CONTROLLED INFLOW TURBULENCE – AEROACOUSTIC INTERACTION OF AXIAL FANS WITH AN ACTIVE TURBULENCE GRID

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SUMMARY

An experimental study of the flow field of an active turbulence grid and the sound emission of an axial fan with this grid is presented. It was shown that the mean flow field of heat exchangers can be reproduced with the grid and similar trends could also be generated in the spatial distribution of turbulence level comparing heat exchanger and active turbulence grid. The sound emissions of the axial fan with heat exchanger and the axial fan with active turbulence grid show similar characteristics. It was demonstrated that there is a linear relationship between the sound radiation of the first blade passing frequency and the spatial variance of the turbulence level.

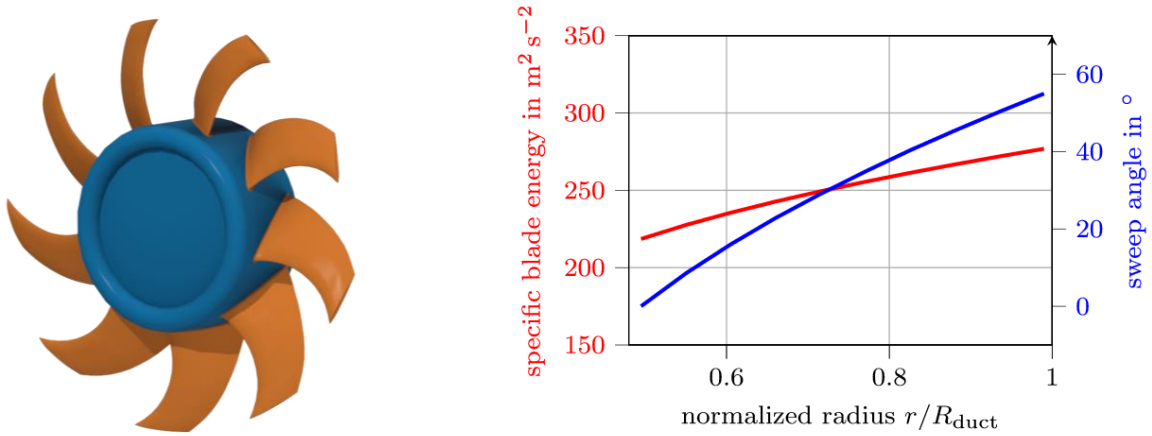
INTRODUCTION

Turbulence grids are used in wind tunnels and axial fan test rigs to generate increased turbulence intensities, modified length scales and defined flow fields. A distinction is made between active and passive turbulence grids. The passive turbulence grids consist of rigid struts, which generate a defined orifice and blocking of the flow field. Active turbulence grids, on the other hand, consist of shafts with attached plates, which are called turbulence generators. The shafts are driven by motors located outside the flow, and a rotation of the shafts including the turbulence generators is generated. The rotation of the turbulence generators can be used to influence the turbulence intensity, the turbulent length scale and the homogeneity of the flow field. The great advantage of the active turbulence grid is that a large number of different flow conditions can be generated with one grid by adjusting the rotation speed of the different shafts. Active turbulence grids have been used for basic investigations of flows and turbulence properties in wind tunnels since 1991 [1, 2]. They are usually operated in a square channel and the flow generated by the grids is analyzed downstream. The main advantage of active turbulence grids is the great variability and the associated possibility to generate different flow fields [3, 4]. In a novel approach, an active turbulence grid is to be used for the replication of real flows fields and the generation of inflow turbulence for axial fans, apart from basic turbulence research. The advantages of the active

turbulence grid will be used to mimic the flow field of various heat exchangers. This offers the possibility to simulate different real scenarios and to investigate axial fans in a variety of inflow turbulences without having to investigate all necessary heat exchanger devices on site [5]. This real-world application of axial fans with suction-side heat exchangers is of particular importance because, on the one hand, they are used in a large number of devices in the direct vicinity of humans and, on the other hand, it is not yet fully understood why a suction-side heat exchanger can increase the sound radiation of axial fans [6, 7]. Also full CFD simulations of the flow field through heat exchangers are not possible nowadays [8]. In addition to this new field of application, it is also possible to adjust the turbulence characteristics of the flow independently of each other by means of active turbulence grids, in contrast to rigid turbulence grids. For example, it was shown that at low speeds of the turbulence generators of the grid, the integral length dimension can be varied while the turbulence level remains constant. At high speeds, on the other hand, it was possible to vary the turbulence level while keeping the length scale unchanged [9]. This property offers the possibility that the influences of the individual turbulent quantities on the sound pressure spectrum of axial fans can be investigated and thus a better understanding of the sound generation mechanisms can be derived. However, for the active turbulence grid to be used in this context, it must be investigated whether, on the one hand, a modulation of the inflow conditions can take place if a circular duct is placed behind the square channel of the active turbulence grid in which the axial fan operates. Such a change in the cross-sectional area has as yet unexplained effects on the generated turbulence of the grid. Furthermore, for the investigation of the sound radiation of axial fans, it must be ensured that the inherent noise of the active turbulence grid is low enough that no significant influence on the sound pressure spectrum takes place.

AXIAL FAN DESIGN

A forward skewed axial fan, which operates in a short duct segment with a diameter of D_{duct} = 500 mm, is used for the investigations. The axial fan, its specific blade energy and its sweep angle are shown in Fig. 1. The total diameter of the fan is D_{fan}= 495 mm, which means that the tip gap is s_{tip}= 2.5 mm. As is common in heat exchanger applications, the fan operates within a short inlet nozzle, which causes the head gap to vary along the axial length of the inlet nozzle. As shown in Fig. 1, the fan has z= 9 blades and is designed for a flow rate of $\dot{V}$= 1.4 m³/s at a rotational speed of n_{fan} =1486 rpm. The blades are based on a NACA 4510 profile and designed according the blade element theory [10]. The aerodynamic and aeroacoustics characteristics of the used axial fan can be found in the literature at various inflow conditions and in interaction with heat exchangers [7, 11, 12].



(a) Forward skewed axial fan. *(b) Specific blade energy and sweep angle.*

Figure 1: Investigated axial fan and its characteristics.

ACTIVE TURBULENCE GRID

An active turbulence grid is used to generate defined inflow conditions. This grid consists of 20 shafts, which can be positioned or rotated independently of each other via stepper motors. Figure 2 shows the active turbulence grid within the anechoic chamber of the axial fan test rig at the University of Erlangen-Nuremberg [13]. Turbulence generators, which have a square shape, are located on the twenty shafts. On each shaft there are 10 turbulence generators, which are moved simultaneously by one motor. This arrangement results in a grid width of M= 80 mm. The turbulence generators are located within a square flow-through area, which has dimensions of 800 mm x 800 mm. This area is the same as heat exchangers, which are often used for fans with a size of D= 500 mm [7]. By maintaining the same geometric parameters, the goal of the active turbulence grid is to reproduce the turbulence characteristics of heat exchangers and to better understand the interaction of turbulence and axial fans. The control and drive motors of the grid are located outside the flow area on another frame. These components are shielded by sound insulation boxes so that as little noise as possible from the motors influences the measurement results of the axial fans under investigation. The active turbulence grid has a total length of 330 mm with the center plane of rotation 200 mm away from the axial fan. Five B&K free-field microphones are used for the characterization of the emitted sound field. The microphones are aligned at a distance of R_{mic}= 1 m from the axial fan (see Fig. 2).

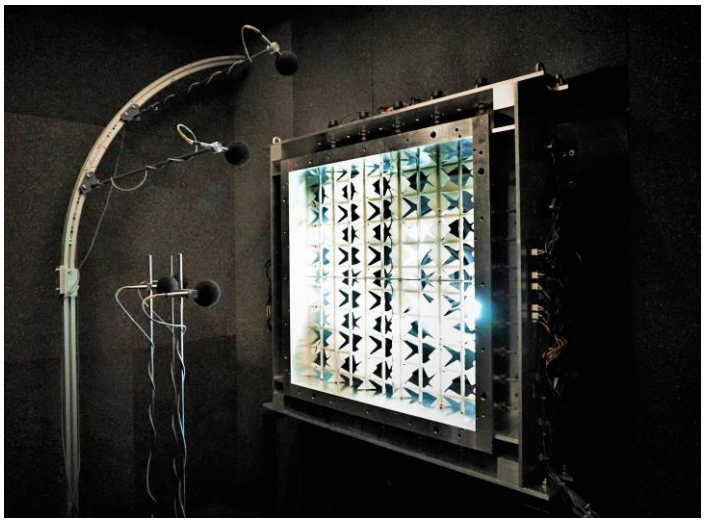


Figure 2: Active turbulence grid inside the anechoic chamber.

In order to better separate the acoustic influence of the active turbulence grid on the measurement results of the sound pressure from the axial fans under investigation, the inherent noise of the active turbulence grid was determined. The noise is low enough that it can be assumed that the intrinsic noise of the grid is masked by the fan noise [5]. Through a run-up and a resulting Campbell plot (see Fig. 3), it can also be determined that the tonal components generated by the active grid are related to the speed of the shafts. In addition, the linear relationship between the tonal components in the sound spectrum and the motor speed is related to the steps and micro steps of the motors used. These perform 200 steps and 3200 micro steps per revolution.

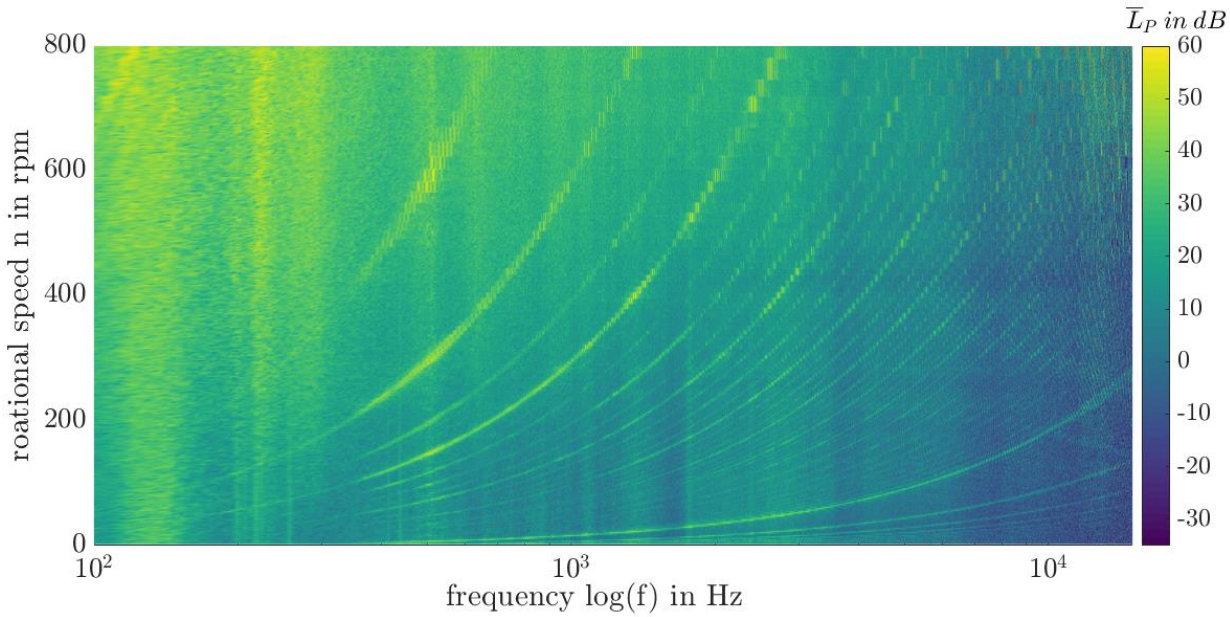


Figure 3: Campbell-plot of the self-induced noise of the active turbulence grid.

CONTROLLED INFLOW TURBULENCE

In connection with axial fans, the inflow turbulence at the leading edge of the axial fans plays a key role [14, 15]. The inflow turbulence interacts with the rotating fan at this point and generates a dominant sound source, which can be attributed to pressure fluctuations at the surface and the increased decay of turbulent structures. To investigate different inflow turbulence, the axial fan is replaced by a bladeless hub. This allows to determine the inflow turbulence at the exact position where the leading edge of the fan blades would actually be. To determine the flow field induced by the active grid, 3D hot-wire anemometry is used, which indirectly measures the velocities present in all three spatial directions for a measurement duration of t_s= 30 s and a sampling frequency of f_s= 48 kHz. One third of the duct segment is measured with 80 uniformly distributed measuring points [8]. From the velocity fluctuations at these points, the turbulence level and the integral length scale are determined. Global values for these turbulence properties are derived by a surface averaging. In Fig. 4, the global values of turbulence level and integral length scale are plotted in a characteristic map. Different operating modes of the active turbulence grid are shown with red dots in the map. The black squares are representative of four different heat exchangers with the dimensions of 800 mm x 800 mm. The distribution of the red dots shows that the active turbulence grid can vary the turbulence level in the range of [6.7 %; 16.3 %] and the length scale in the range of [3.7 mm; 36.2 mm]. In addition, a kind of pareto front [16] is recognizable in the measurement data. Especially the heat exchanger with the highest turbulence level can be reproduced very well in the mean values by the active turbulence grid. Here, the mean values in the length scale between heat exchanger and active turbulence grid deviate by only 2 mm. For heat exchangers in the turbulence range of 6.5 %, the active turbulence grid can reproduce the turbulence level, but the length scale deviates by 6 mm. The heat exchanger with the low turbulence level of 4.3 % cannot yet be reproduced by the active turbulence grid. Further investigations will show whether this range can be achieved in the map via other operating modes.

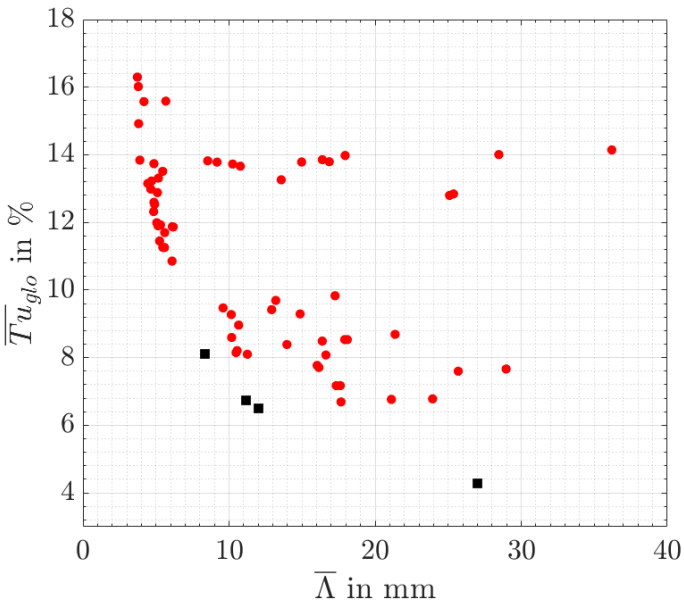
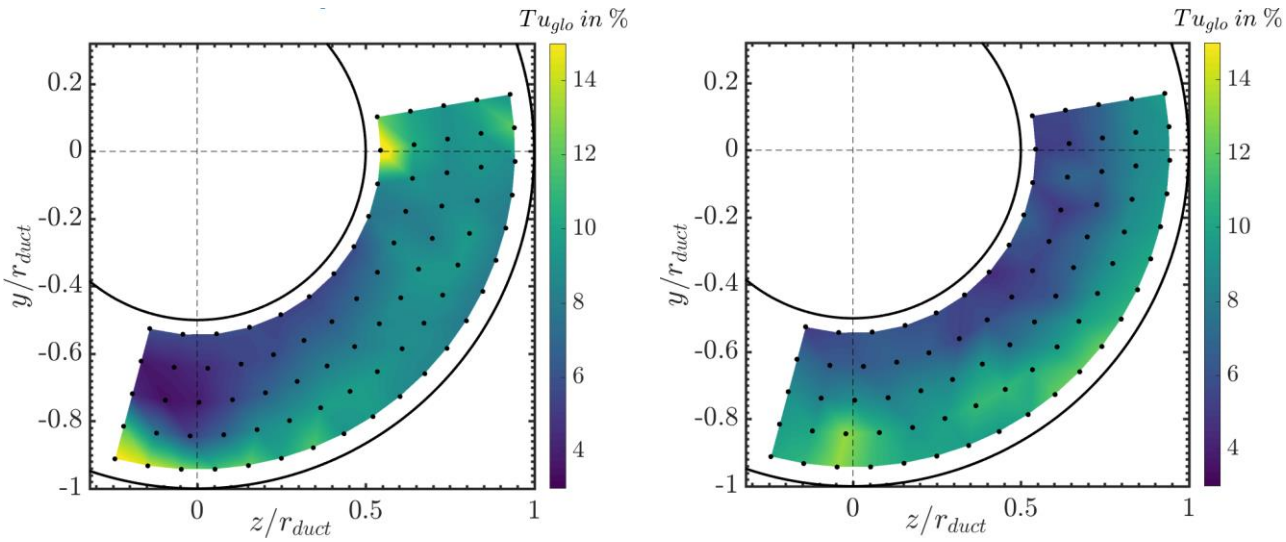


Figure 4: Characteristic map of the induced turbulence characteristics of heat exchangers (■) and different active turbulence grid modes(●).

Since the active turbulence grid seems to be able to reproduce the average flow values of heat exchangers, a comparison between the flow field downstream of a heat exchanger (Tu_{glo} = 8.12 %, λ = 8.34 mm) and the most similar operating mode of the active turbulence grid (Tu_{glo} = 8.15 %, λ = 10.48 mm) is shown in Fig. 5.



(a) heat exchanger. *(b) active turbulence grid.*

Figure 5: Comparison of field of turbulence level between heat exchanger and imitated flow field by active turbulence grid ($\dot{V}$ = 1.4 m³/s).

The surface plots of the turbulence level show that flow conditions similar to those of heat exchangers can be created with the active turbulence grid. It is noticeable that the flow field seems to be rotated by 90 degrees in the plots shown. Nevertheless, inhomogeneities in the flow field, which are mainly generated by the square housing of the heat exchanger and the active grid, can be reproduced [17]. The two-dimensional distribution of turbulence values also shows similar tendencies.

In addition to replicating the flow fields of heat exchangers, the active turbulence grid offers the possibility of varying individual turbulence characteristics independently of the others. Thus, the

characteristic map of the active turbulence grid (see Fig. 4) proves that, for example, the length scale can be varied in the region of Tu_{glo}= 14 % turbulence level, whereas the average turbulence level remains almost constant at this value. At a length scale of about λ= 5 mm, the opposite pattern is seen. In this range, the turbulence level can be changed while the length scale remains constant. Thus, the active turbulence grid allows a separation of the influences of the different turbulence characteristics on the sound spectrum of axial fans to be investigated independently of each other.

SOUND EMISSIONS OF AXIAL FAN AND ACTIVE TURBULENCE GRID

The active turbulence grid is not only intended to reproduce the flow field of the heat exchangers, but also to reproduce the sound radiation of axial fans positioned downstream of heat exchangers as best as possible. The sound pressure spectrum is used to evaluate the sound emissions on the suction side of the axial fans. A comparison of the active turbulence grid and the heat exchanger should only be made at the operating point at which the flow data are available. A change in the flow velocity within the active turbulence grid leads to a change in the grid Reynolds number, which in turn changes the turbulence level and the length scale generated by the grid [9]. Figure 6 shows the sound pressure spectrum of the forward skewed axial fan at the design point. Here the spectrum that is induced when a heat exchanger is placed in front of the axial fan and the spectrum when an active turbulence grid generates similar flow conditions are presented. In both cases, the sound measurements were made inside the anechoic chamber, on the suction side of the axial fan. Accordingly, the sound is influenced during its propagation to the microphones by the heat exchanger or the active turbulence grid, which are also located on the suction side of the axial fan. In this operating point there is a difference of 0.7 dB in the overall sound pressure level of the two variants. From this, follows that the overall sound pressure level is reproduced well by means of active turbulence grids. In addition, the sound pressure spectra have similar trends, in which the subharmonic peak and the higher harmonic blade passing frequencies are reproduced. Only the first blade passing frequency (1.BPF) is higher in the case with the active turbulence grid. This could be due to a higher turbulence spot in the flow field, which is not detected by the measurement in the third segment of the duct. At higher frequencies, the heat exchanger consistently induces a lower sound pressure level. However, this difference is not due to the flow field, but to the sound guiding properties of the heat exchanger. Czwielong et al. [18] have identified that the differences in the high frequencies are due to the periodic arrangement of the coolant tubes in the heat exchanger and to the thermoviscous losses due to the high number of cooling fins. Thus, the differences in the high frequency range can be corrected by knowledge of the heat exchanger's sound transmission characteristics.

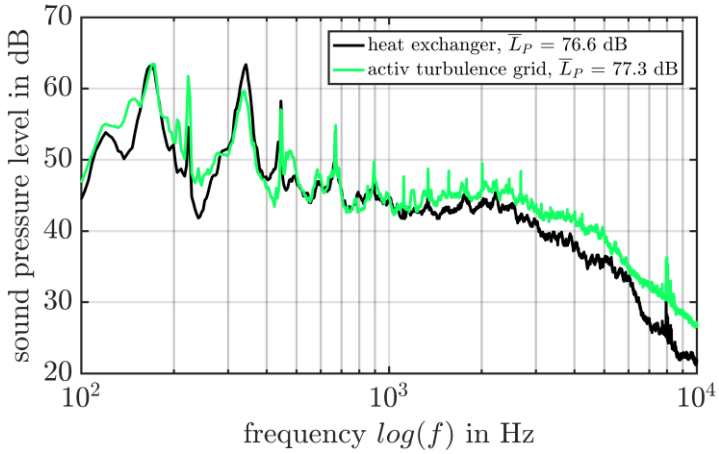
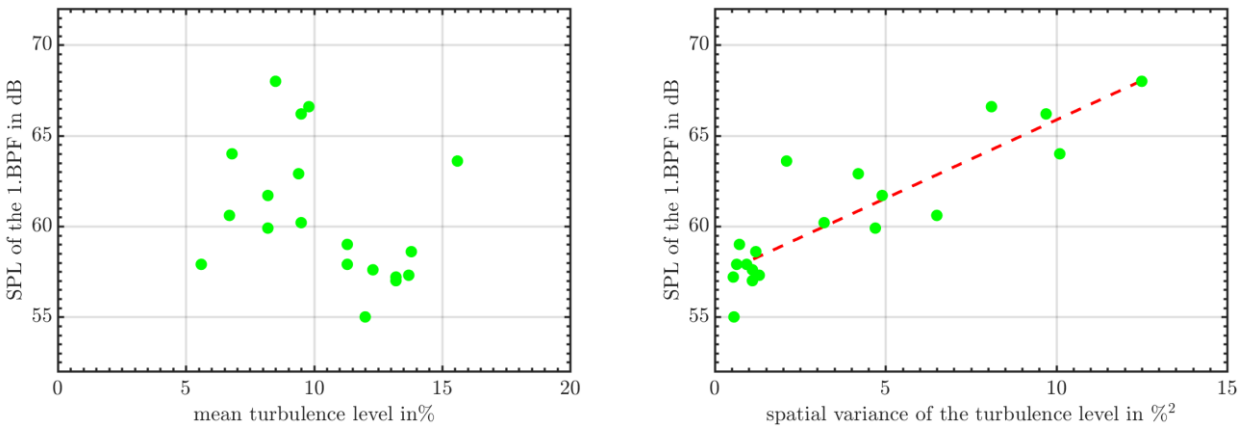


Figure 6: Sound pressure spectra of the heat exchanger (Tu_{glo} = 8.12 %, λ = 8.34 mm).and the active turbulence grid (Tu_{glo} = 8.15 %, λ = 10.48 mm) at the design point ($\dot{V}$ = 1.4 m³/s).

Based on the flow field investigations, the sound emissions of the axial fan were investigated in selected operating modes of the active turbulence grid. The sound pressure spectra obtained at the design point were analyzed, for example, on the basis of the sound radiation at the first blade passing frequency (1.BPF= z·n/60= 222.9 Hz, with z= 9 blades and n=1486 rpm). The first blade passing frequency is often a dominant tonal component in the sound pressure spectrum of low-pressure axial fans and should be kept as low as possible so that the fan noise is not perceived as annoying. Figure 7(a) displays the relationship between the acoustic emission of the axial fan at the first blade passing frequency and the spatial mean value of the turbulence level. From the data distribution, it can be concluded that no correlation between the mean spatial turbulence level and the sound emission at the first BPF can be found. This indicates that it is often not sufficient to give only mean turbulence levels when the sound pressure spectra of axial fans are discussed in literature. Figure 7(b) displays the relationship between the first BPF and the spatial variance of the turbulence level. It can be clearly identified that as the variance increases, the acoustic radiation of the axial fan increases at the first BPF. A linear fit is given as a red dashed line in the plot. Formula (1) is the equation of the linear fit curve, in which the turbulence level in percent is inserted.



(a) Spatial mean of turbulence level. *(b) Spatial variance of turbulence level.*

Figure 7: Influence of the spatial mean of the turbulence level and the spatial variance of the turbulence level on the sound radiation of the axial fan at the first blade passing frequency (1.BPF).

$$SPL_{1.BFF,Fit} = 0.8656 \cdot \mathrm{Var(Tu)} + 57.2\ \mathrm{dB} \tag{1}$$

The existence of a linear relationship between the sound radiation at the first BPF and the spatial variance of the turbulence level indicates that the knowledge and specification of spatial parameters is critical for the understanding of the sound emission of axial fans. The spatial variance offers the possibility not only to represent discontinuities in the turbulence level by surface plots but also to describe these discontinuities in a statistical value. A high spatial variance means that spatial gradients in the turbulence level are present. When the fan blade moves through these gradients, it experiences a change in the blade forces, which in turn leads to a change in the sound radiation of the 1.BPF. Low values of the spatial variance of the turbulence level indicate a homogeneously distributed turbulence level. The fan blade moves through fewer gradients and therefore experiences fewer changes in blade forces as it rotates through the flow field. As a result, the sound emission decreases at the 1.BPF. In heat exchanger applications, strong inhomogeneities in the flow field may occur. To minimize sound radiation at the 1.BPF, these inhomogeneities in the flow field should be avoided as far as possible.

CONCLUSION

An active turbulence grid was presented, which can be used for the generation of inflow turbulence of axial fans. The acoustic characterization of the active turbulence grid showed that the inherent noise of the grid is low enough and no significant influence on the sound pressure spectra of the axial fans under investigation is to be expected. In addition, it was found that the tonal peaks of the active turbulence grid depend on the rotational speed of the shafts and the number of steps of the stepper motors. On the basis of 3D hot-wire investigations, it was demonstrated that the active turbulence grid can be used to generate a large number of different inflow conditions, and that the mean turbulence characteristics and the spatial turbulence characteristics induced by heat exchangers can be reproduced by means of the active turbulence grid. The sound radiation of the investigated forward skewed axial fan between a heat exchanger and the active turbulence grid, which both induce comparable inflow conditions, also had similar tendencies. Here, differences in the high frequency range could be attributed to the absence of cooling fins. Based on a large number of sound radiation investigations, a linear relationship was found between the sound pressure level at the first blade passing frequency and the spatial variance of the turbulence level. No correlation was found between the sound radiation at the 1. BPF and the spatial mean of the turbulence level. This result indicates on the one hand that the spatial flow field significantly influences the sound radiation of axial fans and on the other hand that for the understanding of the sound generation mechanisms at the axial fan not only mean values are sufficient.

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