Cross Wind Influence on the Flow Field and Blade Vibration of an Axial Fan

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Abstract In the operation of air cooled condensers in power plants, axial ventilators are often subject to natural wind. It has been found that this additional cross flow at the fan inlet may have a significant impact on the effective heat exchange at the condenser. Therefore, it was investigated how the resulting velocity profile at the fan outlet would be influenced by cross flow. Laser Doppler Anemometry measurements were performed in the ducted flow field produced by an axial fan at its outlet, with the open fan inlet subject to a wind tunnel flow field. At different fan flow rates, the test fan was approached by a 10m/s ambient wind, perpendicular to its rotation axis. It could be shown how the cross flow induced an asymmetry to the resulting mean velocity profile behind the fan. This was caused by the altered relative velocities at the fan blades, due to the overlap of cross flow and blade motion. The flow field's turbulence intensity was amplified and became asymmetric, too. Magnitude and nature of both effects were found to depend on the fan operating point and the local stability of the flow field around the blade. In order to estimate the influence of cross wind to blade vibration, co-rotating Laser Scanning Vibrometry measurements were performed at two distinct flow rates. Overall magnitude and spectral resolution of the surface velocities were very alike inside a quiescent environment at both operating points. Under the influence of cross flow at the fan inlet, blade vibration showed a strong excitation for all rotational frequency harmonics at the low fan flow rate. The overall amplification at the higher flow rate was smaller, but it also showed a significant increase of the response of the blade passing frequency. In a comparison to microphone measurements, the increased blade tip vibration at blade passing frequency showed a correlation with the intensification of the sound pressure level at the same frequency.

1. Introduction

Within the scope of the EU funded MACCSol project (MACCSol 2014), a novel design of a modular air cooled condenser (ACC) is being developed. It consists of several axial fans operating in an open air environment, where natural cross wind at the fan’s inlet may have a significant impact on fan performance. Within the scope of this work, the authors wish to present the influence of cross flow at the fan inlet onto the behavior of the fan in laboratory conditions. For this, a fan test rig was built inside a wind tunnel environment. Laser Doppler Anemometry (LDA) measurements were used to investigate the changes in the velocity distribution behind the fan outlet. This is where the heat from the condenser tube bundles is supposed to be removed in the target technical application, so the flow field at this location is of great interest for the condenser efficiency, see Walsh et al. (2012). Secondly, the fan blade motion was captured using Laser Scanning Vibrometry (LSV) with a de rotator, a co-rotating optical device. Apart from evoking additional strain in the rotor, it may help to understand to which degree additional fan noise may be attributed to blade vibration. Results from microphone measurements are therefore shown as a comparison.

Based on general heat exchanger behavior, the equality of velocity distribution over the tube bundles is important to a high ACC efficiency (Walsh 2012). The minimized heat strain on the fan components and the additional swirl component at the fan exit suggest an operation in forced draft mode, i.e. mounting the ACC to the fan outlet, as investigated by Moore et al. (2011). Generally, axial ventilators can be designed to equalize the radial distribution of axial velocity at the fan outlet (see Aungier 2006). Under the premise of an undistorted flow field at the fan inlet, it is valid to assume an axisymmetric velocity distribution at the fan outlet. With the presence of an ambient cross flow, i.e. directed perpendicularly to the fan axis, the inflow conditions are no longer undisturbed or axisymmetric. The aim of this work is to investigate, to which degree
and in which exact way the flow field at the fan outlet may be distorted by such influence. The need for such investigation is evident from research by Maulbetsch et al. (2011), who found ACC fan flow rate reductions at an operating power plant in Nevada, USA, to be reduced by up to 60% by ambient wind. In his PhD thesis, Thiart (1990) already investigated the effects of cross wind to the fan flow field at one operating point, using a five-hole total pressure probe inside the duct at the fan outlet. He found an introduction of asymmetry to the mean flow field, and he was able to reproduce these effects with computational methods. He also found the fan static efficiency not to be influenced by the cross wind, even though more power was necessary to impose the same fan flow rate. Nevertheless, his experimental setup was limited to this configuration and no examinations to the turbulence intensity of the flow field were possible.

Numerical investigation about the effects of distortions in the inlet flow field was carried out by Meyer (2005). He describes how the formation of inlet vortices is to be expected under cross flow, which was induced from multiple neighboring fans arranged in an array. The asymmetry of the flow field and additional vortex formation may impose an additional load to the blades, as predicted by Thiart (1990). LSV investigation of the blade surface was used in this work to evaluate the resulting vibration of the fan.

The fan blade surface vibration may also contribute to the generation of sound. Blake (1986) explains the interaction of solid body, flow field and noise generation. Increased turbulence at the blade leading edge is expected to induce low frequency noise. Additional blade vibration in the same frequency could contribute as an additional source of sound. Results from surface vibration measurements are therefore also compared to microphone measurements of the fan operating under cross wind influence.

![Figure 1: CAD Sketch of wind tunnel setup (0° configuration)](image)

2. Experimental Setup

To investigate the impact of wind on the fan behavior, a fan test rig was designed and built inside a wind tunnel environment at the University of Erlangen-Nuremberg (FAU). The basic concept was based on an open inlet ducted outlet fan test rig, as prescribed in ISO 5801 (2007), set up inside the open test section of the 1.87×1.40 m² nozzle Göttinger Type wind tunnel at FAU. Its nozzle contraction was 5:1 and the wind tunnel’s turbulence intensity was approximately 0.4%. The design was aimed at a minimization of wind tunnel blockage in the test rig, as well as to being able to realize different on-flow angles φ between wind direction and fan axis. Fig. 1 shows a sketch of the wind tunnel fan test rig. The test fan was mounted with its pressure side to a straight square duct with a $D \times D = (300 \text{ mm})^2$ cross section and a length of $l = 4.5 \cdot D$ before a 180 degree bend downstream. The fan flow was then guided through the test section floor and into a settling chamber, where the flow rate was measured with multiple nozzles. Auxiliary fans helped to overcome the system pressure losses. Fig. 2 shows the test fan in $\phi = 0^\circ$ configuration. For the
\( \varphi = 90^\circ \) configuration, which was examined within the scope of this work with wind tunnel flow velocity \( u_{\text{wind}} = 10 \text{ m/s} \), the duct with the test fan was rotated within the test section. For the measurement of fan pressure, an array of Kiel probes, as described by Heinemann et al. (2014), could be mounted \( D \) downstream from the fan outlet inside the duct, with a coarse flow straightening device in between to ensure tolerable deviations of the local flow direction to the probe axes.

**Test Fan Configuration**

The experimental investigations were carried out on a commercially available, forward skewed five blade axial ventilator with diameter \( D = 300 \text{ mm} \). Its geometry can be seen in Fig. 2. Due to different requirements caused by the measurement technologies, it was operational in different configurations during LDA and LSV measurements.

![Fig. 2: Test fan geometry](image)

LDA tests were performed with a guard grille at the fan inlet, in accordance to target application requirements. This configuration is shown in the sketch on the right in Fig. 2, with the air flow in direction “A”. Differing from that, LSV tests were carried out without the guard grille, and in reversed flow direction through the nozzle, i.e. with operating flow direction “V”. An estimation of the discrepancy caused by the different operating conditions can be made with the help from Fig. 3. There we see the fan total to static pressure coefficient \( \Psi_{ts} \)

\[
\Psi_{ts} = \frac{2 \Delta p_{ts}}{\rho (D \pi n)^2}
\]

(1)

and fan efficiency \( \eta_{ts} \)

\[
\eta_{ts} = \frac{\dot{V} \Delta p_{ts}}{P_{in}}
\]

(2)

over flow rate coefficient \( \Phi \)

\[
\Phi = \frac{4 \dot{V}}{\pi D n (D^2 - D_{f}^2)}
\]

(3)

with data captured from a standard fan test rig at FAU. The fan was mounted to a 1D short duct section in an otherwise open inlet to open outlet fan test rig in accordance to ISO 5801:2007. Here, \( \rho \) is the air density, \( \dot{V} \) the fan flow rate, and \( \Delta p_{ts} \) the difference in pressure between the open outlet (environmental static pressure), and the chamber in which the fan inlet was located. The fan’s rotation speed was \( n \) and its hub diameter \( D_{f} = 0.3 \cdot D \). The electric power input to the fan is labelled \( P_{in} \). In Fig. 3, the data set labelled “LDA config” refers to the fan configuration used during LDA measurements, without flow straightener, with guard grille and nozzle direction “A”. The fan configuration with no guard grille, flow straightener downstream from fan, operating direction “V” is labelled “LSV config”, since it was used during LSV measurements.

Fig. 3 shows clearly how the different operating direction in combination with the flow straightener decreased fan pressure and efficiency for flow rates \( \Phi \geq 0.14 \). The extra obstacle from the guard grille at the fan inlet seems to induce comparably little negative influence to the fan performance. Nonetheless, the
experiments are in both configurations valid to illustrate the influence of cross wind at the fan inlet, since the results are presented independently. LDA and LSV measurements each may help to understand the influence of cross wind to axial fans.

![Graph](image)

**Fig. 3:** Static fan pressure coefficient $\Psi_{ts}$ and efficiency $\eta_{ts}$ for LDA and LSV fan test configuration

**LDA measurement setup**

LDA measurements were used to capture the ducted outlet flow field behind the fan. The cross section inside the duct $0.57 \cdot D$ downstream (i.e. in $x$-direction) from the fan outlet was probed at $9 \times 9$ locations, equally spaced $0.1 \cdot D$ from one another. It is to be seen in Fig. 4 that the respective duct section was replaced with glass screens to enable optical access. As illustrated, LDA results are presented in contour plots, with the view in opposite to the axial flow direction. Fan rotation is clockwise and the cross wind is effective from the right.

The two component LDA system was designed in backscatter orientation, with a beam from a 5 W argon-ion Spectra Physics Model 2060. Beam splitting, frequency shifting, signal processing and analysis were performed with DANTEC technology. The laser wave lengths used were 514.5 nm and 488 nm, emerging from a probe attached to a traversing system (see Fig. 4). To capture the third velocity component, the probe lens could be rotated by $90^\circ$ to scan the test section from above and capture the velocity component in $y$-direction.

**LSV measurement setup**

The LSV measurements were performed using a Polytec PSV-500 Scanning Vibrometer with a co-rotating optical device (“derotator”). It was synchronized with the signal from the rotary encoder of the test fan, to capture one surface point motion after another. Fig. 5 shows the test setup. For practical reasons, the duct with the test fan had to face the opposite direction compared to the LDA test configuration, nevertheless exposing the fan to a $\varphi = 90^\circ$ cross wind at its inlet. To enable optical access from the front, the fan guard grille was removed. A picture of the fan operating during measurement in shown in the top left of Fig. 5. The trace from the co-rotating laser beam is clearly visible. To capture the fan pressure for the LSV configuration, the Kiel probe array was mounted inside the duct at $D$ downstream from the fan outlet, together with the flow straightener (shown at the top center of Fig. 5) in between them.
Using a sampling frequency of 12.5 kHz and 30 times 640 ms of measuring time on each scan point (averaging the 30 samples with 70% overlap), signal frequencies up to 5 kHz were computed with a resolution of $\Delta f = 1.5625$ Hz. Distributed equally over the five fan blades, 400 scan points were probed.

**Fig. 4:** LDA measurements of the flow field inside the duct at the fan outlet inside a wind tunnel environment

**Fig. 5:** Test setup using LSV with a derotator on the fan blades' surface inside the wind tunnel

**Acoustic Measurements**

Acoustic measurements in a wind tunnel environment could not be carried out inside the facilities at FAU, so the fan test rig was adapted to the configuration shown in Fig. 6. This test setup was realized inside the aeroacoustic wind tunnel of BMW in Munich. The respective fan configuration was analogue to the LSV setup, i.e. air flow direction “V” was used without the presence of a guard grille, and the flow straightener was used behind the fan outlet to decrease the swirl before the Kiel probe array. Deviating from the experimental setup shown in Fig. 1 and Fig. 5, the quadratic duct section did not bend downstream, but merged into a circular pipe, which lead into the settling chamber.

As to be seen in Fig. 6, a 0.5” free field microphone was placed 10 $D$ away, straight in front of the test fan
inlet. It was located well outside of the wind tunnel flow section. To avoid any sound sources but the test fan itself, no auxiliary fans were used. Therefore, no large $\Phi$ could be captured.

![Wind Tunnel Setup](image)

**Fig. 6:** Fan test setup in aeroacoustic wind tunnel for microphones measurements

### 3. Cross Wind Influence on the Flow Field at the Fan Outlet

As described in Section 2, the flow field was scanned $0.57 \cdot D$ downstream from the fan in a cross section inside the duct at $9 \times 9$ points. A velocity component individual $C_i$ can be split into a mean value $c$ and a fluctuating value $c'$, as

$$C_i = \frac{1}{N} \sum_{k=1}^{N} C_{ik} + \frac{1}{N} \sum_{k=1}^{N} C_{ik} = c + (C_{i} - c) = \bar{C} + c'_i = c + c'_i$$

(4)

The average flow field inside the cross section is shown in Fig. 7. The contour plot represents the mean velocity component in axial direction $c_m$, scaled by the flow rate $\bar{V}$ and duct dimensions $D \cdot D$. Arrows in Fig. 7 illustrate the magnitude and direction of the velocity components inside the plane, i.e.

$$c_\theta \cdot \bar{e}_\theta + c_r \cdot \bar{e}_r = c_y \cdot \bar{e}_y + c_z \cdot \bar{e}_z$$

(5)

also scaled by the averaged duct velocity $\bar{V}/D^2$.

The left hand side plots contain the results from reference measurements without wind influence, and the effect of wind at the same $\Psi$ can be seen in the right figure. Three distinct operating points are shown. As compared to ‘LDA config’ data in Fig. 3, the first reference data set at $\Phi = 0.184$ was located in the unstable operating range, under low efficiency. It is shown in Fig. 7 (a) and (b). The second reference data set at $\Phi = 0.274$ in Fig. 7 (c) and (d) refers to an operating condition close to the optimum, where $\eta_m(\Phi)$ is at its peak. The last configuration (Fig. 7 (e) and (f)) refers to a very high flow rate and low fan pressure, well beyond the point of maximum static fan efficiency.

As to be expected, the results from experiments without wind show high axial symmetry in all velocity components. The fan rotated clockwise upstream from the plot plane, and so does the flow field. It can easily be seen that the circumferential velocity component $c_\theta$ decreased with $\Phi$ in accordance to fan theory, where fan pressure is proportional to $c_\theta$, as the additional energy in the flow is introduced by an increase in angular momentum. For an axial fan with ideal irrotational inflow, it is

$$\Delta p_{ts} \sim c_\theta \cdot \pi \cdot D \cdot n$$

(6)
Also in accordance to fan theory, radial velocity components $c_r$ appear to be negligibly small.

The radial distribution of the circumferential velocities $c_\theta$ seems to be quite uniform for all captured operating conditions without cross flow influence. The axial velocity component was also quite constant over the radius, with a zone of lower velocity or even stagnation in the wake of the fan hub, located in the center of the contour plots in Fig. 7. It could be observed, how the wake of the hub became a lot smaller with increasing flow rate, equalizing the $c_m$ distribution even more.

![Fig. 7: Distribution of mean axial velocity $c_m$ (contour) and mean plane velocity components (vector field) over duct cross section with and without cross flow at the fan inlet](image)

The right hand plots of Fig. 7 represent the influence of 10 m/s cross flow effective from the right side in the view plane, to the velocity distribution at the fan outlet. While keeping the fan test rig loss coefficient steady, the fan flow rate coefficient $\Phi$ was decreased a bit by the cross wind influence. In the plots with effective cross flow at the fan inlet, it can still be found that the magnitude of the mean velocities in the cross plane $c_\theta \cdot \hat{e}_\theta + c_r \cdot \hat{e}_r$ decreased with $\Phi$, and that $c_m$ distributed more uniformly with $\Phi$. But the graphics show a significant loss of axis symmetry in the flow field. The axial velocity distribution in Fig. 7 (b) shows a distinct shift of the hub wake region in wind direction, presumably resulting from the overlap of axial acceleration from the fan blades and initial momentum in wind direction at the fan inlet. The shift of the wake in wind direction became less prominent with increasing flow rate (Fig. 7 (d).
Another asymmetric feature in the flow field is the increased axial acceleration to be found in the upper regions of the plots \((z/D > 0)\). In this region, the direction of blade motion is opposite to the approaching ambient flow. The kinematic effect is illustrated in the right sketch of Fig. 8. There, \(u_{\text{wind}}\) is in opposite direction to the blade motion \(2\pi nr\), such that the relative velocity \(\vec{w}_1\) approaching the blade is increased.

Aerodynamically, this will introduce more angular momentum into the fluid. This locally increases the energy in the fluid, partially converting into greater \(c_m\). This effect is especially prominent for \(\Phi = 0.260\). The addition of blade motion and cross flow presumably leaves the blade operating in a stable and effective manner. At the lower flow rate in Fig. 7 (b), the mean axial velocity was influenced less, presumably due to low aerodynamic efficiency in the reference configuration to begin with. For \(\Phi = 0.314\) (Fig. 7 (f)), the intensification of \(c_m\) in the area of blade counter-motion \((z/D > 0)\) was less intense. Presuming similar aerodynamic behavior of the fan blades, the increase in relative velocity approaching the fan blades \(\vec{w}_1 = |\vec{w}_1|\) will become weaker with increasing \(c_m\), as

\[
\frac{\partial}{\partial c_m} \left( \frac{(2\pi nr - u_{\text{wind}})^2 + c_m^2}{(2\pi nr)^2 + c_m^2} \right) = 2 \cdot u_{\text{wind}} c_m \frac{2(2\pi nr - u_{\text{wind}})^2 + c_m^2}{(2\pi nr)^2 + c_m^2} \tag{7}
\]

becomes negative if \(u_{\text{wind}}\) opposes the blade motion \(2\pi nr\) at radius \(r\), i.e. \(u_{\text{wind}} < 0\) in Eq. (7).

The opposite phenomenon can be found in the lower half of the plots, where the effect of a decreased relative velocity at the blades shows in the downstream flow field. The kinematic reduction of \(\vec{w}_1\) is illustrated in the left sketch of Fig. 8. Comparing Fig. 7 (d) and (f), the effect shifted towards a small wake region in the lower right corner of the cross section.

The vectors indicating the velocity components in the plane \(c_\theta \cdot \vec{c}_\theta + c_r \cdot \vec{c}_r\) also reflect the local relation of wind velocity against blade motion. In the upper half of the domain, a reduction of horizontal velocities can be found, since the wind momentum opposes the effective momentum induced by the blade. The opposite effect can be found in the lower half, where cross plane velocity components appear unchanged or even slightly increased in comparison to the uninfluenced plots on the left.

The reaction of the fluctuating velocity components \(c'\) to cross wind influence is show in Fig. 9 at the different operating points. The contour plots show the local turbulence intensity \(Tu\), which is the mean square deviatory velocity scaled by the total mean velocity,

\[
Tu = \sqrt{\frac{\frac{1}{3}(c_\theta'^2 + c_r'^2 + c_r'^2)}{c_m^2 + c_\theta^2 + c_r^2}} \tag{8}
\]

where the bar over the fluctuating components denotes sample averaging, i.e.

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1 In Fig. 8, the variables with a prime do not represent fluctuating values. They are velocities altered by wind influence.
\[ \overline{c'^2} = \frac{1}{N} \sum_{i=1}^{N} c_i'^2 = \frac{1}{N} \sum_{i=1}^{N} (C_i - c)^2 = \frac{1}{N} \sum_{i=1}^{N} (C_i - \frac{1}{N} \sum_{j=1}^{N} C_j)^2 \]  

(9)

The turbulence intensity for the resulting flow field without wind influence at the fan inlet (left plots of Fig. 9) shows an axis symmetric distribution for all three flow rates. It is interesting to notice, how \( Tu \) is more uniformly distributed over the radius for small \( \Phi \), where the fan swirl is more prominent. With increasing flow rate, the turbulence intensities formed a ring shaped area of lower values, while remaining high in the hub wake region. Less turbulence was produced from the fan swirl, which was weaker, but the hub wake remained to create high values \( Tu \) in the centers of Fig. 9 (c) and (e).

Cross wind at the fan inlet induced more turbulence to the flow. Comparing the distribution of \( Tu \) with and without wind influence for low \( \Phi \) in Fig. 9 (a) and (b), there is an increase at \( z/D > 0 \), where cross flow opposed blade motion. It is presumed, that the instable blade flow was additionally distorted by the cross wind, intensifying the detachment of turbulent vortices from the blade. The large value at the center node was decreased by the higher mean velocity in the hub wake, caused by greater velocities in the cross plane of Fig. 8 (b) at \( y/D = z/D = 0 \). This effect is also apparent in Fig. 9 (d) and (f).

![Fig. 9: Distribution of turbulence intensity \( Tu \) over duct cross section with and without cross wind](image-url)

With greater \( \Phi \) (Fig. 9 (d) and (f)), opposing wind to blade motion did not increase the fluctuating velocities \( c' \) as much as the mean velocities, such that the resulting turbulence intensities remained as low as in the
reference configuration where \( z/D > 0, y/D < 0.1 \). At greater angles, where \( z/D > 0, y/D > 0.1 \) and \( z/D < 0 \), the resulting flow field did show larger values \( Tu \). This may indicate the relative position of the blade with a more unstable, detaching flow, which was caused by the cross flow at the fan inlet.

4. Cross Wind Influence to Blade Vibration

The influence of cross wind at the fan inlet to blade surface vibration was analyzed using the LSV test setup described in Section 2. A total of four configurations is presented here, as the product of two operating points \( \Phi_1 \approx 0.17 \) and \( \Phi_2 \approx 0.28 \), and the absence or presence of cross wind at the fan inlet. Time signals of the surface velocities \( u_i \) were captured for 30 times 640 ms on each scan point \( i = 1 \ldots 400 \). Results were Fast Fourier transformed and averaged to the frequency domain with \( \Delta f = 1.5625 \) Hz.

Table 1 shows the configurations and their characteristics. Total fan pressure coefficient \( \Psi_{tt} \) and efficiency \( \eta_{tt} \) refer to the difference in mean total pressure downstream from the fan, measured with the Kiel probe array, to ambient pressure. The root mean square value of the surface velocities \( u_{rms} \) computes in accordance to Eq. (9) and measures the total magnitude of vibration over all frequencies.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>( \Phi )</th>
<th>( \Psi_{tt} )</th>
<th>( \eta_{tt} )</th>
<th>( f_0/Hz )</th>
<th>( u_{rms}/(m/s) )</th>
<th>( 20 \cdot \log_{10} (u_{rms} \cdot m/s) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low ( \Phi ), no wind</td>
<td>0.166</td>
<td>0.148</td>
<td>0.220</td>
<td>43.27</td>
<td>0.623</td>
<td>-4.111</td>
</tr>
<tr>
<td>Low ( \Phi ), 10m/s wind</td>
<td>0.172</td>
<td>0.111</td>
<td>0.130</td>
<td>41.13</td>
<td>0.719</td>
<td>-2.865</td>
</tr>
<tr>
<td>High ( \Phi ), no wind</td>
<td>0.277</td>
<td>0.091</td>
<td>0.261</td>
<td>44.18</td>
<td>0.622</td>
<td>-4.124</td>
</tr>
<tr>
<td>High ( \Phi ), 10m/s wind</td>
<td>0.286</td>
<td>0.015</td>
<td>0.040</td>
<td>43.42</td>
<td>0.676</td>
<td>-3.401</td>
</tr>
</tbody>
</table>

Table 1: Configurations used in LSV measurements

Table 1 shows that both \( \Phi \) without wind influence showed a very similar overall response in \( u_{rms} \). This is very interesting to notice, since the two operating points differed significantly in their position on the characteristic fan curve (‘LSV config’ in Fig. 3). \( \Phi = 0.166 \) lay in an unstable operating domain, while \( \Phi = 0.277 \) was located close to maximum efficiency. Nevertheless, the magnitude of the overall blade vibration was approximately the same. The introduction of cross flow to the fan inlet led to a significant excitation in blade vibration at both operating points. The averaged root mean square surface velocity over all captured points \( u_{rms} \) was increased by more than 15% at \( \Phi = 0.172 \), and 8.7% at \( \Phi = 0.286 \). Obviously, cross wind induced more instability to the fan at the low flow rate operating point than to the higher one, and it showed a lot greater influence than a mere change in fan pressure / fan flow rate.

Fig. 10 and Fig. 11 show the mean amplitude spectrum of the signal over all 400 scan points as a function of the frequency \( f \). Due to the wind influence on the rotation speed, the representation of the data is scaled to the respective fan rotation frequency of the test configuration \( f_0 \). The ordinate shows the logarithmic presentation of the surface velocity \( u \), scaled to 1 \( m/s \). The diagrams show the low frequency domain of the spectrum, where the greatest amplitudes in vibration were to be found. Fig. 12 to Fig. 19 show the distribution of the vibration amplitude over the fan blades in logarithmic root mean square values around the first, second, fifth and sixth harmonics for the low \( \Phi \) and high \( \Phi \) measurements, comparing the reference results on the left with the ones under wind influence on the right.

Fig. 10 illustrates the influence of cross wind to the fan blade vibration at a low flow rate. The reference data set (without wind influence, blue line) shows the highest amplitudes at the harmonic frequencies of \( f_0 \). The amplitudes decreased from \( 1 \cdot f_0 \) down to \( 4 \cdot f_0 \), with a slight increase at the blade passing frequency (BPF) \( 5 \cdot f_0 \). A non-harmonic excitation was found between the fourth and fifth harmonic, around 180 Hz.

The surface velocity spectrum under cross wind influence looks very similar in general. Distinct differences were found at \( 2 \cdot f_0 \), where the mean \( u \) over the entire fan was a lot larger under wind influence. While the first and the third harmonic hardly showed any difference, the higher harmonics were excited a lot more. Especially the fifth and sixth harmonic vibrated with much greater velocities.
In Fig. 11, smaller influence on the frequency spectrum at high Φ can be seen. The second harmonic was increased only very little in amplitude, and so were the third and fourth. The BPF again was decisively amplified, and so were 6 \cdot f_0 and 7 \cdot f_0.

The effects to the low multiples of the fan rotation frequency f_0 can be examined further in Fig. 12 to Fig. 19. The first harmonic 1 \cdot f_0 (Fig. 12 and Fig. 16) showed a strong radial vibration mode over the entire blade with maximum amplitudes at the blade tips. Cross wind influence induced little change to the blade mode, only a minor asymmetry could be distinct towards the blade shown at the bottom.\(^2\) The second harmonic (Fig. 13 and Fig. 17) modes look very similar, with lower amplitudes. The magnitude of excitation was increased significantly by cross wind for Φ = 0.17 (Fig. 13), and a bit less for Φ = 0.29 (Fig. 17). Under cross wind influence, the fifth harmonics (Fig. 14 and Fig. 18) were evoked to show strong vibration at the blade tips, while the reference configurations in the left figures showed comparably small amplitudes in excitation. A very interesting pattern could be found at f = 6 \cdot f_0, where wind caused the blade modes to oscillate strongly around a radial strip of low vibration across the fan blades.

In conclusion, the cross wind tended to increase the blade vibration for the higher harmonics. At the lower flow rate operating point, this was true for nearly all the harmonics, while at the higher Φ, especially the BPF and the sixth harmonic were excited more strongly.

\(^2\) Blade locations in the plot do not refer to a position in space with respect to the wind direction. Each surface scan point was probed for over 800 revolutions of the fan with a co-rotating beam. The spectral representation in the surface velocity plots shown here has no reference to the angular position of the blade.
Fig. 12: Surface velocity around $1f_0$, without wind (left, $\Phi=0.17$) and with wind (right, $\Phi=0.17$)
Fig. 16: Surface velocity at $1/f_0$, without wind (left, $\Phi=0.28$) and with wind (right, $\Phi=0.29$)

Fig. 17: Surface velocity at $2/f_0$, without wind (left, $\Phi=0.28$) and with wind (right, $\Phi=0.29$)

Fig. 18: Surface velocity at $5/f_0$, without wind (left, $\Phi=0.28$) and with wind (right, $\Phi=0.29$)

Fig. 19: Surface velocity at $6/f_0$, without wind (left, $\Phi=0.28$) and with wind (right, $\Phi=0.29$)
5. Comparison of Blade Vibration to Acoustic Radiation

The results from the surface vibration measurements can be compared to acoustic measurements. The test setup in the aeroacoustic wind tunnel is described in Section 2. To avoid additional sound sources, no auxiliary fans could be operated for the acoustic measurements. Therefore, only the results of blade vibration at the lower flow rate (compare Fig. 10) are compared to microphone measurements.

Fig. 20 and Fig. 21 respectively plot the low frequency spectra without and with cross wind influence for surface velocities $u = u_{\text{surface}}$, and the sound pressure level $SPL = L_p$ in dB.

It was found that the harmonics of the fan rotation frequency $f_0$ also reflected as peaks in the $L_p$ curve, both with and without wind influence. Without wind (Fig. 20), especially the third harmonic was dominant, while the blade passing frequency did not stand out in the low frequency spectrum. All other harmonics peaked around the same level of sound pressure. Under the influence of cross wind at the fan inlet, the A-weighted over all sound pressure level $L_{p, \text{rms}}$ was increased by 4 dB(A). While increasing the broad band noise for all frequencies, the addition to $L_{p, \text{rms}}$ was by great part generated at the rotation frequency $f_0$ and the blade passing frequency $5 \cdot f_0$, which became the most dominant frequency in the SPL spectrum. The third harmonic even decreased in amplitude. Comparing this result to the effect the cross wind had on the blade vibration, a correlation between the strong increase in blade tip vibration at blade passing frequency (see Fig. 14 and Fig. 18), and the additional generation of radiated noise at this frequency could be found.

![Fig. 20: Sound pressure level and surface vibration without wind influence](image)

![Fig. 21: Sound pressure level and surface vibration with 10 m/s cross flow at the fan inlet](image)
6. Conclusion

It was possible to gain more insight into the fan behavior under cross wind influence. LDA measurements of the resulting ducted flow field behind the fan outlet showed significant changes in the velocity distribution. For all tested fan flow rates, the velocity profile was shifted in wind direction, less so at higher flow rates. Domains where the perpendicular inflow met the fan blades in opposing directions were accelerated in axial direction, with reduced circumferential velocities in the cross section, and vice versa. These effects were most dominant with the fan operating close to maximum static efficiency, i.e. with fan blades in optimized aerodynamic conditions. Cross flow at the fan inlet also increased turbulence intensity in the outlet flow field, and introduced an asymmetry to it. At low fan flow rates, $Tu$ became maximum where blade motion opposed the ambient inlet flow, i.e. the blade flow was presumably destabilized with greater turbulence being produced from a detaching blade flow. With increasing fan flow rate, this region was found to be relatively less turbulent. Instead, turbulence was intensified where inlet cross flow had the same direction as the blade motion. The nature of the asymmetry of the produced velocity profile was therefore found to depend strongly on the local relative flow field at the fan blade. If it is in stable conditions, then cross flow may help to accelerate the fluid more effectively, but it can also destabilize the flow around the fan blade and intensify turbulence, decreasing the effective local mean velocities.

Blade surface vibration was measured at two very different fan operating points, showing very little difference in the distribution and magnitude in the absence of additional cross flow at the inlet. Under the influence of cross wind, the harmonics of the rotation frequency were excited more strongly. At both tested fan flow rates, especially the blade passing frequency was amplified in its reaction to cross flow at the inlet. At the higher flow rate, the overall increase in surface vibration was smaller than at the lower fan flow rate, where almost each harmonic was significantly intensified by cross wind. Fan blade vibration was found to be a lot more affected by cross flow at low fan flow rates.

The excitation of the blade passing frequency was also found in microphone measurements of the acoustic radiation of the fan operating at a low flow rate. Overall, cross wind increased the sound pressure level at the tested position by 4 dB(A). Large contributions to this were made at the base frequency, and at the blade passing frequency, at which was also surface vibration of the fan blade tip was also highly excited by cross wind. It is well possible that the discovered additional blade tip vibration contributed to the increase in noise at this frequency.

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