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UNSTEADINESS OF BLADE-PASSING FREQUENCY TONES OF AXIAL FANS

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The unsteadiness of fan tones at blade passing frequency, is a well known and much observed problem and is often in the same order as the overall sound attenuation by means of innovative active or passive measures. This cannot be explained by a spatial inflow distortion which in general is considered as the source for steady-state BPF-tones from isolated fan rotors. This study reveals that even in test rigs with a certain undisturbed volume upstream the fan according to the current standards, the fan tones undergo a distinctive unsteadiness. By analyzing both acoustic and flow field data for three inflow conditions, it is shown that the mechanism which lead to the unsteadiness of the tones are fluctuations in the circumferential and radial inflow velocity components. This indicates that the disturbed inflow is induced by the large scale environment far upstream the fan intake. A hemispherical inflow conditioner proved to be most effective to suppress temporal fluctuations in the inflow.

1. Introduction

Typically the acoustic spectrum of an axial fan shows distinct tonal peaks at blade passing frequency (BPF) and a broadband floor. Theoretically, these tonal peaks should not occur in case of an isolated axial fan that is running at low speed, i.e. when (i) struts and guide vanes are non-existent or very far up- or downstream, (ii) the inlet geometry is symmetric and (iii) the circumferential tip Mach number is very low[1]. However, fan measurements carried out in state-of-the-art test rigs show that it is practically impossible to find fan spectra without tones at BPF and its higher harmonics. In a previous study[2], those tonal components at blade passing frequency were attributed to vortex-like coherent flow structures induced by the room from which the fan takes the air. It is essential to note that the room was much larger than the region of undisturbed inflow as required by standards for fan noise test rigs. The detected flow structures act as a stationary spatial distortion of the ingested flow and interact with the rotating blade. Eventually this results in periodic pressure fluctuations on the blades and the emission of tonal sound at blade passing frequency with *constant* amplitude. However, in reality the amplitude of the tones is *not* constant, as seen in Fig. 1, which is usually hidden by looking at time or ensemble averaged spectra only. Spectra from three different windows of a time signal representing sound from a fan are shown in Fig. 1. The blade passing frequency is indicated by the value 1 of a Strouhal number defined accordingly. It is evident that the tonal peak shows a temporal variation up to 6 dB. Even when characterizing the level of the tone by integrating the spectral power of the tones over a small bandwidth (a method proposed recently by Kohlhaas and Carolus[3]), the tone scatters from 73 dB to 78 dB. This unsteadiness of fan tones has been addressed only in a few studies. Barry and Moore[4] probably were among the first who brought the unsteadiness of fan tones into focus. In case of experimental data from outdoor tests of an axial

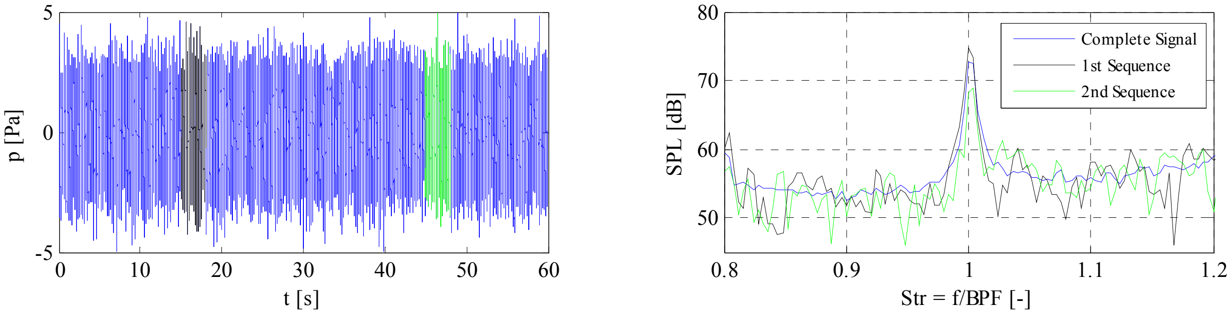


Figure 1: Typical sound signal from an axial fan (left) and spectral analysis of three time windows (right)

fan typically used in ventilation systems, they found temporal scatter of up to 20 dB. They attributed this scattering to time varying distortions like atmospheric turbulence or an unsteady annulus boundary layer but did not perform any flow field measurements. Another study is due to Hanson[5]. In his experiments he investigated the sound from large axial fans taking air from the free atmosphere. He found tones varying by up to 10 dB. By comparing the time histories of measured inlet velocities, blade surface pressures and the variation of the blade passing frequency tone, he proposed a link between the unsteadiness of the ingested flow and the tone. He explained his findings with the presence of atmospheric eddies which were elongated during suction and chopped by the blades of the fan. Since his experiments have been performed on an outdoor test rig with the presence of ambient wind, this is a quite plausible explanation but not directly applicable to low turbulence acoustic test sections with a large undisturbed volume upstream of the fan section. Balombin[6] investigated the steadiness of fan tones for several inflow conditions by using a probability density function technique to quantify the degree of fan tone steadiness. In his study, a fan section with a rotor/stator setup has been investigated. He reduced the fan speed to a level at which the blade passing frequency tone due to rotor/stator interaction is cut-off, i.e. does not propagate according to the theory of Tyler and Sofrin[7]. Among others it was found that an hemispherical honeycomb/screen inflow control device, which is typical of those used to achieve in-flight conditions, reduces the unsteadiness of the tones at blade passing frequency remarkably. Since no field measurements of the inflow have been carried out, Balombin supposed that the inflow control device might introduce spatially steady inflow distortions which leads to a higher steadiness of the residual tone in cut-off region. For the cut-on frequencies, he explained the reduction of tones unsteadiness with a supposed removal of random turbulence. However, it has been shown by the authors that the inflow control devices used in this study do not introduce any spatially steady inflow distortions but rather reduce them significantly[2]. Hence, the explanations of Balombin for the reduction of the unsteadiness while using the inflow control device in case of the cut-off tones can not be applied here. All these studies have in common that they explain the unsteadiness of BPF-related tones by an inflow that is varying in time. The time variation is related to obvious reasons like atmospheric disturbances or unsteady boundary layers. However, in the case investigated in this study, a small axial fan (impeller diameter 0.3 m) takes air from an empty, very large (4.50 m×3.50 m×3.23 m) room in a laboratory environment. There are neither atmospheric cross-winds nor an obvious source of turbulence. It is the objective of this study to quantify the mechanisms which lead to unsteady BPF-related tones. Eventually the results may serve to specify more adequately test rigs for standardized fan noise measurements.

2. Methodology

2.1 Experimental test rig

The standardized test rig used in this study is shown schematically in Fig. 2. The ducted axial fan takes the air from a large room (here the semi-anechoic chamber at the University Siegen) and

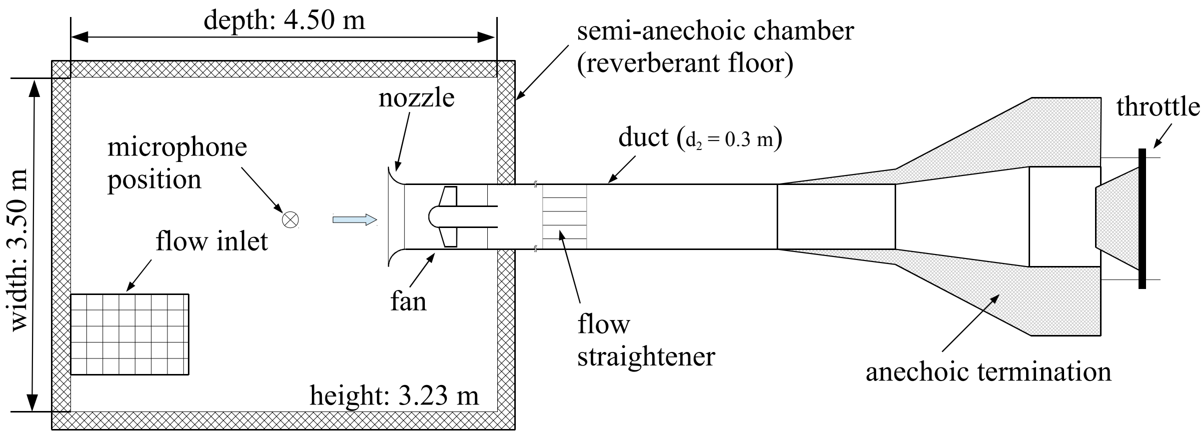


Figure 2. Top view of the standardized test rig for acoustic measurements (not to scale).

exhausts into a duct with an anechoic termination. The flow inlet is located off-center in the reverberant floor. The operating point is controlled by a throttle downstream of the termination, while the flow rate is determined by a calibrated hot film probe in the duct. The low pressure fan unit contains five cambered and swept blades which are designed with an in-house design software for axial fans. The rotor is manufactured and balanced with very high precision to avoid any non-aeroacoustic sound sources. A bell mouth type inlet nozzle with a ¼ rotor diameter radius is employed. Thin supporting struts are positioned one rotor diameter downstream of the rotor. No other obstructions are present. Table 1 shows the main fan characteristics. The dimensions of the semi-anechoic chamber are much larger than required by the current standards, which specifying an undisturbed volume of two diameters in each direction upstream of the fan[8,9].

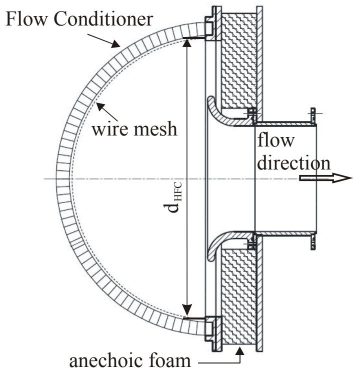
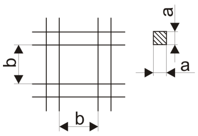
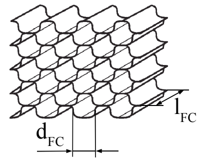
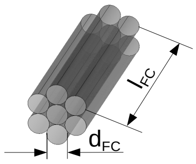
Table 1. Main characteristics of the fan.

Duct diameter d_2	0.3 m	Hub diameter d_1	0.135 m
Number of blades z	5	Tip clearance ratio s/d_2	0.1 %
$\mathrm{Re} = \pi \cdot d_2^2 \cdot n / \nu$ at d_2	$9.36 \cdot 10^5$	Circumferential $Ma = \pi \cdot d_2 \cdot n / a$ at d_2	0.139
Rotational speed n	3000 min^{-1}	Design flow rate $\dot{V}_{opt}$	0.65 m^3/s
Design flow rate coefficient $\varphi_{opt} = 4 \cdot \dot{V}_{opt} / (\pi^2 \cdot d_2^3 \cdot n)$	0.195		

The acoustic measurements are conducted by using a microphone (Brüel & Kjaer type 4190) located on the fans axis, as indicated in Fig. 2. The time signal of the sound pressure is measured by the microphone with a distance of 1.3 m from the fan and captured with a sampling frequency f_s = 25.6 kHz. Two inflow conditioners were used (see Table 2) to homogenize the inflow and allow an investigation of the influence of the inflow conditions on the acoustic emission. The Hemispherical Flow Conditioner (HFC) consists of a combination of a fluid conditioner structure, which reduces the lateral turbulences, and a downstream layer of wire mesh to reduce axial disturbances[10]. Other geometric dimensions are shown in Table 2 and based on recommendations in the literature[10-12]. The hemispherical device can be flange mounted to the nozzle of the test rig, see Table 2. A second inflow conditioner is a plane layer of small tubes, inserted into the duct upstream the fan (Tubular Flow Conditioner, TFC). The axial position of the TFC with respect to the rotor is shown in Fig. 3. Preliminary acoustic studies showed that the self noise of the inflow conditioners is negligible.

The flow field in the intake is measured in the measurement plane according to Fig. 3 by hot wire anemometry. To capture spatial variations of the velocities a cross section of the duct $0.96d_2$ upstream of the leading edge of the fan is scanned by a 3D hot-wire probe (Model 1299A from TSI

Table 2. Inflow conditioners.

HFC		diameter ratio of the device $d_{HFC} / d_2 = 2.84$ 	solidity of the wire mesh $\sigma = 1 - b^2 / (a+b)^2 = 0.44$  length ratio of the flow conditioner element $l_{FC} / d_{FC} = 10$ 
TFC		length ratio of the flow conditioner element $l_{FC} / d_{FC} = 10$	

Incorporated). The probe consists of three wires arranged crosswise and is connected with a universal anemometry system Streamline 90N10 from Dantec Dynamics. The anemometer operates in a constant-temperature mode. The measured signals are temperature corrected by using a temperature probe A8B from Dantec Dynamics. A 24-bit data acquisition system (Module PXI-4495 from National Instruments) is used, with which up to 16 channels can be measured synchronously. The probe is calibrated in a low turbulence wind tunnel. A slot extending over the half of the circumference is cut in the duct, so that the probe can protrude into the duct. Preliminary investigations have shown that the influence of this slot on the flow is small.

The orientation of the polar coordinate system for the hot wire measurements is indicated in Fig. 3 (right). The probe is attached to an automatic three axis positioning system from Isel, which has a positioning accuracy of 0.02 mm. The measurement points in the cross sectional plane of the duct have an angular distance of $\Delta\rho = 5°$ and radial distance of $\Delta r = 5$ mm, respectively. Together with the center of the plane, this leads to 2017 measurement points. All measurements are conducted over a time interval of one second with a sampling rate of $f_s = 25.6$ kHz. Note, that the measurements always were conducted at a rotational speed of $n = 3000$ min^{-1} and at the fan design operating point $\varphi_{design} = 0.195$. This leads to a blade passing frequency of $BPF = n \cdot z = 250\,\mathrm{Hz}$.

2.2 Signal analysis

To examine the unsteadiness of fan tones at blade passing frequency, the measured acoustic time signal has to be filtered accordingly. For this purpose, a Butterworth filter was designed. The band-pass filter has an order of $N = 4$ and a first and second cutoff frequency of $f_1 = 245$ Hz and $f_2 = 255$ Hz, respectively. The filter width of 10 Hz on the one hand allows a deviation of the rotational speed of $\pm$ 60 min^{-1} during the measurements, which was strictly satisfied by the used experimental equipment, and on the other hand avoids an artificial amplitude variation due to a too small filter width. For the filtering the function *filtfilt* in Matlab™ R2012a is used, which performs the filtering by processing the input data in both the forward and reverse directions[13]. An envelope is then calculated to capture the temporal variation of the amplitude of the tone at blade passing frequency by detecting the local maxima of the filtered time signal and connecting them using a piecewise cubic polynomial interpolation. The result of this filtering procedure is shown for instance in Fig. 4

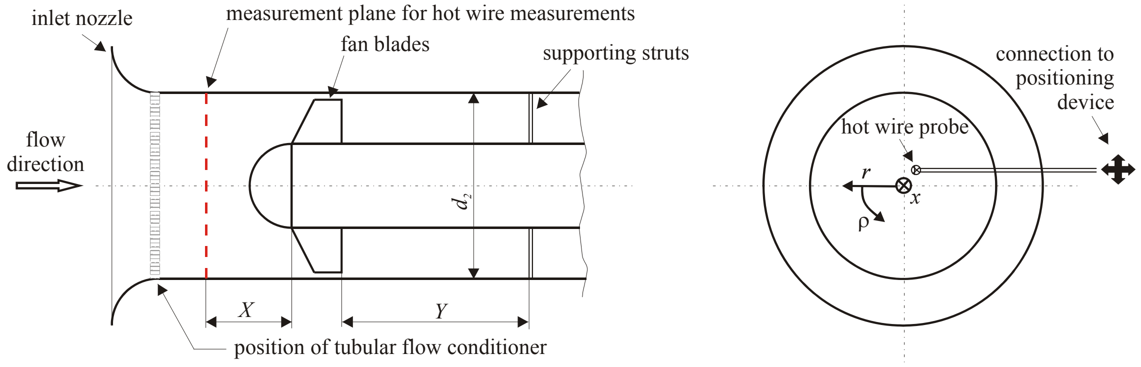


Figure 3. Lateral view of the axial fan section investigated (left; $X = 0.96d_2$, $Y = d_2$) and top view (right).

on the basis of the measured acoustic time signal for the case of free inflow, see Tab. 2. The frequency is depicted in terms of the Strouhal number $Str = f/BPF$. Figure 4 (left) shows the signal in the time domain, while in Fig. 4 (right) the microphone data has been transformed in the frequency domain. The blue curves represent the raw, i.e. unfiltered signals, the green curves the signals filtered by the mentioned band-pass filter. Fig. 4 (right), it is clearly visible that the band-pass filter causes a signal that only contains the content of the peak at the blade passing frequency. All other frequencies are damped. By transforming this filtered signal back to the time domain one obtains the time signal of the peak at blade passing frequency, see the green curves in Fig. 4 (left). The envelope of this time curve (see red curves in Fig. 4, left) then represents the temporal variation of the amplitude of the tone at the blade passing frequency.

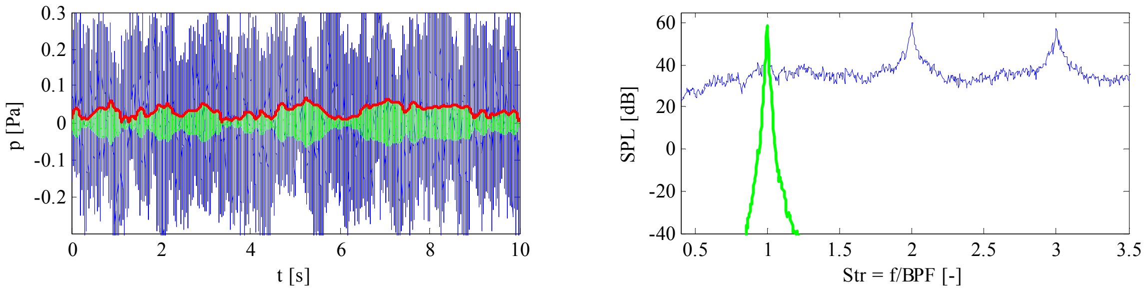


Figure 4: Filtered and unfiltered signals of the acoustic microphone data for the case of free inflow in the time (left) and frequency (right) domain: raw signal (−), filtered signal (−), envelope (−).

Beside the filtering technique, spectrograms are calculated to observe the temporal variation of the tones in the frequency domain. By using the Short-Time Fourier Transform (STFT), spectrograms provide here the sound pressure level (SPL) as a function of frequency and time. This is done by breaking up the time signal to be analyzed in several segments. Each of these segments is then multiplied by a window function and is Fourier transformed. The square of the STFT gives then the spectrogram. The resolution of the spectrogram depends on the number of segments: a high number of segments results in a high resolution in time, but low resolution for the frequency. For a small number of segments it is the opposite. In this study, the number of segments is chosen to be $N_{seg} = 50$ as a reasonable compromise between time and frequency resolution. By using the function *spectrogram* in Matlab™ R2012a with an overlapping of 50% and the parameters *window = hann(nfft), noverlap = nfft/2, nfft = length(time signal)/number of segments* for the measured time signal of t = 10 s, this results in a frequency resolution of Δf = 5 Hz and a time resolution of $\Delta t = 0.1$ s. The SPL is then calculated by using a reference pressure of $p_0 = 2 \cdot 10^{-5}$ Pa.

3. Results

The unsteadiness of the fan tones for all of the three different inflow configurations is shown in Fig. 5. Figure 5a shows the envelopes of the filtered microphone signals as a function of time, evaluated as shown in Fig. 4. The envelope in case of the free inflow shows a clear wavy behavior and hence a strong temporal variation of the tone amplitude. This variation is dramatically reduced by the use of the HFC. The TFC shows a similar amplitude as the free inflow configuration but a less wavy behavior. The standard deviation σ of the corresponding envelopes quantifies the tendencies for the unsteadiness of the different configurations: σ_{free} = 0.0145 Pa, σ_{HFC} = 0.0014 Pa and σ_{TFC} = 0.0116 Pa. In Fig. 5 b-d, the STFT of the acoustic microphone data for each of the different inflow configurations is depicted. The ordinate shows the frequency, the time is plotted on the abscissa and the contours indicating the SPL. The temporal variation of the tones at blade passing frequency in case of the free inflow amounts up to 10 dB, whereas the use of the flow conditioning device results in a much more constant behavior of the tone amplitude. The TFC still shows a temporal variation of about 7 dB, the HFC only 2-3 dB. In case of the free inflow the first harmonic of the tone (*Str* = 2) is also very dominant. One can observe that at each minimum of the fundamental tone, the first harmonic is particularly high. In case of the HFC the first harmonic nearly vanishes. Another outcome is the broadening of the tonal peaks as a function of the inflow conditions. This effect is also described in Barry and Moore[4] as a result of an amplitude modulation of the tones due to a varying distortion, which ends up in skirts of the tonal peaks. This broadening can be clearly seen in the diagrams of the STFT: in case of the free inflow configuration the high amplitude areas at integer Strouhal numbers are scattered over a certain frequency range while the STFT shows confined lines when the devices are used.

The results of the hot wire measurements for the different inflow conditions are depicted in Fig. 6, where contour plots in terms of the standard deviation of the velocity components in polar coordinates are shown in a plane upstream the fan. The standard deviation σ represents the fluctuation of a value around his mean value and is used here as a measure for the unsteadiness of the signals. The free inflow configuration shows strong fluctuations for the circumferential velocity and radial velocity, respectively. The axial component varies less. The HFC reduces these fluctuations of all three components dramatically to a uniform and low level. The TFC also diminishes the temporal fluctuations of the circumferential and radial components. However, the temporal variation of the axial component is rather increased compared to the free inflow configuration. For the circumferential and radial velocity components, the results of the hot wire measurements reveal the same tendencies for the temporal variation as the acoustic data: the free inflow configuration shows strong fluctuations in time, the HFC causes almost constant conditions and the TFC still reduces the

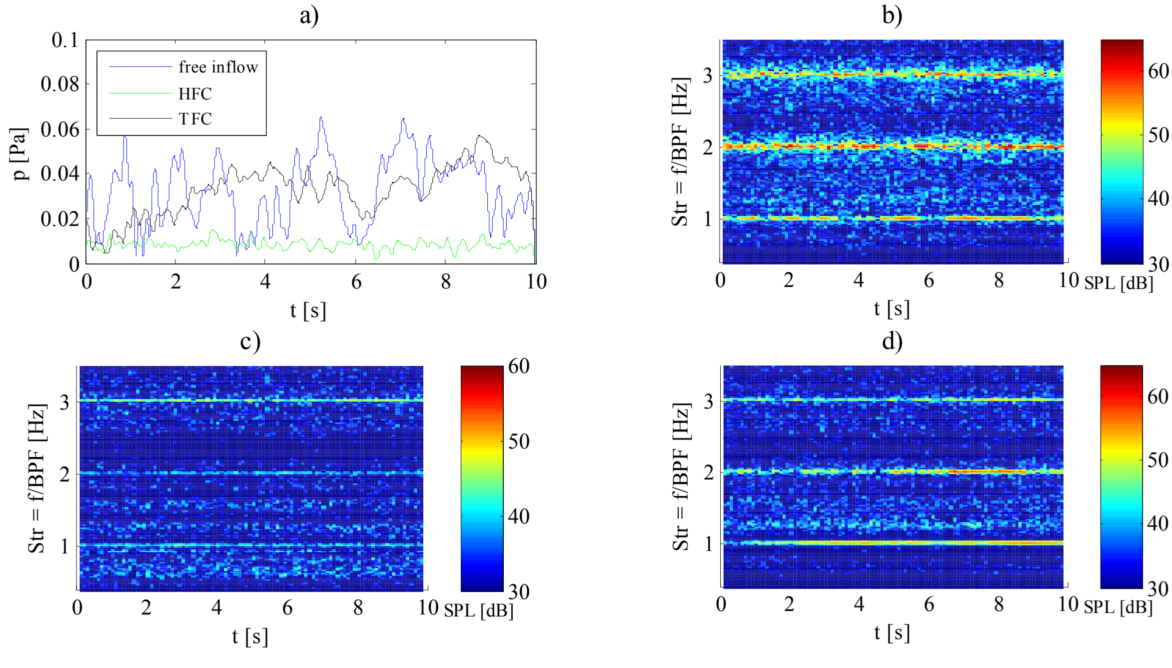


Figure 5: Unsteadiness of the acoustic microphone data for different inflow conditions: envelopes of the filtered time signals (a) and spectrograms for the cases of the free inflow (b), HFC (c) and TFC (d).

fluctuations but not as efficient as the HFC. One can therefore conclude that the temporal variations of the tones at blade frequency are closely linked rather to the conditions of the circumferential and radial velocities than to the axial component, since the fluctuations of the axial velocity in case of the TFC are even higher, but the temporal variation of the tones in the acoustic spectrum are lower compared to the free inflow configuration.

4. Conclusions

Time averaged acoustic spectra are often used in many studies but representing the actual acoustic emission of fans only partially. Analysis of acoustic data reveal a distinctive unsteadiness of the amplitude of the tones at blade passing frequency even on test rig with an undisturbed region upstream of the fan section that is much larger than required by the current standards. Using flow conditioning devices these variations can be reduced, which shows that the unsteadiness of the tones is closely linked to the inflow conditions. By comparing the unsteadiness of hot wire and acoustic measurements, respectively, it has been proved that the mechanism which lead to the temporal variations of the BPF related tones are fluctuations in the inflow velocities. In addition, it turned out that rather fluctuations in the circumferential and radial velocity components than the axial components cause the unsteadiness of the fan tones. Presumably, this unsteadiness of the inflow is induced by the flow conditions in the large room which is much larger than the recommended intake domain for standardized acoustic test rigs. Other potential mechanisms such as unsteady secondary flow structures self-induced in the vicinity of the rotor, like tip vortex flows or vortices at the bottom of centrifugal fans[14], can be excluded here as a source for fan tone fluctuations at the blade passing frequency. Further investigations, preferably by numerical computations, should be conducted to study the unsteady flow conditions inside the large room. Moreover, these

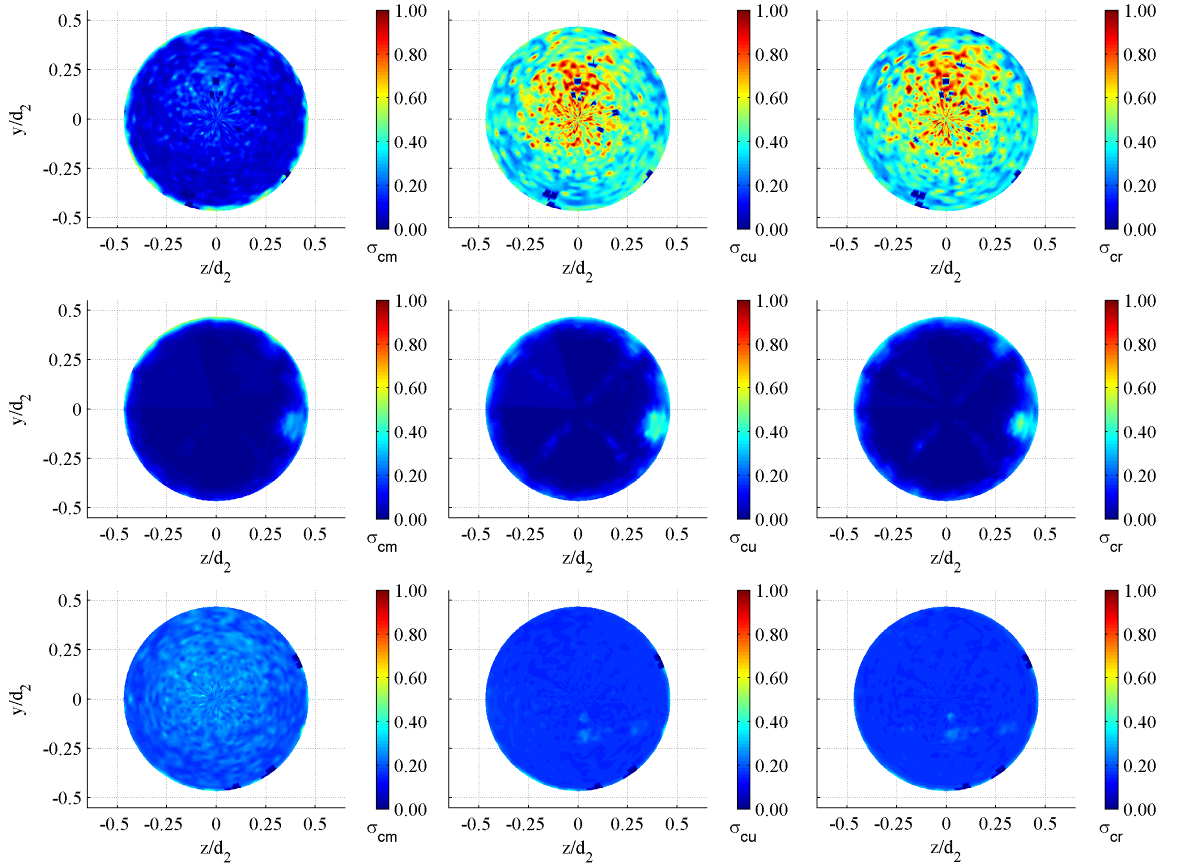


Figure 6: Contour plots of the standard deviation of the velocity components at the measurement plane upstream the fan for different inflow conditions: free inflow (top), HFC (middle) and TFC (bottom); standard deviation of the axial velocity (left), circumferential velocity (middle) and radial velocity (right).

findings are limited to that kind of test rig, i.e. sucking air out of a large anechoic chamber with a ducted fan instrumented with a nozzle, and need to be verified for different test conditions. To reduce the unsteadiness of the fan tones and eventually increase the reliability of the acoustic measurements, the actual origin of the inflow fluctuations needs to be investigated to find an environment which prevents the emergence of a disturbed inflow. However, since this will probably end up in a very large room, another opportunity would be the aid of flow device that carefully conditions the inflow. The hemispherical inflow conditioner, similar to those used in aircraft engine tests, proved to be most effective to suppress temporal fluctuations in the inflow.

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REFERENCES

[1] Goldstein, M. E., *Aeroacoustics*, McGraw-Hill, USA, (1974).

[2] Sturm, M., Carolus, Th., *Large Scale Inflow Distortions as a Source Mechanism for Discrete Frequency Sound from Isolated Axial Fans*, AIAA 2013-2105, 19th AIAA/CEAS Aeroacoustics Conference, May 27-19, Berlin, Germany, (2013).

[3] Kohlhaas, M., Carolus, Th., *Acoustic Minimization of Rotor-Stator Interaction Noise of Axial Fans by Trailing-Edge Blowing*, GT2014-25113, Proceedings of the ASME Turbo Expo 2014, June 19-20, Düsseldorf, Germany, (2014).

[4] Barry, B., Moore, C. J., *Subsonic Fan Noise*, Journal of Sound and Vibration, Vol. 17 (2), pp. 207-220, (1971).

[5] Hanson, D. B., *Spectrum of Rotor Noise Caused by Atmospheric Turbulence*, J. Acoust. Soc. Am., Vol. 56, No. 1, (1974).

[6] Balombin, J. R., *Variation of Fan Tone Steadiness for Several Inflow Conditions*, AIAA 11th Fluid and Plasma Dynamics Conference, Seattle, Washington, USA, (1978).

[7] Tyler, J., Sofrin, T., *Axial Flow Compressor Noise Studies* SAE Technical Paper 620532, (1962).

[8] ISO 13347, *Industrial Fans - Determination of Fan Sound Power Levels Under Standardized Laboratory Conditions,* International Organization for Standardization, 1st Edition, Siwtzerland, (2004).

[9] DIN EN ISO 5801. *Industrial Fans - Performance Testing Using Standardized Airways,* Deutsches Insitut für Normung e. V., Beuth-Verlag, Berlin, Germany, (2011).

[10] Scheimann, J., Brooks, J. D. *A Comparison of Experimental and Theoretical Turbulence Reduction from Screens, Honeycomb and Honeycomb-Screen Combinations*, 11th Aerodynamic Testing Conference, Colorado Springs, USA, AIAA Technical Papers, pp. 129-137, (1980).

[11] SAE International, *Methods of Controlling Distortion of Inlet Airflow During Static Acoustical Tests of Turbofan Engines and Fan Rigs*, Aerospace Information Report, (1985).

[12] Groth, J., Johansson, A. V., *Turbulence Reduction by Screens*, Journal of Fluid Mechanics, Vol. 197, pp. 139-155, (1988).

[13] The MathWorks, Inc., *Product Help for Matlab R2012a*, (2012).

[14] Wolfram, D., Carolus, T., *Experimental and numerical investigation of the unsteady flow field and tone generation in an isolated centrifugal fan impeller*. Journal of Sound and Vibration, Vol. 329, pp. 4380-4397, (2010).