impeller B appear “smeared” in the acoustic pictures since the rotational filter applied with the speed and direction of impeller A, while they actually belong to impeller B. As at 5 kHz the noise at impeller moves to leading edge when the modification is applied.

The differences in the sound pressure levels for the third octave frequencies of samples 2, 3 and 4 in relation to sample 1 are presented in Figure 15. Although there are small differences, there is a tendency of lower noise emission for frequencies higher than 800 Hz. The highest noise reduction is performed by sample 4 at high frequencies. Low frequency noise reduction instead can be performed by sample 2. The blade passing frequencies are 224 Hz and 100 Hz for impeller A and impeller B, respectively. The noise reduction at 250 Hz third-octave band corresponds the blade passing frequency of impeller A.

Figure 14: Segment-wise averaged acoustic image with color-coded sound pressure level of the 2 kHz third octave band for the different samples (horizontally oriented) and impeller A (top row) and impeller B (bottom row)
Flow Measurements

The flow measurements with laser-optical methods are still under investigation. The results will be presented in the oral presentation at the conference. One impression of the measurement results is presented in Figure 16 which illustrates the magnitude of the velocity in the range of 0 and 35 m/s.

Figure 16: PIV-measurement at stationary blades

Measurement Uncertainty

The uncertainty of the performed measurements depend among other influences on the used beamforming algorithm. Sarradj et al (2010) evaluated the influence of different formulations of the steering vectors on the source strength and position for a three-dimensional scenario. This of course concerns the absolute source strength. In the present case the relative certainty of the measurements are of higher interest. Due to the stable rotational speed of the fans the sound power level emitted by the impellers is stable, too. Furthermore the measurement conditions (test stand, camera position, working point etc.) can be reproduced reliable. Consequently one may assume that the evaluated process is highly constant, though stochastic. To avoid further deviations the integration interval for the source evaluation has been chosen quite long (32 s). This leads to a reliable averaging of the sound field within this time period and therefore a low measurement uncertainty when comparing different types if impellers.
To yield an impression of the uncertainty level the complete measurement signal of 32 s for a sample measurement has been evaluated at 2 s-intervals. For each interval the source strength was calculated using rotational beamforming. The results are presented in Figure 17.

Figure 17: analysis of a 32 s-sample measurement at 2 s-intervals in comparison with the average of this intervals and the overall average.

The source strength within the complete 32 s-sample interval is calculated with 47.30 dB. The average of all 16 2 s-intervals is 47.35 dB with a standard deviation of 0.09 dB. This difference might be caused by averaging effects between the 2 s-intervals. Based on this analysis, a repeatability uncertainty of 0.1 dB is assumed.

The measurement uncertainty $\Delta c$ of the PIV depends on the uncertainty of a measured way $\Delta x$ in a measured time $\tau$. The velocity is defined by $c = \frac{x}{\tau}$. Using the Gaussian law of error propagation the uncertainty is defined by $\Delta c = \frac{\partial c}{\partial x} \Delta x + \frac{\partial c}{\partial \tau} \Delta \tau = \frac{1}{\tau} \Delta x + \frac{x}{\tau^2} \Delta \tau$. The resolution error of the camera is typically one pixel with a scaling factor of $9.8 \times 10^{-5} \text{ m pix}^{-1}$. The time between pulses is about 100 $\mu$s with an error of 1 ns. This results in a uncertainty of 0.5 %.

**CONCLUSIONS**

Investigations of the noise emissions and the flow structures at a contra-rotating axial fan have been presented here.

A permutation of standard blades, serrations at the trailing edge of the first impeller and sinusoidal modifications at the leading edge of the second impeller has been designed and investigated by an Acoustical Camera and particle image velocimetry. In general, the application of modifications may lead to a change of noise source locations and measurable sound pressure reductions. The correlation between flow structure around the couple of two blades and the noise emission is visible but still requires further investigation for a deeper understanding.

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